Chilled Water Plant Design Guide

energydesignresources

December 2009

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1. OVERVIEW

This document is a design guide for chilled water plants. It identifies the target audience, describes the organization of the material, summarizes what is in each of the chapters, and offers guidance on how to use the document.

Introduction

Many large buildings, campuses, and other facilities have plants that make chilled water and distribute it to air handling units and other cooling equipment. The design operation and maintenance of these chilled water plants has a very large impact on building energy use and energy operating cost.

Not only do chilled water plants use very significant amounts of electricity (as well as gas in some cases), they also significantly contribute to the peak load of buildings. The utility grid in California, and in many other areas of the country, experiences its maximum peak on hot summer days. During this peak event, chilled water plants are often running at maximum capacity. When temperatures are moderate, chilled water plants are shut down or operated in stand-by mode. This variation in the rate of energy use is a major contributor to the peaks and valleys in energy demand, which is one of the problems that must be addressed by utility grid managers.

Most buildings and facilities that have chilled water plants have special utility rates where the cost of electricity depends on when it is used and the maximum rate of use. For instance, PG&E has five time charge periods: summer on-peak, summer mid-peak, summer off-peak, winter mid-peak and winter off-peak. The price of electricity is several times higher during the summer on-peak than it is during the off-peak periods. Not only does the cost of electricity vary, but most utility rates also have a monthly demand charge based on the maximum rate of electricity use for the billing period. Since chilled water plants operate more intensely during the summer peak period, efficiency gains and peak reductions can result in very large utility bill savings.

In addition to new construction, the chilled water plants of many existing buildings are being replaced or overhauled. Older chilled water plants have equipment that uses ozone-damaging refrigerants. International treaties, in particular the Montreal Protocol, call for ozone damaging chemicals (in particular CFCs) to be phased out of production. As the availability of CFCs is reduced, the price will skyrocket, creating pressure for chilled water plants to be overhauled or replaced.

Target Audience

The CoolTools Design Guide is written for mechanical engineers who design, redesign or retrofit chilled water plants. The guide provides engineering information on how to estimate plant loads; details on chillers, towers and other plant equipment; system piping arrangements and configurations; controls; design approaches; contract documents; and commissioning. While design engineers are the primary audience, the guide also provides useful information for operation and maintenance personnel, mechanical contractors, and building managers.

Organization of Material

The design guide is organized in eight chapters. This first chapter is the overview.

2) ***Loads***. Chapter Two discusses the nature of chilled water loads and how they should be considered in the design of chilled water plants. In the past, most engineers have only estimated the peak or maximum load; however, accounting for the time pattern of loads can be just as important. It is also important to anticipate future growth in load and to build in excess capacity when appropriate. Methods of calculating peak loads and hourly loads are reviewed. These include site measurements (for existing facilities), computer simulations, rules of thumb and prototype buildings.

3) ***Equipment***. Chapter Three reviews some basics on chillers, cooling towers, pumps, and other plant equipment. The chapter discusses the basic refrigeration cycle; water chillers; cooling towers; air-cooled condensers; and pumps.

4) ***Systems***. Chapter Four discusses different ways of arranging chilled water equipment in the system to achieve energy efficiency and operational simplicity. The pros and cons of constant flow and variable flow systems are discussed along with different primary only and primary/secondary pumping systems. Other topics include interconnecting multiple chilled water plants and heat recovery chillers.

5) ***Controls***. Chapter Five explores the many design and performance issues related to controls and instrumentation of chilled water plants. Topics include: types of sensors; styles of and selection criteria for control valves; controller requirements and interfacing issues; the importance of performance monitoring; types and configuration of local instrumentation; and recommended control sequences for chilled water plants.

6) ***Design***. Chapter Six provides procedures and analysis techniques for optimizing chilled water plant design. Topics include optimizing the size and selection of the chillers and other plant components and the sequencing of the chiller plant equipment. A design approach is recommended which combines detailed analysis and rule-of-thumb recommendations. This approach will achieve better results than traditional design procedures.

7) ***Procurement***. Chapter Seven discusses strategies for procuring chilled water plant design and construction services. It also recommends specific procedures for evaluating chiller options and selecting an energy-efficient and cost-effective chiller. Case studies of the chiller selection process are provided for a new building and for a retrofit project. A sample chiller bid specification and a sample chiller bid form are also provided.

8) ***Commissioning***. Chapter Eight provides an overview of the commissioning process. The overview includes: a definition of commissioning; why commissioning is important; benefits and costs; levels of commissioning; and who should act as the commissioning authority. This overview is followed by a discussion of the commissioning phases for a typical chilled water plant, from the development of the commissioning program to post-occupancy commissioning activities. Examples of a commissioning plan, a commissioning specification and test procedures are also included.

Mechanical engineers who design chilled water plants are the target audience for the guide. All of the material in the guide is relevant to this group, although experienced engineers can briefly review Chapter 2 on loads and Chapter 3 on equipment and then refer to this material as necessary.

Building owners and managers should read the chapters on controls (5), procurement (7) and commissioning (8). The other chapters can be skipped or reviewed skimmed.

Plant operation and maintenance personnel should study the chapters on controls (5) and commissioning (8) and read the chapter on equipment (3).

Mechanical contractors and equipment suppliers should read the chapters on equipment (3), systems (4), controls (5), procurement (7) and commissioning (8). The chapters on loads (2) and design (6) are of less significance and may be skipped.

TABLE 1-1:
TARGET AUDIENCE

	TARGET AUDIENCE		**OTHER USERS OF GUIDE**		
Chapter	**Experienced Engineers**	**Engineers in Training**	**Building Owners and Managers**	**Operation/ Maintenance Personnel**	**Mechanical Contractors/ Suppliers**
Overview					
Loads					
Equipment					
Systems					
Controls					
Design					
Procurement					
Commissioning					

Legend

- Should review carefully
- Review as necessary
- Not Important

2. CHILLED WATER PLANT LOADS

This chapter discusses chilled water plant peak loads and annual cooling load profiles and how they affect plant design and equipment capacity. One of the primary thrusts of this Design Guide is to encourage the use of life-cycle cost analysis as the basis for optimizing the plant's design and making prudent equipment selections. Fundamental to the design process is a keen understanding of the chiller plant cooling loads and how they vary with time. If an existing plant is being modified or expanded, it is possible to monitor the cooling load and obtain an accurate estimate of both the peak load and the cooling load profile. A great many plants, however, are designed with only preliminary information available about the building's design and function. Getting accurate peak load and cooling load profile information for these plants is much more difficult. This chapter discusses the uncertainties involved with predicting chiller plant loads and the impact of these uncertainties on the design process.

Understanding Loads and Their Impact on Design

To provide for an optimum chilled water plant design, you need to have both a design (peak) load and a cooling load profile that describes how the load varies over time. The design load defines the overall installed plant capacity including the chillers, pumps, piping and towers. The cooling load profile is required to design the plant to stage efficiently. This includes design decisions like the unloading mechanisms of the chillers; the application of VFD on the chillers, towers and pumps; the application of hot-gas bypass; and the relative sizes of each piece of equipment.

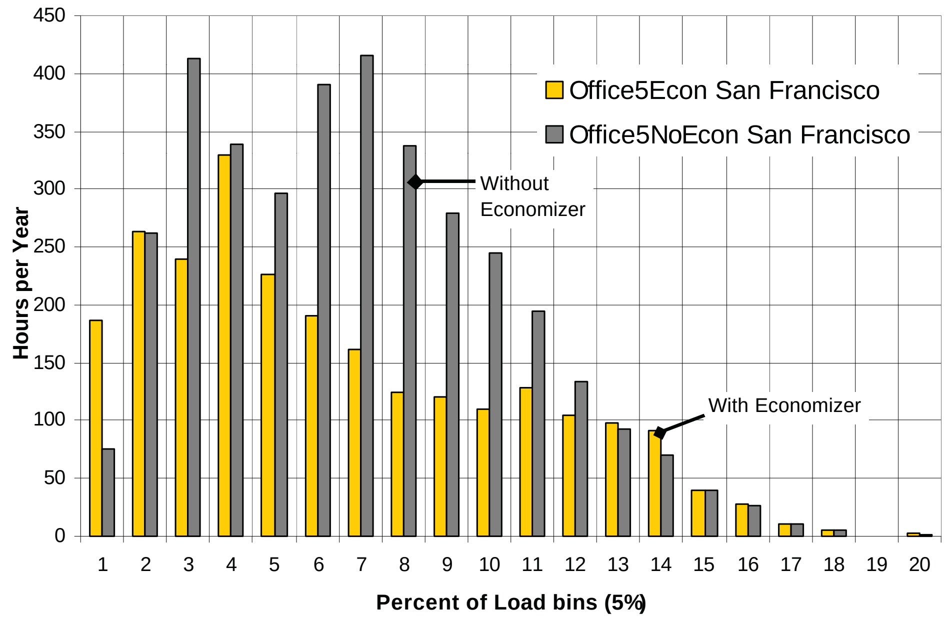
Certain key load parameters affect the cooling load profile and consequently the nature of the plant design. These parameters include:

The use of outdoor air economizers and 100% outdoor air units.

- The climate that the plant is located in.
- Hours of building or facility operation.
- Base (24/7) loads like telecom closets and computer rooms.

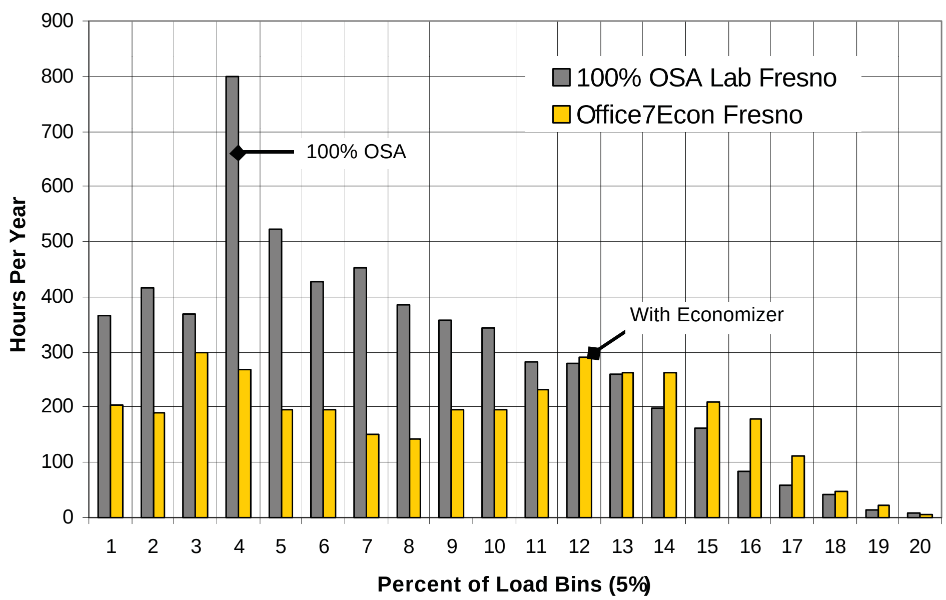
For example, the cooling load profile of a San Francisco office building that operates five days per week was analyzed with and without economizers. As Figure 2-1 shows, the number of hours that the plant operates increases dramatically when an economizer is not used. Additionally the shape of the profile changes dramatically. The profile will influence the optimum selection of the number and capacity of the chillers as well as the full-load and part-load energy efficiency of the machines.

Figure 2-1:
Cooling Load Profiles, 5-day Office in San Francisco



In a similar example, Figure 2-2 shows the cooling load profile of a laboratory in Fresno that utilizes 100% outdoor air and operated 24 hours per day, seven days per week. This load is compared to the cooling load profile of a similar size office building that operates seven days per week and has an economizer. This example dramatically indicates that the 100% outdoor air system has significantly more annual cooling load hours at reduced load than the office. The effects of the load profile will impact the optimum selection of the chiller plant configuration.

Figure 2-2:
Cooling Load Profiles, Office vs. 100% OSA Laboratory in Fresno, CA



In addition to the load profiles, plant design can be strongly influenced by design constraints that restrict the operating temperature ranges of the chilled or condenser water systems. Examples include space humidity control, incorporation of thermal energy storage, condenser heat recovery, combined heat and power, the use of absorption chillers, and a condenser water system that serves water-cooled AC units without head pressure control.

Peak Loads Overview

The process for estimating peak cooling loads in new construction is explained thoroughly in the *Fundamentals Volume of the American Society of Heating, Refrigeration, and Air-Conditioning Engineers (ASHRAE) Handbook* (Chapter 30 in the 2005 version). The basic variables for peak load calculations include weather conditions, building envelope design, internal heat gain, ventilation, and to a lesser extent infiltration. Less obvious but nonetheless important are the diversity between the various load components, and the effects of thermal mass. The diversity of loads is the probability of simultaneous occurrence of dynamic peak loads. In other words, diversity accounts for the fact that each of the envelope, occupancy, lighting, and plug loads will not peak at the same time in all spaces simultaneously.

There is inherent uncertainty in peak load calculations. Any number of elements can make the actual load differ from the calculated load. For instance:

- Local design conditions can differ from the weather station used for calculations.
- Weather conditions can vary over a period of time as a result of increasing urbanization, climate change, and changes in land use.
- Building envelope elements are not always what were planned for, due to change orders in the construction process or modifications after occupancy.
- Changes may occur in the operation and maintenance of the plant.
- Loss of equipment capacity due to degradation of system components (such as chiller heat exchanger fouling).
- Changes in operation (such as ventilation rates).
- Internal loads (lighting, plug loads and people) can be significantly different than were planned for and can vary over time.

Often the characteristics of the loads served are not clear at the time of the plant design. This is often the case with district or campus systems where you are building capacity and infrastructure to support future growth. Simulation tools (discussed later) and budgets based on measured existing buildings' usage can be quite helpful. In some campuses these budgets are assigned to the future buildings and it is up to the future design teams to make their buildings efficient enough to meet the budgets.

A plant expansion or remodel provides the opportunity to monitor the existing plant for peak and operating loads. Most energy management and control systems (EMCS) have the capability of supporting trend logs. Of course, the plant must also be provided with instrumentation (such as flow meters and temperature sensors) to provide useful load information. Also, a good operator can often accurately report on the percent of full load that the plant sees during peak weather conditions.

For most designers the perceived risks of understating the peak load condition (and undersizing the cooling plant) are much greater than overstating the peak load. An undersized cooling plant may not meet the owner's expectations for comfort and may affect the owner's ability to manufacture products or provide essential services. Oversizing the cooling plant, however, carries an incremental first-cost penalty that is not always easy to identify. Oversizing can have a positive or negative energy impact depending on the piece of equipment and how it is controlled. Oversized cooling towers and pipes tend to reduce the energy costs of operating the plant. Oversized pumps and chillers often run inefficiently at low loads although the use of variable speed drives mitigates this to some extent. Since oversizing always carries a first cost premium it is prudent to not oversize plants. Where future growth is uncertain, provisions for addition of future pumps, towers and chillers can be provided.

Annual Load Profiles Overview

A cooling load profile is a time series of cooling plant loads and correlated weather data. The primary role of a cooling load profile is to facilitate the correct relative evaluation of competing design options.

An accurate understanding of the cooling load profile affects the plant configuration. For example, a plant that serves a hotel complex with long periods of very low loads would be designed differently than a plant that serves widely varying loads only in mild and warm weather during the daytime (for instance, an office building).

If the actual cooling loads are closely related to weather data, then *temperature bin-estimating techniques* may produce satisfactory analysis results. But **CoolTools**™ research indicates that in most cases the load is not strongly correlated with weather data. Using bin-weather data alone for optimization calculations will seldom provide the accuracy needed for a truly optimized plant. Therefore, to accurately address the impact of the expected load profile in a chiller plant's design, it is necessary to have hourly load data for an entire "typical" year.

The designer's ability to accurately project hourly load profiles into the future adds a level of uncertainty to the entire analysis procedure. Other significant sources of uncertainty include:

- Changes in building occupancy over time
- Changes in ventilation rates (for instance, deployment of demand controlled ventilation devices)
- Problems with the controls that cause false loads (like simultaneous heating and cooling)

Designers should acknowledge uncertainty in the development of the annual load profiles. One approach is to consider a range of load profiles when designing a plant representing a reasonable range of changes in operating conditions.

Oversizing / Undersizing Considerations

Because of the uncertainty inherent in design parameters and the risks associated with undersizing the plant, most chilled water plants are larger than needed to meet maximum load conditions. Note that hourly load profiles, when properly prepared, include diversity. Diversified plant loads (as opposed to total connected load) should be used for the design of chilled water plants. While this may appear unwise from an energy conservation view, the issue is complex and underscores the need to understand part-load operations. Here are some impacts of oversizing/undersizing the chilled water plant:

- When operating at part loads, an oversized fixed speed chiller may not perform as efficiently as a smaller machine. Conversely, a variable speed chiller at part load may operate more efficiently than a smaller machine at full load.
- Oversized chillers have larger chilled and condenser water pumps that will consume more energy if the pumps are constant speed. This penalty can be significantly reduced if the pumps have variable speed drives or if the chilled water plant consists of multiple smaller chillers.
- The larger piping in the oversized plant will have less pressure drop (lower pumping energy) than that of a plant whose piping was "right sized."
- An oversized plant's cooling towers may save energy by allowing the fans to run slower (with VFDs). Also, they may produce lower condenser water temperatures for more efficient part-load operation of the chillers. Conversely, oversized cooling towers may have flow turndown problems that force the operators to use fewer cells at higher fan speeds which can increase plant energy use.
- Oversized plants always cost more to build. While a plant's cost may not vary linearly with its total capacity, larger plants have more expensive chillers, larger pumps and possibly larger piping.

Sometimes providing additional capacity is unavoidable. The owner's criteria may call for incorporating redundant chillers or for increasing plant capacity in anticipation of a future load. Redundant or spare equipment is a separate issue from oversizing.

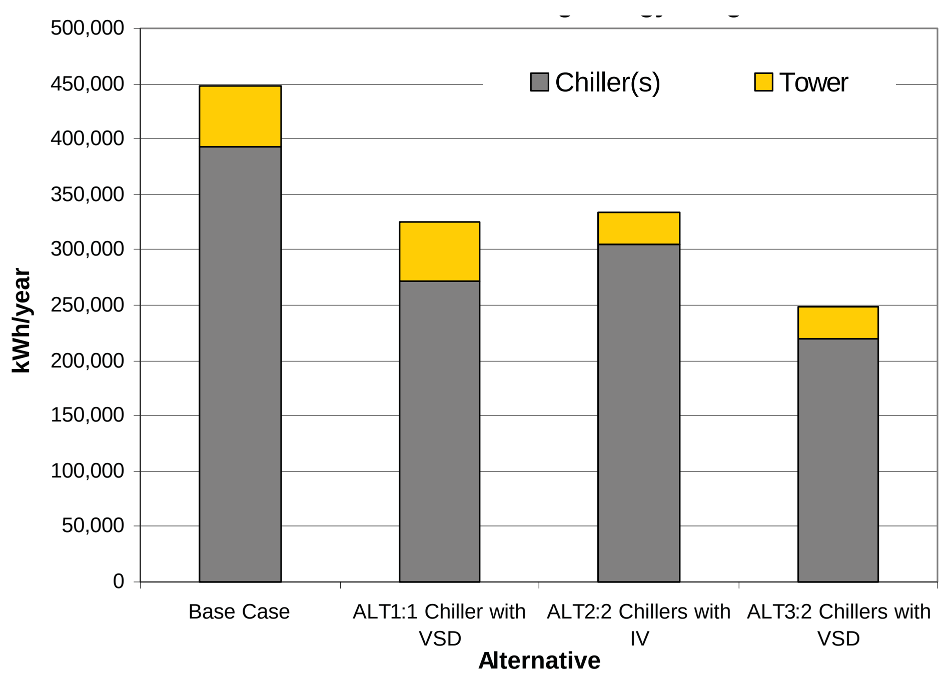
To mitigate problems with oversized plants, the chilled water plant must run efficiently at low loads. Chapter 6 discusses strategies for achieving optimum selection of chiller configurations. The following example from a computer simulation model helps demonstrate the issue of oversizing. In this case, an 800-ton cooling plant serves an office complex that operates on a basic five days per week schedule. Typical load profiles were scaled for peak cooling load of exactly 450 tons. The plant was modeled with the following scenarios:

- A single 800-ton machine with inlet vane control
- A single 800-ton machine with variable-speed drive control
- Two 400-ton machines with inlet vane control
- Two 400-ton machines each with variable-speed drive control

Figure 2-3 shows the results of this simulation. Note the dramatic reduction in annual cooling energy consumption when the variable-speed drive is added to the 800-ton machine, and also when multiple machines are added.

Although other scenarios may produce similar or better results, this example illustrates that the energy penalty for an oversized plant can be dramatically reduced if efficient turndown is incorporated into the design. By either adding a variable speed drive on a single chiller or providing two smaller fixed speed chillers the annual energy is reduced by approximately one third. Combining these measures (two chillers with variable speed drives) reduces the annual energy by nearly one half.

Figure 2-3:
Cooling Energy Usage for Four Design Alternatives



Determining Peak Loads

Calculations/Simulations

Fundamentals Volume of the ASHRAE Handbook defines accepted methods and procedures for cooling load calculations. These well-known procedures include information on ventilation and infiltration, climatic design information, residential and non-residential load calculations, fenestration, and energy estimating methods. In discussing cooling load principles, the *Handbook* emphasized the importance of analyzing each variable that may affect cooling load calculations:

"The variables affecting cooling load calculations are numerous, often difficult to define precisely, and always intricately interrelated. Many cooling load components vary in magnitude over a wide range during a 24-h period. Since these cyclic changes in load components are often not in phase with each other, each must be analyzed to establish the resultant maximum cooling load for a building or zone."

Starting in the 2000 edition, the *Handbook* supports only two methods of load calculation: the heat balance method (a fundamental first principals approach) and the radiant time series (RTS) method (an approximation of the heat balance method). For all practical purposes both of these methods require computer simulations to analyze.

A number of computer programs have been developed to implement one or the other of these methods including:

- Trane's Trace 700: http://www.trane.com/commercial/software/index2.asp?flash=no
- Carrier's HAP: http://www.commercial.carrier.com/commercial/hvac/general/0,,CLI1_DIV12_ETI3906_MID1738,00.html

- York's YorkCalc: http://www.york.com/products/esg/products/YorkEngineeredProducts.asp?cnt_Model_ID=138&Display=54&View=ON&ShowSubsID=138&Model=138
- WrightSoft: http://www.wrightsoft.com/default.asp?country=US
- EnergyPlus: http://www.eere.energy.gov/buildings/energyplus/
- EnergyPro: http://www.energysoft.com/
- Elite's CHVAC: http://www.elitesoft.com/web/hvacr/chvacx.html

A more comprehensive list of load calculation tools is maintained by the Department of Energy on their website at http://www.eere.energy.gov/buildings/tools_directory/subjects.cfm/pagename=subjects/pagename_menu=whole_building_analysis/pagename_submenu=load_calculation.

Although these calculation techniques have worked very well over the years, designers must be aware of the limitations of these techniques and recognize that the methods do not all predict the same loads. Because of the uncertainties previously discussed, the design load calculations may be different than the actual chiller plant peak load. Selecting the maximum capacity of the plant is important, but it is perhaps even more important to consider the plant's energy efficient part-load performance.

Site Measurements

When an existing chiller plant is being remodeled or expanded, it is possible to monitor the actual peak cooling load to obtain invaluable information. The monitoring can be short term (several months) to establish peak load and daily trends or can be long term (one year or longer) to determine annual load profiles. Successfully measuring energy and load performance of a cooling plant requires rigorous monitoring protocols. These monitoring protocols comprise four stages, which include:

1. Survey of Monitoring Sites (Stage One): Conduct a complete audit of the chilled water plant, including a comprehensive systems diagram and lists of all equipment, energy performance characteristics, motor sizes and control strategies.
2. Monitoring Plan (Stage Two): From the comprehensive systems diagrams prepare a plan for determining the data to be monitored and the monitoring equipment needed. Necessary monitoring equipment includes data loggers, flow measurement devices, temperature measurement devices, power measurement devices, and instrumentation tables. Determine the time periods to be monitored.
3. Field Installation (Stage Three): Install instrumentation in accordance with the monitoring plan and the installation instructions. Take spot measurements to assure that the equipment is calibrated properly and that all sensors and instruments are working correctly. Provide guidelines to operators. Have a plan for removal of instrumentation and patching of insulation, etc.
4. Data Collection and Analysis (Stage Four): Obtain data and provide validation. Perform analysis on both a basic level (for example, simple temperature logs of chiller energy usage), and a more detailed level (for example, chiller plant energy performance as a function of various elements such as time and weather).

From this procedure, the peak loads will emerge, as well as the relationship and interaction of the various components. The quality of the monitoring protocol will determine the accuracy and usefulness of the results.

Rules of Thumb

There are a number of helpful rules of thumb for sizing chilled water plants. Rules of thumb are useful for providing check figures on load calculations. Also, if the buildings connected to the chilled water plant are not fully defined, rules of thumb may be the only way to effectively determine the plant's cooling load. Such is the case in many central plants that serve campus distribution systems where future—but presently undefined—buildings may be a prominent portion of the ultimate load. Rules of thumb are best used in the early analysis stages of plant design. It is not appropriate to use rules of thumb for the final equipment selection.

ASHRAE has published load check figures in its *Pocket Guide for Air-Conditioning, Heating, Ventilation and Refrigeration*. When using this guide for the California climate, it is best to use low and average columns. This is because this table represents the full spectrum of climate conditions across the United States and was developed prior to energy codes being in place.

Determining Hourly Load Profiles

There are several methods for determining annual cooling load profiles depending on what stage the project is in and the resources available for analysis. The following are common methods for determining annual cooling load profiles:

- Computer simulation models (customized and prototypical)
- Site measurements

For new construction, the two methods typically used are custom computer simulations and prototypical simulations. Customized simulations have the greatest potential for accuracy but can be costly to develop and are subject to modeler error. Prototypical simulations offer quick and relatively inexpensive analysis but may not be as accurate as the customized simulation.

For retrofit and expansion of existing plants, it is possible to conduct site visits to measure profiles. This technique yields the most accurate results but requires special planning, technical expertise, equipment, budget and time.

Each of these methods can be combined with statistical and mathematical techniques from a variety of sources including short-term measurements, site data, and billing data. These hybrid approaches offer the best possibility to balance accuracy and effort. The following sections discuss each technique.

Computer Simulation Models

As discussed above, cooling load profiles generated by computer simulation models can be customized for the specific project. These custom models can take between a few hours to several person-weeks of time depending on the complexity of the building geometry and the effort spent on making the model accurate. With recent advances in simulation tool data exchange (for example, GBXML - http://www.gbxml.org/, the IAI - http://www.iai-na.org/

and ASHRAE's *Guideline 20, "XML definitions for HVAC&R"* - http://gpc20.ashraepcs.org/) the effort to build these models has significantly decreased—building geometry from CAD programs can be simply imported into load or simulation tools. Examples include EnergyPlus (that supports both IAI IFCs and GBXML), GBXML's DOE2 and Trane's Trace (that supports GBXML). Third-party services like Green Building Studio (http://www.greenbuildingstudio.com/About.aspx) can translate CAD files into building geometry for multiple simulation and load calculation tools. Although these tools are far from "plug and play," they are used by many engineering firms as a routine part of their work.

For projects that are early in design and evaluations for campus systems, prototypical models are a useful tool. Many of the simulation tools now incorporate wizards that enable designers to develop a "typical" building for analysis in a matter of minutes. Examples include VisualDOE and eQuest.

Simulation tool resources include:

- VisualDOE: http://www.archenergy.com/products/visualdoe/
- EnergyPlus: http://www.eere.energy.gov/buildings/energyplus/
- eQuest: http://doe2.com/
- EnergyPro: http://www.energysoft.com/
- Trane's Trace 700: http://www.trane.com/commercial/software/index2.asp?flash=no
- Carrier's HAP: http://www.commercial.carrier.com/commercial/hvac/general/0,,CLI1 DIV12_ETI3906_MID1738,00.html
- York's YorkCalc: http://www.york.com/products/esg/products/YorkEngineered Products.asp?cnt_Model_ID=138&Display=54&View=ON&ShowSubsID=138&Model=138

A more comprehensive list of simulation tools is maintained by the Department of Energy on their website at http://www.eere.energy.gov/buildings/tools_directory/subjects.cfm/pagename=subjects/pagename_menu=whole_building_analysis/pagename_submenu=energy_simulation.

A simulation tools comparison guide can be found at: http://www.eere.energy.gov/buildings/tools_directory/pdfs/contrasting_the_capabilities_of_building_energy_performance_simulation_programs_v1.0.pdf.

Computer simulation models require experienced modelers for inputting data and checking results To assess the impact of uncertainties, the modeler should consider a range of input variations representing: the best estimate, possible but likely low loads, and possible but likely high loads.

Site Measurements

Site monitoring for peak loads was discussed earlier in this chapter. The same site monitoring protocol can be used for determining cooling load profiles based on either short-term or long-term measurements. Long-term monitoring is not likely to become standard practice because it is costly and time-consuming to obtain the data. Also, experience with long-term data indicates that due to weather and other variables, a single year's measurement wouldn't match the second year's data and as a result is not deterministically exact. Short-term data could potentially be used to define the basic shape of the typical 24-hour load profile by season or month. However, the data is climate sensitive and the associated weather/load profiles are difficult to record, especially considering the solar aspect of the load. In the future, perhaps methods will be available that use short-term data to normalize simulation-based or prototypical-based load profiles.

3. CHILLED WATER PLANT EQUIPMENT

Introduction

Chilled water plants are complex systems. Design engineers seeking to maximize the performance and economic benefits of upgraded or new chilled water plants need a thorough understanding of the major equipment used in these plants. This chapter provides an in-depth look at the equipment, as well as essential information on how the components relate to one another, how they are controlled, and what their physical and operational limitations are. This chapter discusses:

- the basic refrigeration cycle;
- the components commonly used in commercially available packaged water chillers;
- methods of heat rejection, with an emphasis on cooling towers and air-cooled refrigerant condensers;
- the characteristics of different types of pumps;
- pump and system curves, with an emphasis on understanding the nature of variable-speed pumping; and
- the application and efficiency of variable-speed drives.

Water Chillers

Manufacturer Data

Chiller manufacturers have been hard at worked to developing new products and product refinements in the past few recent years. This section of the guide presents an overview of the current technologies. The reader is encouraged to browse the manufacturer's websites for the most current information on technologies and refrigerants. The major manufacturer websites are:

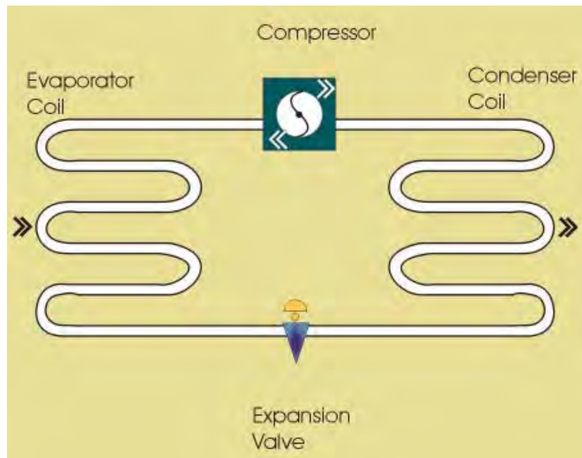
- Carrier Corporation: http://www.commercial.carrier.com/commercial/hvac/general/0,,CLI1_DIV12_ETI1508_MID4369,00.html
- Copeland Corporation (scroll compressors): http://www.copeland-corp.com/
- Danfoss (Turborcor compressor): http://www.turbocor.com/
- McQuay International: http://www.mcquay.com/McQuay/ProductInformation/AllProducts/AllProducts
- Trane: http://www.trane.com/Commercial/Equipment/Refrigeration.aspx
- York International: http://www.york.com/products/esg/products/YorkEngineeredProducts.asp?Display=24&View=ON

Refrigeration Cycle

The refrigeration cycle is the fundamental thermodynamic basis for removing heat from buildings and rejecting it to the outdoors. The refrigeration cycle requires four basic components:

- Compressor
- Evaporator
- Condenser
- Expansion valve

FIGURE 3-1:
THE REFRIGERATION CYCLE



The refrigeration cycle diagram shows the relationship of these components, as does the pressure-enthalpy chart, also known as a P-H diagram. These diagrams cover the liquid-vapor regions specific to the cycle refrigerant. The following is a description of the refrigeration cycle:

- Starting at point A, the refrigerant is a liquid at high pressure. As it passes through the expansion valve to point B the pressure drops. At point B the refrigerant is a mixture of liquid and gas. At this point the gas is called "flash gas." At point A' the liquid refrigerant upstream of the throttling device has been cooled to a temperature below saturation. This effect is called "subcooling" and has the effect of reducing the amount of flash gas, as shown by point B'.

- From point B to point D the liquid is converted to a gas by absorbing heat (refrigeration effect). Notice the gas leaving the evaporator at point D has been heated to a level greater than saturation as shown by point C. The heat from point C to D is called "superheat." Superheating in the evaporator ensures that there is no liquid in the refrigerant as it moves into the compressor.

- From point D the refrigerant is drawn into the suction of the compressor where the gas is compressed, as shown by point E. At point E the temperature and pressure of the gas have been increased. The refrigerant is now called "hot gas." Notice that this point is to the right of the saturation curve, which also represents a superheated state.

- The hot gas, point E, moves into the condenser where the condensing medium (either air or water) absorbs heat and changes the refrigerant from a gas back to a liquid as shown by point A. At point A the liquid is at an elevated temperature and pressure. The liquid is forced through the liquid line to the throttling device and the cycle is repeated.

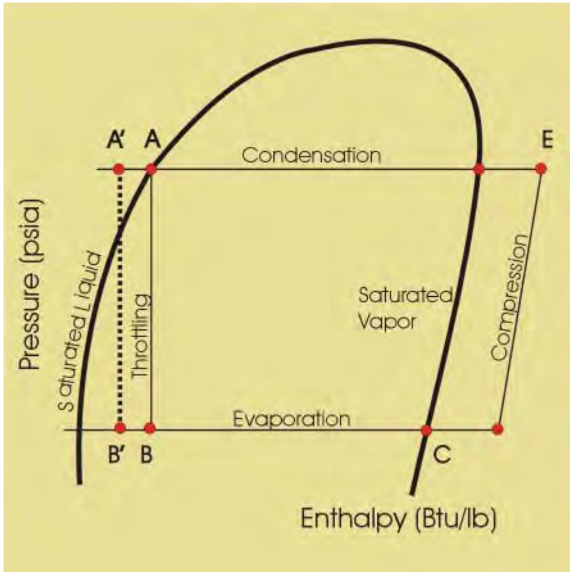


Figure 3-2:
Pressure-Enthalpy Chart

Refrigerants

To address safety and environmental concerns, refrigerants must have low toxicity, low flammability and a long atmospheric life. Recently, refrigerants have come under increased scrutiny by scientific, environmental, and regulatory communities because of the environmental impacts attributed to their use. Some refrigerants—particularly chlorofluorocarbons (CFCs)—are known to destroy stratospheric ozone. The relative ability of a refrigerant to destroy stratospheric ozone is called its ozone depletion potential (ODP).

CFCs have been phased out according to the 1987 Montreal Protocol. The production of CFCs in developed countries ceased in 1995, and their most common replacement, halogenated chlorofluorocarbons (HCFCs) are due for phase-out in the 21st century. Replacements are currently being developed for HCFC R-123 and HCFC R-22, which are commonly used in the industry. For all practical purposes, however, HCFCs should be available well into the middle of the 21st century and certainly within the lifetimes of machines currently being manufactured.

Table 3-1:
Montreal Protocol

The 1987 Montreal Protocol, and subsequent revisions, established the following timeline for the phase-out of chlorinated fluorocarbons (CFC) and hydrochlorinated fluorocarbons (HCFC):

Refrigerant	Year	Restrictions
CFC-11	1996	Ban on Production
CFC-12	1996	Ban on Production
HCFC-22	2010	Production Freeze and ban on use in new equipment
	2020	Ban on Production
HCFC-123	2015	Production Freeze
	2020	Ban on use in new equipment
	2030	Ban on Production
HFC-134a	-	No restrictions

The global warming potential (GWP) of refrigerants is another significant environmental issue. Gases that absorb infrared energy enhance the "greenhouse effect" in the atmosphere, leading to the warming of the earth. Refrigerants have been identified as "greenhouse gases." A chart showing the ODP versus GWP of various refrigerants is shown in Figure 3-3 below. Theoretically, the best refrigerants would have zero ODP and zero GWP, like R-717 (ammonia). Although some refrigerants used in a particular system may have a direct effect on global warming, there will also be an *indirect* effect on global warming as a result of that system's energy consumption for all refrigerants. The indirect effect is caused by the burning of fossil fuels and subsequent release of carbon dioxide. To reduce greenhouse gases to the greatest extent possible, it is critical to focus on the system's overall energy efficiency, not just to consider the refrigerant's GWP.

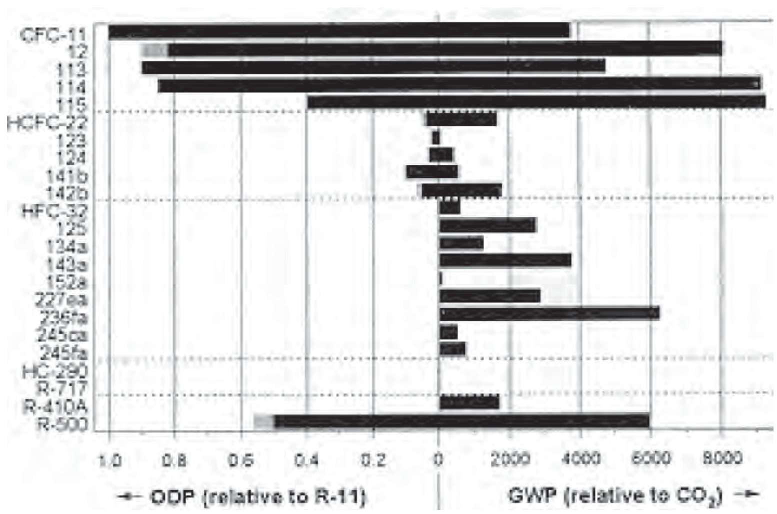


Figure 3-3: ODP vs. GWP for Common Refrigerants from http://www.trane.com/commercial/library/graphics/trade-off02.gif

When comparing the theoretical and practical efficiencies of different refrigerants, only slight differences among various refrigerants becomes apparent, with R-123 being somewhat better than the refrigerants it is designed to replace. Calm et al. (1997)[1] reports "that efficiency is not an inherent property of the refrigerant, but rather achieving the highest efficiencies depends on optimization of the system and individual components for the refrigerant."

[1] Calm, J.M. and D.A. Didion. 1997. *Trade-Offs in Refrigerant Selections: Past, Present, and Future*. Atlanta: American Society of Heating, Refrigerating and Air-Conditioning Engineers, Inc. See also http://www.trane.com/commercial/issues/environmental/trade-offs05.asp.

Compressors

There are four basic types of compressors used in packaged water chillers. These are:

- Reciprocating
- Rotary
- Centrifugal

In addition to these three, there are absorption chillers which are a thermal compression process. Each of these four categories of equipment are described in greater detail in the paragraphs that follow.

Reciprocating

A reciprocating compressor may be a positive displacement machine that works very similarly to an automobile engine. A piston is driven through a pin and connecting rod from a crankshaft, which is driven by a motor. As the piston moves down, the resulting suction opens a valve and allows the refrigerant to be drawn into the cylinder. On the upward stroke the increased pressure closes the suction valve. When the cylinder pressure exceeds the pressure in the discharge line, the discharge valve opens and the hot gas is released to the discharge pipe.

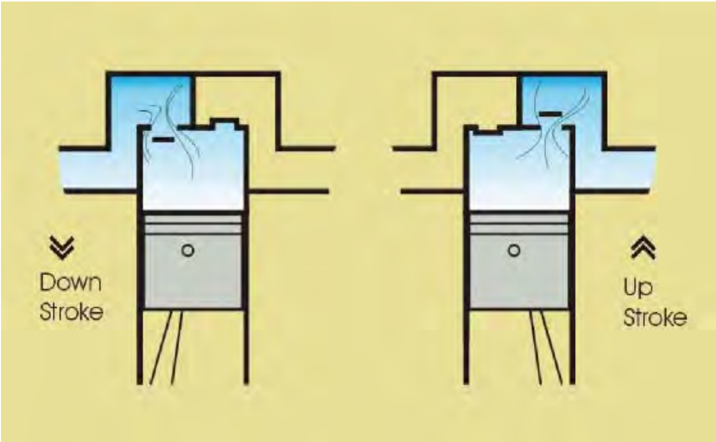
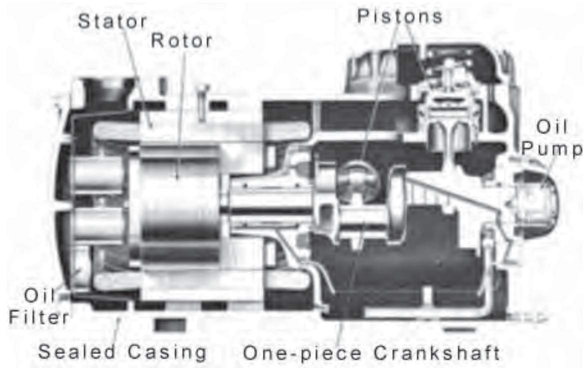


Figure 3-4:
Reciprocating Compressor

The reciprocating compressor is either open or hermetic. Open compressors are those in which the shaft extends through a seal in the crankcase for an external drive. Hermetic compressors are those in which the motor and compressor are contained within the same housing, thus refrigerant us is used to cool the motor. The motor shaft is integral with the compressor crankshaft and the motor is in contact with the refrigerant. Hermetic compressors can be totally sealed within a welded shell or can be semi-hermetic, which is a hermetic compressor that is bolted rather than welded to facilitate field repair. The semi-hermetic compressor is far more common for water chillers (except when ammonia is used). A semi-hermetic compressor can have as many as 16 cylinders but 4 to 6 are most common.

Figure 3-5:
Hermetic Compressors



Typically, the cooling capacity of reciprocating compressors are controlled by:

- cycling them on/off, with or without multiple compressors;
- using cylinder unloaders;
- using hot gas bypass (HGBP); or
- all three methods.

Cycling the compressor on/off is a cost-effective and energy-efficient control strategy, particularly when multiple compressors are used. Cycling the compressor too rapidly can cause motor failure. To prevent motor failure control, circuit relays will delay the restarting of the compressor and other relays will force the compressor to run for a minimum amount of time. These safety devices can cause very uneven chilled water temperature fluctuations.

Unloaders are devices that lift the suction gas valve so that the piston does not compress the gas. Since the hermetic compressor is cooled with the refrigerant, a minimum number of cylinders must always be loaded. Unloaders are a cost-effective and energy-efficient control strategy. Because the pistons continue to move in an unloaded condition, the part-load energy efficiency is somewhat less than the cycling method. Unloads can be cycled often without detriment to the motor.

Hot gas bypass works by diverting hot gas from the compressor discharge into the evaporator. This is usually controlled from low suction pressure. When used it should be set to operate only after the last stage of unloading has occurred. Hot gas bypass allows the machine to run at very low and no cooling loads without cycling the compressor. With hot gas bypass there is no energy savings as the machine unloads. As a result, hot gas bypass should only be used for critical low-load applications.

Larger chillers (up to 230 tons) using reciprocating compressors will have multiple compressors, usually with two separate refrigerant circuits. During light loads, one of the refrigeration circuits is deactivated.

Currently, scroll or screw compressors are largely supplanting reciprocating chillers.

Rotary

There are a number of types of rotary compressors used in the HVAC industry including scroll, single blade (fixed vane), rotating vane, and screw (helical-rotary). Single blade and rotating vane compressors are generally used in smaller applications and will not be discussed further here. Scroll compressors are largely replacing reciprocating compressors for the smaller chiller sizes (although there are scroll machines up to 400 tons in capacity). In packaged water chillers the most commonly used compressor is the screw. There are two types in use today: the single screw and the multiple screw.

- **Single Screw**. The single screw consists of a single cylindrical main rotor that works with a pair of gaterotors. The compressor is driven through the main rotor shaft and the gaterotors follow by direct meshing action. As the main rotor turns, the teeth of the gaterotor, the sides of the screw, and the casing trap refrigerant. As rotation continues, the groove volume decreases and compression occurs. Since there are two gaterotors, each side of the screw acts independently. Single-screw compressors are noted for long bearing life as the bearing loads are inherently balanced. Some single-screw compressors have a centrifugal economizer built into them. This economizer has an intermediate pressure chamber that takes the flash gas (via a centrifugal separator) from the liquid and injects it into a closed groove in the compression cycle, with the result of increased efficiency.

  The single screw is controlled from a slide valve in the compressor casing that changes the location where the refrigerant is introduced into the compression zone. This causes a reduction in groove volume and hence the volume of gas compressed varies (variable compressor displacement). These compressors are fully modulating. The single screw has slide valves on each side that can be operated independently. This allows the machine to have a very low turndown with good part-load energy performance.

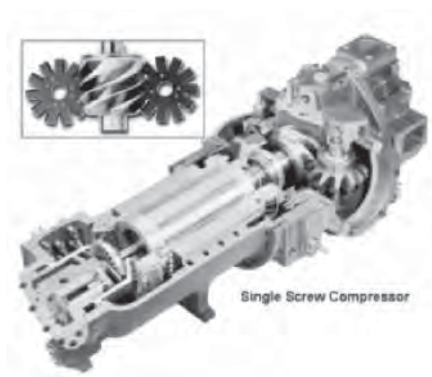


FIGURE 3-6:
THE SINGLE SCREW COMPRESSOR

- **Twin Screw**. The Twin Screw is the most common of the multiple screw compressors. The twin screw is the common designation for double helical rotary screw compressor. The twin screw consists of two mating helically grooved rotors, one male and the other female. Either the male or female rotor can be driven. The other rotor either follows the driven rotor on a light oil film or is driven with synchronized timing gears. At the suction side of the compressor, the gas fills a void between the male and female rotors. As the rotors turn, the male and female rotors mesh and work with the casing to trap the gas. Continued rotation decreases the space between lobes and the gas is compressed. The gas is discharged at the end of the rotors.

The twin screw has a slide valve for capacity control, located near the discharge side of the rotors, which bypasses a portion of the trapped gas back to the suction side of the compressor.

Figure 3-7:
Twin Screw

- **VSD Controls**. In recent years Carrier and York have introduced screw chillers with VSD controls. The Carrier 23XRV is a water-cooled screw chiller with variable speed driven compressor. York's variable speed screw machine (YCAV, Latitude) is air cooled. The York chillers have multiple compressors and are offered in standard and high efficiency modelsies with all compressors on VSDs and only one compressor on VSDS. In addition to the great part-load performance, these chillers offer significantly reduced noise and wear at off design conditions.

Centrifugal

Centrifugal compressors are dynamic compression devices (as opposed to positive displacement) that on a continuous basis exchange angular momentum between a rotating mechanical element and a steadily flowing fluid. Like centrifugal pumps, centrifugal chillers have an impeller that rotates at high speed. The molecules of refrigerant enter the rotating impeller in the axial direction and are discharged radially at a higher velocity. The dynamic pressure of the refrigerant obtained by the higher velocity is converted to static pressure through a diffusion process that occurs in the stationary discharge or diffuser portion of the compressor just outside the impeller.

A centrifugal compressor can be single stage (having only one impeller) or it can be multistage (having two or more impellers). On a multistage centrifugal compressor, the discharge gas from the first impeller is directed to the suction of the second impeller, and so on for each stage provided. Like the rotary compressor, multiple stage centrifugals can incorporate economizers, which take flash gas from the liquid line at intermediate pressures and feed this into the suction at various stages of compression. The result is a significant increase in energy efficiency.

Like reciprocating compressors, centrifugal compressors can be either open or hermetic. Open centrifugal compressors have the motors located outside the casings with the shaft penetrating the casing through a seal. Hermetic centrifugal compressors have the motor and compressor fully contained within the same housing, with the motor in direct contact with the refrigerant. Because the discharge pressure developed by the compressor is a function of the velocity of the tip of the impeller, for a given pressure, smaller diameter impellers result in faster impeller speeds. Similarly, for a given pressure, the more stages of compression there are, the smaller the impeller diameter needs to be. With these variables in mind, some manufacturers have chosen to use gear drives to increase the speed of a smaller impeller, while other manufacturers use direct drives with larger impellers and/or multiple stages. High speed directly coupled motor-impeller compressors are also available. There are pros and cons to each of these designs, but direct drive machines have fewer moving parts, fewer bearings, and are generally simpler machines.

One of the characteristics of the centrifugal compressor is that it can "surge." Surge is a condition that occurs when the compressor is required to produce high lift at low volumetric flow. Centrifugal compressors must be controlled to prevent surge and this is a limit on part-load performance. During a surge condition, the refrigerant alternately moves backward and forward through the compressor, creating a great deal of noise, vibration, and heat. Prolonged operation of the machine in surge condition can lead to failure. Surge is relatively easy to detect in that the electrical current to the compressor will alternately increase and decrease with the changing refrigerant flow. Just before entering surge, the compressor may exhibit a property called "incipient surge" in which the machine gurgles and churns. This is not harmful to the compressor but may create unwanted vibration. The electrical current does not vary during incipient surge.

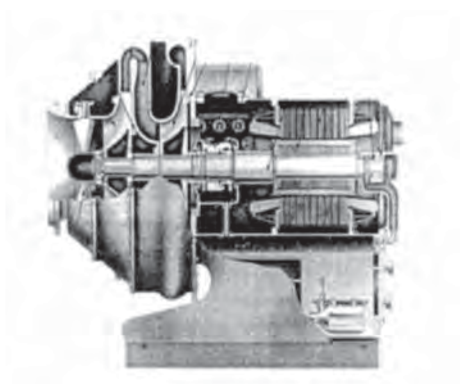


Figure 3-8:
Hermetic Centrifugal Compressor

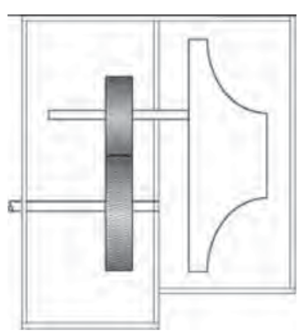


Figure 3-9:
Gear Drives

The capacity of centrifugal compressors may be controlled by three methods. The most common is to use inlet guide vanes or prerotation vanes. The adjustable vanes are located in the compressor's suction at the eye of the impeller and swirl the entering refrigerant in the direction of rotation. This changes the volumetric flow characteristics of the impeller, providing the basis for unloading.

Figure 3-10:
Inlet Guide Vanes

A second control method is to vary the speed of the impeller in conjunction with using inlet guide vanes. Not unlike a variable-speed fan or pump, reducing the impeller speed produces extremely good part-load energy characteristics. The impeller must produce an adequate pressure differential (lift) to move the refrigerant from the low-pressure side (evaporator) to the high-pressure side (condenser). This lift determines the minimum speed of the impeller. The lower the lift, that is, the closer the refrigerant temperature difference between the evaporator and condenser, the slower the impeller can rotate. When the impeller is at the slowest possible speed, further reductions in capacity are obtained by using the inlet guide vanes. With variable-speed drives and aggressive water temperature reset schedules, centrifugal compressors can produce the most energy efficient part-load performance of any compressor. Centrifugal compressors with variable speed drives use both the VSD and inlet vanes for control as the speed to provide the lift required may still result in excess refrigerant flow rate. The inlet vanes are used to reduce the refrigerant flow rate because reducing the speed attempting to provide the proper refrigerant flow rate alone can place a centrifugal compressor in surge. prevent the chiller from getting into surge. For efficient operation, the controls must either dynamically measure or model surge so that the bulk of the unloading can be done by the VSD. This is particularly important with primary-only variable flow plants as some manufacturer's' systems use load as an input to the surge map and they only measure temperature and not flow. There have been cases where VSDs minimums have to be set at 48 HZ or higher to prevent the chiller from tripping from surge at low flows.

The newly released (2003) Danfoss Turborcor compressor offers both variable speed control and magnetic bearings that compensate for any rotational imbalances. This chiller is largely oil free as the shaft floats in a carefully controlled magnetic field. Frictionless bearings improves the efficiency of the compressor, and reduces maintenance and noise. Removal of oil from the system improves heat transfer efficiency. The Turborcor compressor is featured in McQuay water-cooled chillers and is available as a retrofit kit for both screw and centrifugal machines.

As described for the screw chillers above, centrifugal chillers with VSDs have both lower noise and reduced wear at off design conditions.

A third method of capacity control for the centrifugal chiller is hot gas bypass (HGBP). Like other types of compressors, HGBP can be used to unload a machine to zero load by directing the hot gas from the compressor discharge back into the suction. There are no part-load energy savings with HGBP. It is used only as a last resort when very low turndown is required and cycling the machine on/off would not produce acceptable results.

Absorption

The absorption process is another way to compress refrigerants,; however, the process is thermal rather than mechanical. While appearing quite complex, absorption chillers use the same refrigeration process discussed for mechanical compression, except that compression is achieved with an absorber, generator, pump and recuperative heat exchanger. The following description is based on lithium bromide/water, which is the most common process among several possibilities. In the absorption refrigeration cycle, the low-pressure (high vacuum) refrigerant (water) in the evaporator migrates to the lower-pressure absorber where it is "soaked up" by a solution of lithium-bromide. While mixed with the lithium-bromide the vapor condenses and releases the heat of vaporization picked up in the evaporator. This heat is transferred to condenser water and rejected out the cooling tower. The lithium-bromide and refrigerant solution (weak solution) are pumped to a heat exchanger (generator) where the refrigerant is boiled off and the lithium-bromide (strong solution) returns to the evaporator. As the hot lithium-bromide (strong solution) returns to the evaporator, a heat exchanger cools the liquid with the cool mixture of lithium-bromide and refrigerant (weak solution). The boiled-off refrigerant migrates to the cooler condenser where it is condensed back into a liquid and returned to the evaporator to start the cycle again.

Absorption machines can be direct-fired or indirect-fired. The direct-fired absorber has an integral combustion heat source that is used in the primary generator. An indirect-fired absorber uses steam or hot water from a remote source.

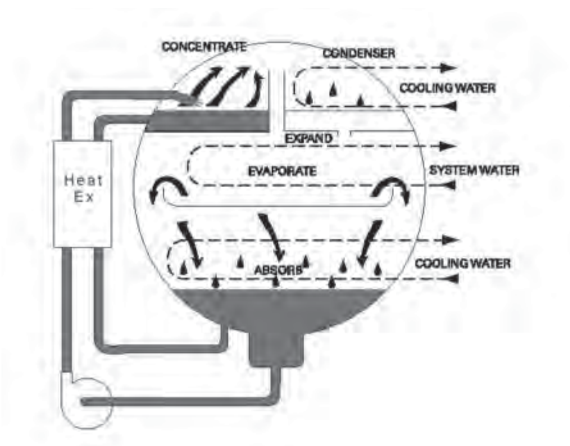
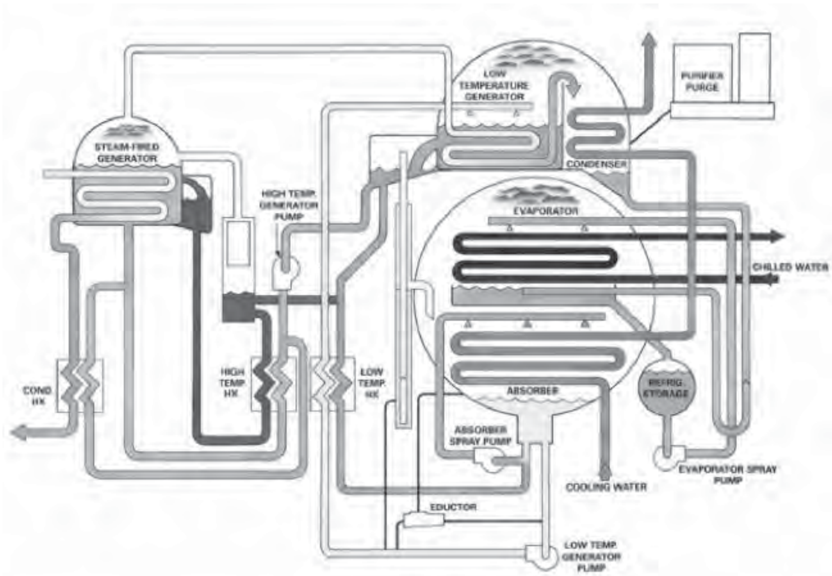


Figure 3-11:
Absorption Refrigeration Cycle

A double-effect absorption process is similar to that described above except that a generator, condenser, and heat exchanger are added. The refrigerant vapor from the primary generator runs through a heat exchanger (secondary generator) before entering the condenser. The secondary generator with the hot vapor on one side of the heat exchanger boils some of the lithium-bromide and refrigerant solution (weak solution), creating the double effect. The double-effect absorption process is significantly more energy efficient than the single-effect absorption process.

Figure 3-12:
Double-Effect Absorption



The lithium-bromide is a salt with a crystalline structure that is soluble in water. If the saturation point of the solution is exceeded, the salt will precipitate out and form a slush-like mixture that can plug pipes and make the machine inoperable. Crystallization does not harm the equipment but is a nuisance. Usually air leakage or improper (too cold) temperature settings cause crystallization. However, crystallization is generally not a problem in modern equipment that uses microprocessor-based controls. The microprocessor continuously monitors solution concentration and automatically purges the system. Absorption machines are controlled by modulating the firing rate of the direct-fired machine or modulating the flow of steam or hot water in the indirect-fired machine. Variable-speed refrigerant and solution pumps greatly enhance the controllability of the absorption machine.

Evaporators

Two types of evaporators are used in water chillers—the flooded shell and tube and the direct expansion evaporators (DX). Both types are shell and tube heat exchangers. Flooded shell and tube heat exchangers are typically used with large screw and centrifugal chillers, while DX evaporators are usually used with positive displacement chillers like the rotary and reciprocating machines. While water is the most common fluid cooled in the evaporator, other fluids are also used. These include a variety of antifreeze solutions, the most common of which are mixtures of ethylene glycol or propylene glycol and water. The use of antifreeze solutions significantly affects the performance of the evaporator but may be needed for low temperature applications. The fluid creates different heat transfer characteristics within the tubes and has different pressure drop characteristics. Machine performance is generally derated when using fluids other than water.

Flooded Shell and Tube

The flooded shell and tube heat exchanger has the cooled fluid (usually water) inside the tubes and the refrigerant on the shell side (outside the tubes). The liquid refrigerant is uniformly distributed along the bottom of the heat exchanger over the full length. The tubes are partially submerged in the liquid. Eliminators are used as a means to assure uniform distribution of vapor along the entire tube length and to prevent the violently boiling liquid refrigerant from entering the suction line. The eliminators are made from parallel plates bent into Z shape, wire mesh screens, or both plates and screens. An expansion valve maintains the level of the refrigerant. The tubes for the heat exchanger are usually both internally and externally enhanced (ribbed) to improve heat transfer effectiveness.

Manufacturers typically limit water flow on the high end to prevent erosion of the piping and on the low end (typically around 3 feet per second) to maintain heat transfer. It is best to check with the manufacturers for their specific flow rate limitations on each chiller. Typically, water pressure drops for evaporators in the HVAC industry do not exceed 25 to 30 feet of water column (11 to 13 psi). Flooded shell and tube heat exchangers are available with numerous passes. The greater the number of passes, the lower are the minimum flow requirements. One convenient accessory with the shell and tube heat exchanger is the addition of marine water boxes. These allow the mechanical cleaning of the tubes without disassembling the connecting piping.

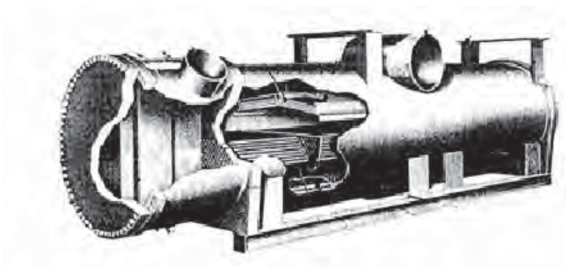


Figure 3-13: Flooded Shell and Tube Heat Exchanger

Direct Expansion

The direct expansion (DX) evaporator has the refrigerant inside the tubes and the cooled fluid (usually water) on the shell side (outside the tubes). Larger DX evaporators have two separate refrigeration circuits that help return oil to the positive displacement compressors during part-load. DX coolers have internally enhanced (ribbed) tubes to improve heat transfer effectiveness. The tubes are supported on a series of polypropylene internal baffles, which are used to direct the water flow in an up-and-down motion from one end of the tubes to the other. Water velocities do not exceed approximately 1½ to 2½ feet per second due to pressure drop considerations.

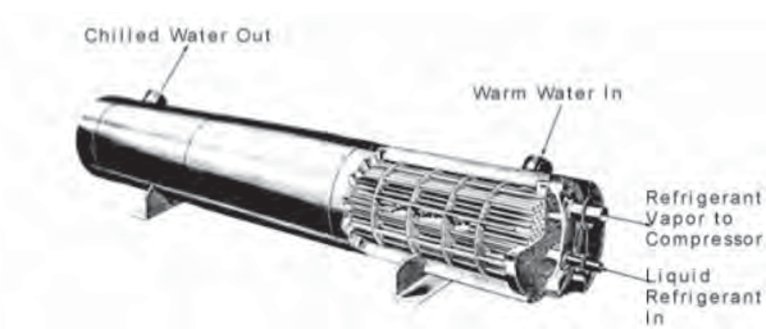
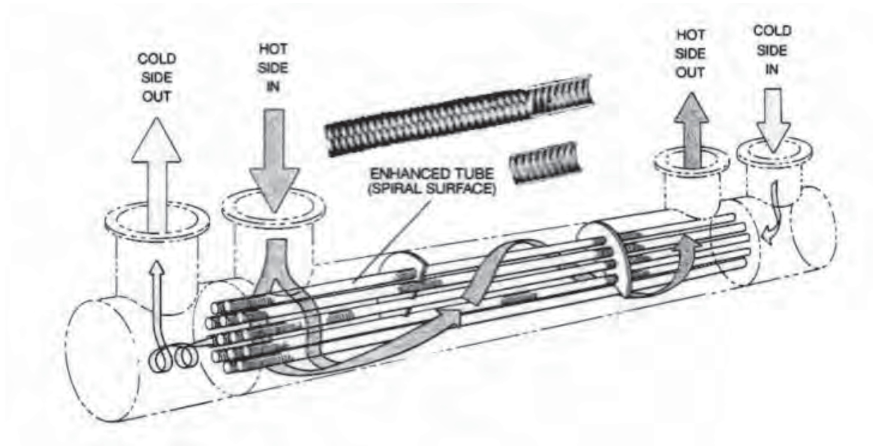


Figure 3-14: Direct Expansion (DX)

Figure 3-15:
Polypropylene
Internal Baffles



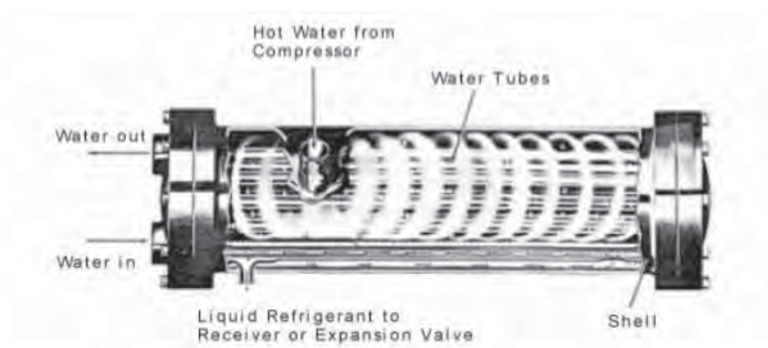
Condensers

There are a number of different kinds of condensers manufactured for the packaged water chiller. These include water-cooled, air-cooled, and evaporative-cooled condensers. (Air-cooled and evaporative condensers will be discussed later in this chapter with cooling towers and heat rejection devices.) Numerous types of water-cooled condensers are available including shell and tube, double pipe, and shell and coil. This discussion focuses on the condenser most commonly used on packaged water chillers—the shell and tube heat exchanger.

A horizontal shell and tube condenser has straight tubes through which water is circulated while the refrigerant surrounds the tubes on the outside. Hot gas from the compressor enters the condenser at the top where it strikes a baffle. The baffle distributes the hot gas along the entire length of the condenser. The refrigerant condenses on the surface of the tubes and falls to the bottom where it is collected and directed back to the evaporator. The bottom tubes are usually the first pass (coldest) of the condenser water and are used to subcool the refrigerant. Often the condenser is used as the refrigerant receiver where it is stored when not in use.

The tubes can be enhanced (ribbed) on both the inside and outside. However, since the condenser water often comes from an open cooling tower, the inside of the condenser tubes may become fouled and require mechanical cleaning. Inside enhancement—usually with straight or spiral grooves—may be problematic because the grooves will be the first areas to become fouled. Research indicates that fouling becomes a problem when concentrations of dissolved solids increase greatly above recommendations and when tube velocities drop below 3 feet per second. Even considering decreased performance of the enhanced condenser tube due to fouling, the heat exchange effectiveness with the enhanced tube may still be greater than a smooth bore tube.

Figure 3-16:
Horizontal Shell and
Tube Condenser



High water velocity is recommended in the condenser tubes as it increases the heat transfer effectiveness and reduces fouling. Water velocities of at least 3 feet per second and a maximum of 12 feet per second are recommended. A minimum flow is needed to maintain flows above the laminar range and the maximum flow protects the tubes from erosion and keeps pressure drops to a minimum. In the HVAC industry, water pressure drops through the condenser are seldom over 25 to 30 feet of water column.

Water-cooled condensers are usually multiple pass, with 4 pass being most common but up to 8 pass available. On positive displacement compressors it is not uncommon for the condenser to be split into two circuits. The condenser waterside can be split into two separate tube bundles to accommodate a heat recovery mode or to add a level of redundancy in the event the tubes need cleaning while the machine is still operational.

Safety Controls, Accessories, and Options

Safety Controls

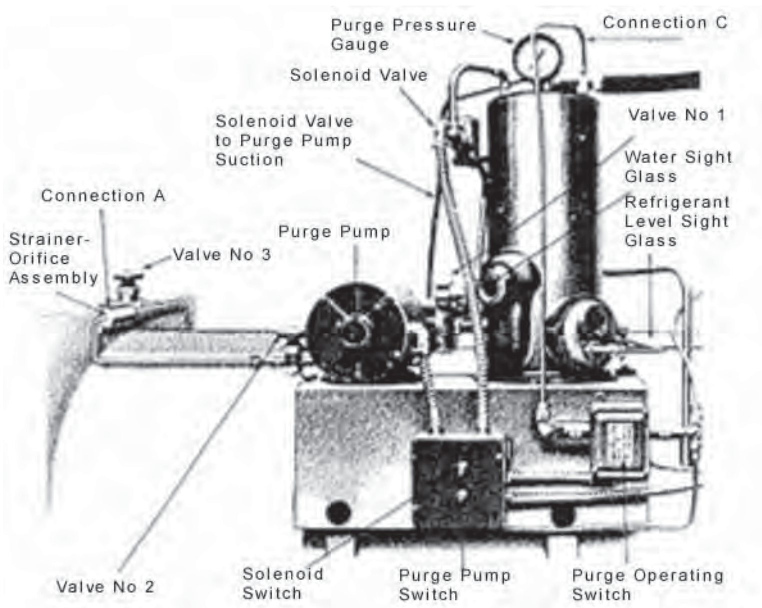
Safety controls protect the unit under abnormal conditions. Safety controls usually trip out the compressor motor and will require manual reset. Typical safety controls include:

- High Refrigerant Pressure – Cutouts, Relief Valves
- Low Pressure – Suction Gas, Lubrication
- High Temperature – Motors, Refrigerant, Lubrication
- Low Refrigerant Temperature
- Time Delay
- Low Voltage/Phase Loss/Phase Reversal
- High Current
- Evaporator and Condenser Proof of Water Flow

Purge Units

Centrifugal chillers that use low-pressure refrigerant, such as R-11 and R-123, operate below atmospheric pressure. When they leak, air and moisture are drawn into the machine. Purge units remove the non-condensables that collect in the condenser during normal operation and ultimately reduce the heat transfer effectiveness, causing greater refrigerant head pressures. Moisture inside the unit causes the formation of acids that break down the oil and increase internal corrosion. Purge units consist of compressors, motors, separators, and condensers that can be automatic or manual. Automatic purge units are preferred because they maintain the highest chiller efficiencies possible. Purge units that reduce refrigerant losses during operation should always be used. Discharge from purge units must be piped outdoors.

Figure 3-17:
Purge Unit



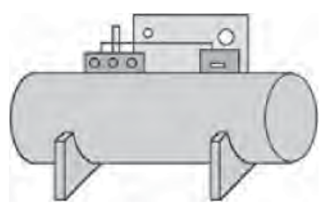
Oil Coolers

Lubricants must be cooled, especially those used with screw machines. A small heat exchanger is provided for this purpose. The heat can be rejected through a city water connection or a chilled water connection, or may be air cooled or internally cooled by the refrigerant.

Pump-out Unit

A refrigerant transfer unit may be provided to make it easier to service machines. The pump-out unit consists of a storage tank to hold the refrigerant charge, a small compressor, an air- or water-cooled condenser, a lubricant reservoir and separator, valves and interconnecting piping. Sometimes the refrigerant can be pumped into the condenser and valved-off, which is sufficient for most maintenance procedures.

Figure 3-18:
Pump-Out Unit



Free Cooling

Centrifugal chillers can be provided with a free cooling option. If the machine must operate during cold weather, and the condenser water can be made colder than the chilled water, the chiller can operate as a thermal siphon. During this operation, low temperature condenser water condenses the refrigerant, which either drains by gravity or is pumped back into the evaporator. Since the chilled water temperature is higher than the condenser temperature, the refrigerant will evaporate and migrate back to the condenser. Free cooling can produce up to 40% of the capacity of the machine depending on the chilled water and condenser water temperature differential applied.

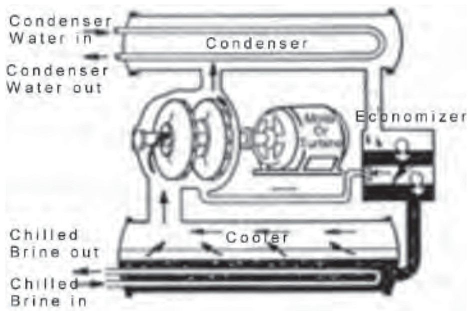


Figure 3-19:
Free Cooling in
Centrifugal Chillers

Heat Recovery

Heat recovered from chillers can be used to heat buildings, domestic hot water, or a wide variety of low temperature heating applications. Two types of heat recovery can be applied to chillers; a desuperheater condenser placed in immediately at the discharge of the compressor and in series with the chiller's main condenser, and parallel condensers called a double bundle condenser. Condensers used to heat potable hot water have to be double tube vented such that any refrigerant leaks can not contaminate the domestic water circuit. The economics of applying heat recovery condensers must consider the load profile of the source to be heated.

Desuperheater condensers are generally applied on reciprocating chillers, particularly air cooled machines. These condensers roughly recover 30% of a chiller's heat rejection capacity and can often make hot water as high as 140° F depending on load and condensing temperatures. Desuperheaters can slightly improve chiller efficiencies. As these condensers are located in series with the unit's main condenser, they must be designed for a very low refrigerant pressure drop. Often desuperheaters are field retrofit to chillers.

Double bundle condensers are usually applied to centrifugal chillers and can recover the machine's entire heat rejection capacity. A double bundle condenser is typically split into two separate tube bundles, or two separate condensers, with the heating water piped to one side and the cooling tower water piped to the other side. Heat is first rejected to the heating bundle and when the heating requirement decreases, the extra heat is rejected to the cooling tower. A control strategy, called a "load shed economizer," provides an intermediate step before the heat is rejected to the cooling tower. A typical control strategy requires the cooling tower water temperature to be elevated to achieve the heat recovery condenser's leaving water temperature. This step increases the outdoor air to various cooling coils throughout the system, thereby decreasing the cooling load on the machine. Only when the cooling load is decreased as much as possible is the heat rejected to the cooling tower. On low pressure refrigerant chillers, hot water temperatures are limited to 110° F as these chillers' condensers are not pressure rated. Also double bundle condensers can be highly inefficient as the condensing head pressure will be elevated to achieve even the smallest amount of heat recovery, and at low heat recovery loads and high machine cooling loads, the extra energy expended at the compressor could have more inexpensively been used to heat the water in an electric water heater. Due to the low temperatures recovered with double bundled condenser and load matching requirements to recover heat efficiently, double bundle condenser are rarely applied.

Demand Limiter

A demand limiter or current limiter reduces compressor capacity during periods of high power consumption (like start-up), thus helping prevent high utility demand charges. Some chiller control strategies use the demand limiter to control the staging of the chillers.

Automatic Tube Cleaners

Automatic tube cleaners send wire brushes through the condenser tubes to keep fouling to a minimum. These devices help keep the chiller operating at peak efficiency and are most appropriate where water quality is a problem.

Performance Characteristics and Efficiency Ratings

Performance Issues

There are a number of variables that determine the operational characteristics and energy performance of water chillers. A chiller is selected to meet a specific maximum capacity requirement at certain design conditions; to meet this capacity at specific (maximum) power draw; and to have specific part-load operation characteristics. To design chillers that meet the performance specifications, manufacturers of packaged water chillers must consider a very wide range of variables. These variables include:

- Compressor Design
- Internal Refrigerant Pressure Drops
- Heat Gains – Motors, Oil Pumps, Casings
- Over/Under Compression
- Motor Efficiency
- Use of Refrigerant Economizers
- Surface Area of Evaporators/Condensers
- Tube Heat Transfer Coefficients – Fouling, Tube Enhancement, Velocity of Fluids
- Refrigerant

Each design decision has first-cost implications. Because of this complexity, products on the market have a wide variety of performance characteristics. The following discussion and comparison chart lay out the broad performance and efficiency issues and provide information that will help in selecting the appropriate equipment for the job.

Table 3-2: Water Cooled Chiller Comparison Chart

Chiller Type	Capacity [1] Range (tons)	First Cost [2] Range ($/ton)	COP Range	IPLV Range (COP)
Reciprocating/Scroll	50 - 230 (400)	$200 - $250	4.2 - 5.5	4.6 - 5.8
Screw	70 - 400 (1250)	$225 - $275	4.9 - 5.8	5.4 - 6.1
Centrifugal	200 - 2000 (10,000)	$180 - $300	5.8 - 7.1	6.5 - 7.9
Single-Effect Absorption	100 - 1700	$300 - $450	0.60 - 0.70	0.63 - 0.77
Double-Effect Absorption	100 - 1700	$300 - $550	0.92 - 1.20	1.04 - 1.30
Engine Driven	100 - 3000 (10,000)	$450 - $600	1.5 - 1.9	1.8 - 2.3

[1] Capacities in parentheses are maximum sizes available
[2] First cost includes allowance for contractor mark-ups

Chiller Efficiency Ratings

At peak design conditions the efficiency of water chillers is rated by "coefficient of performance" or COP. The COP is the ratio of the rate of heat removal to the rate of energy input in consistent units for a complete refrigerating system or some specific portion of that system under designated operating conditions. The formula for COP is:

$$COP = \frac{Net\ Useful\ Refrigerating\ Effect}{Energy\ Supplied\ from\ External\ Sources}$$

EQUATION 3-1

The higher the number, the more energy efficient the machine. ASHRAE Standard 90.1-20047 and Title 24-20058 provide minimum energy efficiency standards for water chillers. Chiller efficiencies are also discussed in terms of "kW/ton" for peak ratings. This is another way of describing the COP [COP = 3.516/(kW/ton)]. The lower the kW/ton, the more energy efficient the machine. Standard chiller ratings are based on "ARI conditions," which set standard parameters for the rating capacity of different machines. These parameters are established in American Refrigeration Institute (ARI) Standards 550/590 (vapor-compression chillers) and 560 (absorption chillers). For water chillers the ARI rating conditions are:

TABLE 3-3:
ARI 550/590-2003 AND 560-2000 RATING CONDITIONS FOR WATER CHILLERS

Leaving Chilled Water Temperature	44°F
Evaporator Water Flow Rate	2.4 gpm/ton
Entering Condenser Water Temperature	85°F
Condenser Water Flow Rate (Electric)	3.0 gpm/ton
Condenser Water Flow Rate (Absorber)	3.6 gpm/ton (single stage) 4.5 gpm/ton (two stage)
Ambient Air (for air-cooled)	95°F
Fouling Factors	0.00010 h-ft^2-F/Btu (Evaporator) 0.00025 h-ft^2-F/Btu (Condenser)

Another useful energy efficiency rating is the "integrated part-load value" or IPLV. The IPLV is a single-number figure of merit based on part-load COP or kW/ton. Part-load efficiency for equipment is based on the weighted operation at various load capacities for the equipment. The equipment COP is derived for 100%, 75%, 50%, and 25% loads (with consideration for condenser water relief) and is based on a weighted percentage of operational hours (assumed) at each condition. A "weighted average" is determined to express a single part-load efficiency number. Condenser water relief assumes that the temperature of the water entering the condenser declines as a straight line from 85°F at 100% load to 65°F at 50% load and below, implying a correlation between weather and cooling load. This represents a 4°F decline for a 10% change in load.

The "nonstandard part-load value" or NPLV is another useful energy efficiency rating. This is used to customize the IPLV when some value in the IPLV calculation is changed, such as using 42°F leaving chilled water in lieu of 44°F, or modifying the number of hours at each load.

While IPLV and NPLV are useful energy performance indicators for individual chillers, recent ARI data shows that 80% of all chillers are installed in multiple chiller plants. Individual chillers operating in a multiple chiller plant may be more heavily loaded than single chillers within single chiller systems. When evaluating a multiple chiller plant, a comprehensive analysis must be used to predict the chilled water system performance.

Chiller refrigerants and performance ratings are a moving target. At the time of this writing, the phase out of R-22 has begun and the refrigeration manufacturers are reengineering many of their products. The best source for current product offerings are the manufacturer websites presented at the front of this documentchapter.

Reciprocating and Scroll Chillers

These chillers are widely used in tonnage ranges from 50 to 230 tons although they are available up to much larger sizes (400 tons and up). Most frequently the compressor design is semi-hermetic compressors. Although a few of the manufacturers are still offering reciprocating compressors, scrolls have largely taken over this market. Capacity modulation is achieved through staging of multiple compressors that are grouped (piped in parallel) in several circuits. This creates some redundancy should a compressor fail. As positive displacement machines, they retain near full cooling capacity even when operated at conditions above the design conditions, and they are, therefore, very suitable for air-cooled applications. For the same reason, they are also suitable for use as heat recovery machines. Control is achieved by stepping unloaders and cycling compressors on/off, which creates a choppy part-load performance curve. Reciprocating and scroll chillers tend to be low first-cost machines.

Rotary Screw Chillers

Rotary screw chillers are also positive displacement machines. Like reciprocating and scroll machines, they are particularly suitable as air-cooled chillers but are popular in both air- and water-cooled configurations. Screw chillers tend to be most cost competitive in the 100 ton to 300 ton range although they are available in a wider range of capacities. In the low capacities they cannot compete with scroll chillers and in the high capacities centrifugals tend to be more cost effective. Centrifugal chillers are also more efficient.

There is no practical design advantage of single versus twin screw except the single screw may be slightly quieter. The machines have excellent turndown capability. Some chillers incorporate multiple compressors. This provides additional efficiency advantages during part-load and allows unloading below 10%. Screw chillers are inherently more efficient than scroll compressors because they incorporate refrigerant economizers (discussed above). They have very few moving parts and have balanced forces on the main bearings. As a result these machines are very reliable. Screw machines are traditionally controlled with a slide valve and are fully modulating. At least one manufacturer uses unloaders and as described above, VSD control is now being offered. Screw chillers tend to be very noisy at design conditions due to the high speed of operation. The variable speed driven screws offer significant acoustical benefits at low loads and have less wear and tear on the bearings.

Centrifugal Chillers

Centrifugal chillers have the highest efficiency ratings of all the chillers discussed. They are available in sizes from 80 tons to 10,000 tons but the most common sizes are from 200 to 2,000 tons. Above 2,000 tons they are generally custom -built. They are available in both air-cooled and water-cooled versions but because of very low COPs and very high initial cost, air-cooled centrifugal chillers are very seldom used.

Centrifugal chillers are controlled with inlet guide vanes which allow for full modulation to as low as 10% to 15% capacity (with condenser water relief). Note that chiller efficiency drops off severely at low loads. Variable-speed drives can be added, as discussed above, to enhance the part-load operation characteristics and are often cost effective when evaluated through life-cycle cost analysis. In addition to the energy savings, centrifugal chillers with variable speed drives are quieter at part load and are likely to last longer. HGBP can also be used but should only be considered if very low turndowns (at elevated condenser temperatures) are required. Centrifugal chillers are the flagship products of the major manufacturers and, as such, are generally of the highest quality and reliability.

Because of the economics of centrifugal chiller manufacturing, there are product differences among all the major manufacturers. There are countless pros and cons to the various features of these products; the following discussion presents some of the main differences:

- ***Gear Drive vs. Direct Drive***. Direct-drive chillers (except Turbocore) operate at 3,600 rpm. Gears allow the impeller to rotate at speeds up to 35,000 rpm. This allows smaller impellers to be used, reducing the machine's first cost. There is an efficiency loss in the gear train of 1.5% to 2%. Also the gears have additional bearings and require regular maintenance, which direct drive machines do not.

  The proper selection of impeller diameter and gear ratio allows the machines to be selected very near their highest performance level or sweet spot, whereas the direct-drive machines, because of limited impeller diameter choices, sometimes are selected several efficiency points away from their sweet spot. Direct-drive machines sometimes have multiple stages (more than one impeller). In this situation, economizers can be added to enhance the energy performance of the machine.

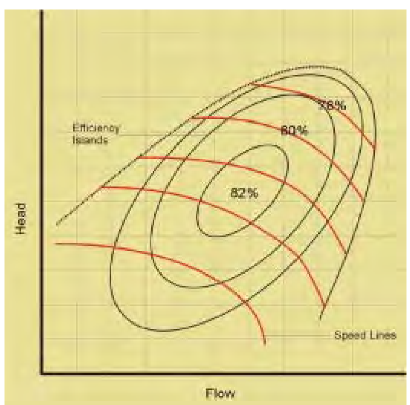


Figure 3-20: Performance Level

- ***Open Drive versus Hermetic***. Hermetic centrifugal chillers have the motor totally enclosed within the chiller casing. The motors are kept clean and are cooled by the refrigerant stream. Hermetic machines have a lower likelihood of refrigerant leakage than open machines. Motor failures in hermetic machines are almost always catastrophic, requiring long downtime periods and great expense to correct.

  Open-drive machines have the motor located outside the casing. Efficiency ratings do not include motor losses (4% to 5% on larger machines). The heat from the open-drive motor must be removed from the machine room, which usually requires additional mechanical cooling. In the event of a catastrophic motor failure, the open-drive machine can be repaired and placed back in service relatively easily, whereas the hermetic machine will require significantly more attention. Luckily motor failures are rare.

Open-drive machines have seals that leak and are subject to failure. On high-pressure machines refrigerant can leak out with dire consequences and on low-pressure machines air can leak in, causing more purge compressor time and loss of efficiency. If a high-pressure open machine is purchased, it is important to obtain a special warranty for refrigerant replacement (not just parts and labor).

- ***Fixed Orifice versus Float Valve***. When a fixed orifice is used as the thermal expansion device, a minimum differential pressure must be held between the condenser and evaporator to assure proper refrigerant flow. This may limit the degree of condenser water relief that can be obtained during off-peak time, with the consequence that the machine will not have as good a part-load performance as a machine with a float valve.
- ***Oil Return***. Chillers may have an oil pump, but most require a minimum differential pressure between the condenser and evaporator to maintain to assure proper refrigerant flow. This condition is often exerted by the manufacturer requiring a minimum 20°F between the leaving chilled water temperature and the entering condenser water temperature. Although the manufacturer, who provides the oil pump as a standard, cites much improved chiller efficiencies at condenser water temperature at 10° F above leaving chilled water temperatures, this efficiency improvement is not realized when considering the whole system energy consumption. Generally, at the low chiller loads where this cold condenser water temperature can be generated, the additional cooling tower fan consumption to make this water temperature offsets the compressor savings.

Absorption Chillers

Absorption chillers can be either single or double effect. Single-effect chillers have COPs of 0.60 to 0.70 and double-effect chillers have COPs of 0.92 to 1.20. Because the double-effect machines are 50% to 100% more efficient than the single-effect chillers, there is little doubt about which to choose if absorption is being considered. Single-effect chillers are beneficial where waste steam is available or where hot water temperatures are not high enough to fuel a double-effect absorption chiller. Triple-and quadruple-effect machines are being developed but are not yet on the market.

Absorption machines are quite simple, and require just a few moving parts, including pumps and burner. Modulating the steam valve or burner controls capacity, and part-load characteristics are very linear over the range of operation to a minimum of about 40% of peak capacity. Variable-speed refrigerant and solution pumps allow much closer control than previous designs.

Absorption machines can be direct- or indirect-fired. Direct-fired machines have the advantage that they can also be used to heat the building and/or domestic hot water. If a direct-fired absorption machine is also used as a heater, the avoided cost of a separate boiler and boiler room (space) may help offset some of the added cost of the machine.

Sizes for absorption chillers range from 100 to 1,700 tons. Absorption machines typically cost two or more times that of an electric-driven chiller. Because a double-effect absorption machine will require 3.5 to 4.5 gpm/ton of condenser water, cooling towers are larger than with an electric chiller plant.

Sometimes the economics of using this type of chiller make it the best choice. The following are reasons an absorption chiller may be chosen:

- High electrical cost including demand, with low natural gas cost
- Hybrid of absorption and electric chillers to reduce demand charge
- Electrical service not available or too costly to upgrade
- Gas from landfill, solar, or biomass available
- Waste steam or low cost steam readily available
- Need for chiller during prolonged periods on emergency power
- No CFCs, no ODP, and low direct GWP comparable with other alternatives

Absorption chillers have some distinct operating disadvantages that should be considered when designing a plant:

- They cannot produce water temperatures as low as electric chillers. The minimum chilled water supply temperature is typically 43°F which limits their use with thermal energy storage systems.
- They take longer to start up and to shut down, thus requiring longer time between cycles than electric chillers. The start up is in part due to the capacitance of their refrigerant. The cycling is due to the chemistry. As chilled water flow is maintained through the chiller during start-up and shut-down periods, at lower or no produced cooling capacity, maintenance of system chilled water supply temperature can be an issue. This limits their use in plants like data centers where rapid deployment is an issue.
- They cannot abide low flows or temperatures on the condenser water side. This limitation can hamper the performance of mixed fuel plants where variable speed driven electric centrifugal chillers might be optimized by low condenser flows and temperatures at part load conditions. Primary/secondary condenser water pumping may be required for most efficient plant operation.
- They are significantly larger than electric chillers and require larger towers.
- They may not last as long as electric chillers and are subject to failure if not properly maintained. The absorption chiller's chemistry is corrosive and will eat the chiller up if the inhibitors are not properly maintained.

Turbine-Driven and Engine-Driven Chillers

While not a large segment of the chiller market, turbine-driven and engine-driven chillers are sometimes economically viable. Both use the same vapor compression cycle as an electric machine except it uses a reciprocating engine, or a gas- or steam-driven turbine as the prime mover. For larger applications, the refrigeration component is usually an open-screw or centrifugal chiller. Because they use variable-speed technology, the part-load characteristics are very good.

Engines use natural gas or diesel fuel. Some are hybrid units that have both an engine and an electric motor so that the fuel may be switched depending on the utility rates at the time. Engines require heat rejection from the jacket water. Heat can be rejected out the cooling tower (through a heat exchanger) or, with smaller units can be air-cooled. The jacket water heat is available for heat recovery of domestic water or other loads occurring at the same time as the engine runs. Heat recovery water temperatures at 180° F to 200° F are easily produced, availing heat recovery to a wider range of loads, which if amply available can significantly impact the economics.

Engines need additional maintenance, with top end overhauls required every 12,000 hours and complete overhauls at 35,000 hours. Reciprocating engines are much louder than electric-driven or absorption machines and may require special enclosures or acoustical abatement. Natural gas and steam turbines are a very small part of the market and are used in very large plants (up to 10,000 tons).

As there are limited manufacturers of these products, care is required in procureing them. A flat specification for a turbine driven chiller on a large plant can give a single manufacturer an unfair advantage on bidding the entire plant including the turbine and electric chillers.

Heat Rejection

One of the prime purposes of the chilled water plant is to reject unwanted heat to the outdoors. This is accomplished in a number of different ways. Utility supplied water can remove heat from the condenser and dispose of it down the drain. This method has lost all favor since the cost of the water and disposal has become prohibitive, and the need to conserve resources has been recognized. Ground water has been used in a similar manner. Coupled with precooling coils in the airstream and reintroduction wells, ground water has been successfully but only very rarely used. Environmental concerns have limited the use of rivers and lakes as a heat rejection source but cooling ponds are still sometimes used. Given the high cost and poorly understood performance of cooling pond heat rejection, this alternative is not discussed in this guide. The primary means of heat rejection in the HVAC industry are the cooling tower, the air-cooled refrigerant condenser, and the evaporative refrigerant condenser.

Manufacturer Data

The cooling tower manufacturers are continually modifying their product offerings. The reader is encouraged to browse the manufacturers' websites for the most current information on technologies and product ratings. The major manufacturer websites are:

- Baltimore Aircoil Company: http://baltimoreaircoil.com/english/products/ct/index.html
- Clearwater System Corporation (the Dolphin non-chemical water treatment system): http://www.clearwater-dolphin.com/index.htm
- Evapco: http://www.evapco.com/
- SPX Cooling Technologies Inc. (Formally Marley Cooling Towers and Ceramic Cooling Towers): http://www.marleyct.com/package/

Cooling Towers

Simply put, evaporation is a cooling process. More specifically, the conversion of liquid water to a gaseous phase requires the latent heat of vaporization. Cooling towers use the internal heat from water to vaporize the water in an adiabatic saturation process. A cooling tower's purpose is to expose as much water surface area to air as possible to promote the evaporation of the water.

In a cooling tower, approximately 1% of the total flow is evaporated for each 12.5°F temperature change. There are several important terms used in the discussion of cooling towers:

- **Range**: The temperature difference between the water entering the cooling tower and the temperature leaving the tower.
- **Approach**: The temperature difference between the water leaving the cooling tower and the ambient wet-bulb temperature.

The performance of a cooling tower is a function of the ambient wet-bulb temperature, entering water temperature, air flow and water flow. The dry-bulb temperature has an insignificant effect on the performance of a cooling tower. "Nominal" cooling tower tons are the capacity based on a 3 gpm flow, 95°F entering water temperature, 85°F leaving water temperature, and 78°F entering wet-bulb temperature. For these conditions the range is 10°F (95-85) and the approach is 7°F (85-78).

Types of Cooling Towers

Cooling towers come in a variety of shapes and configurations. A "direct" tower is one in which the fluid being cooled is in direct contact with the air. This is also known as an "open" tower. An "indirect" tower is one in which the fluid being cooled is contained within a heat exchanger or coil and the evaporating water cascades over the outside of the tubes. This is also known as a "closed circuit fluid cooler."

The tower airflow can be driven by a fan (mechanical draft) or can be induced by a high-pressure water spray. The mechanical draft units can blow the air through the tower (forced draft) or can pull the air through the tower (induced draft). The water invariably flows vertically from the top down, but the air can be moved horizontally through the water (crossflow) or can be drawn vertically upward against the flow (counterflow).

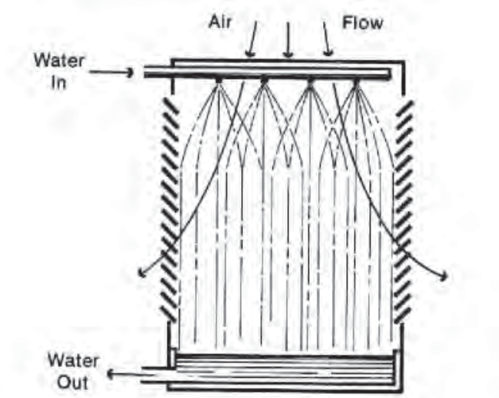
Water surface area is increased by using "fill." Fill can be "splash-type" or "film-type." Film-type fill is most commonly used and consists of closely spaced sheets of PVC arranged vertically. Splash-type fill uses bars to break up the water as it cascades through staggered rows.

Typically, in the HVAC industry, cooling towers are "packaged" towers that are factory fabricated and shipped intact to a site. "Field-erected" towers mostly serve very large chiller plants and industrial/utility projects. When aesthetics play a role in the selection of the type of tower, custom designed field-erected cooling towers are sometimes used. In these towers, the splash-type fill is often made of ceramic or concrete blocks.

The following is a discussion of the most common types of cooling towers encountered in the HVAC chilled water plant.

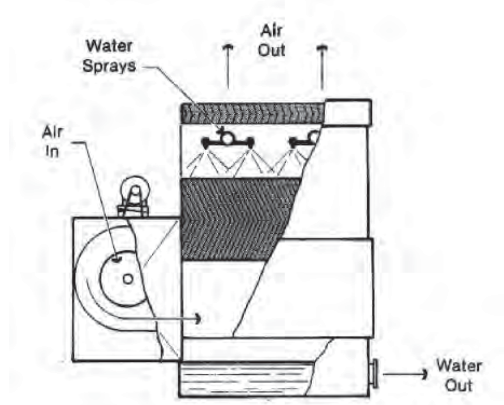
- ***Spray Towers***. Spray towers distribute high-pressure water through nozzles into a chamber where air is induced to flow with the water spray. There are no fans. The air exits out the side of the tower after going through mist eliminators. Spray towers are seldom used. One problem is that the nozzles are easily plugged by the precipitation of mineral deposits and by airborne particulates that foul the water. Capacity is controlled by varying the water flow through the tower. This can be accomplished by using multiple-speed pumps or variable-speed drives on the pumps, or by passing water around the tower. Varying the water flow through the condenser of a chiller is not always recommended, as will be discussed in the next chapter. Because air velocities are very low, spray towers are susceptible to adverse effects from the wind. Spray towers are very quiet and can have a very low first cost.

Figure 3-21:
Spray Tower



Forced Draft Cooling Towers. Forced draft towers can be of the crossflow or counterflow type, with axial or centrifugal fans. The forward curved centrifugal fan is commonly used in forced draft cooling towers. The primary advantage of the centrifugal fan is that it has capability to overcome high static pressures that might be encountered if the tower were located within a building or if sound traps were located on the inlet and/or outlet of the tower. Crossflow towers with centrifugal fans are also used where low profile towers are needed. These towers are relatively quieter than other types of towers. Forced draft towers with centrifugal fans are not energy efficient. The energy to operate this tower is more than twice that required for a tower with an axial fan. Another disadvantage of the forced draft tower is that, because of low discharge air velocities, they are more susceptible to recirculation than an induced draft tower. This is discussed in further detail below.

Figure 3-22:
Forced Draft towers



Induced Draft Cooling Towers. The induced draft tower is by far the most widely used and energy-efficient cooling tower available in the HVAC industry. These towers can be crossflow or counterflow and use axial fans. Most field-erected cooling towers are the induced draft type. Because the air discharges at a high velocity, they are not as susceptible to recirculation. The large blades of the axial fan can create noise at low frequencies that is difficult to attenuate and, depending on the location on the property, could cause problems. The axial fans have either a belt drive or direct (shaft) drive. Direct drive fans use gear reducers to maintain the low speeds of the fan. Belt drive towers have the disadvantage that the motor and belts are located within the moist air stream of the tower exhaust, making them more susceptible to corrosion and fouling and more difficult to maintain. Belt drive towers usually cost less than towers with direct drives. Belt drive towers allow the use of "pony" motors as a means of speed control. This will be discussed further below.

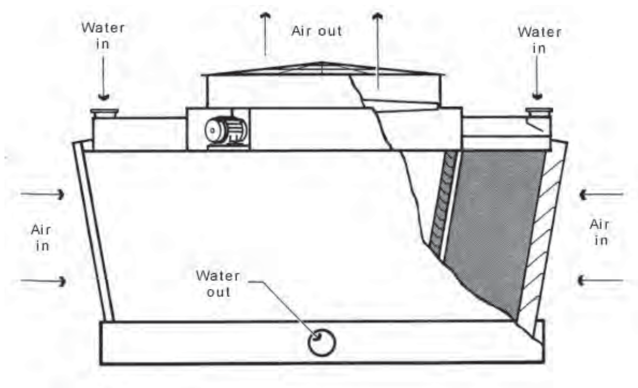


Figure 3-23:
Induced Draft Towers

Closed Circuit Fluid Coolers. As discussed above, one advantage of a closed circuit fluid cooler is that the fluid is located within a coil (rows of tubes) rather than being open to the environment. A pump draws water from a sump and delivers it to a header where the water is sprayed over the coil. The closed piping can be advantageous if the fluid:

- has a high pressure (for instance if the tower is located below the condenser);
- is mixed with fluids from other systems (like the chilled water); or
- has the primary pump located remotely from the tower.

With proper initial chemical treatment, the fluid (usually some form of glycol solution) does not foul the condenser tubes, so chiller maintenance is reduced and energy efficiency is always at peak. Because of the additional heat exchange process, for the same capacity as an open tower, the closed circuit fluid cooler is physically much larger and significantly more expensive than conventional open towers.

Application Issues

- ***Siting and Recirculation***. When the saturated air leaving the cooling tower is drawn back into the intake of the tower, the recirculation that occurs degrades the performance of the tower. Wind forces create a low-pressure zone on the downwind (lee) side of the tower that causes this phenomenon. Wind forces on the lee side of the building can also create downward air movement. When cooling towers are located in such a way that the discharge from one tower is directed into the intake of an adjacent tower, recirculation can also occur. Recirculation is a greater problem when cooling towers are confined within pits, or have screen walls surrounding them. If the tower is sited in a pit or well, it is essential that the tower manufacturer be consulted to determine the proper location of the outlet and minimum clearances for the air intake. As previously discussed, the potential for recirculation is greater with forced draft towers than with induced draft towers.

The Cooling Tower Institute (CTI) recommends that recirculation effects be accounted for in the selection of the tower. Their tests show that as much as 8% of the discharge air could be recirculated back into the intake and that the worst conditions occur with winds of 8 to 10 miles per hour. Where recirculation is a concern, a rule of thumb is that the entering wet-bulb (EWB) temperature used to select the tower should be increased by 1°F above the ambient temperature to account for recirculation effects.

- ***Capacity Control***. Like most air-conditioning equipment, cooling towers are selected to maximum peak capacity at design weather conditions. Of course, most of the time they operate at less than peak capacity. There are a number of methods used to control the temperature of the water leaving the cooling tower, including:

  On/Off: Cycling fans is a viable method but leads to increased wear on belts and drives (if used) and can lead to premature motor failure. This is the least favorable method of controlling temperature.

  Two-Speed Motors: Multiple wound motors or reduced voltage starters can be used to change the speed of the fan for capacity control. This method is cost effective and well proven. Because of basic fan laws, there are significant energy savings when the fans are run at low speed. One pitfall with two-speed fans is that when switching from high to low speed, the fan rpm must reduce to below low speed before energizing the low-speed step. Strategies for optimum operation of two speed fans will be discussed in the next chapter.

  Pony Motors: This is another version of the two-speed approach. A second, smaller motor is belted to the fan shaft. For low-speed operation the larger motor is de-energized and the smaller motor energized for a lower speed. This is a cost-effective and energy efficient approach. Again, when going from high speed to low speed, the fan must slow down sufficiently before energizing the low-speed motor.

  Variable-Speed Drive (VSD): Adjustable frequency drives can be added to the motors for speed control. This method provides the best temperature control performance and is the most energy-efficient method of control. It may also be the most expensive. A life-cycle cost justification should be done before selecting this method. When comparing VSDs with other approaches, the cost of control points for each alternative should be carefully factored into the analysis. One pitfall to avoid with VSDs is to not run the fans at the "critical" speeds. These are speeds that form resonance frequency vibrations and can severely damage the fans. Consult with cooling tower manufacturers before using VSDs. Gear drives, where used in cooling towers, will limit minimum fan speed to 50% to provide adequate gear lubrication unless an oil pump is installed. Otherwise minimum fan speeds of 10% are required to provide necessary motor cooling.

  Modulating Discharge Dampers: Used exclusively with centrifugal fans, discharge dampers built into the fan scroll can be modulated for capacity control. This is a cost-effective way to accomplish close temperature control. Although it does save energy by "riding the fan curve," other methods of capacity control may provide better energy savings results.

Title 24 requires (prescriptive requirement) the use of either variable speed drives, two-speed motors or pony motors on two-thirds of the fan motors for cooling towers when the motors are 7-1/2 hp or larger. In general VSDs are recommended over the other options for the following reasons:

- Lower first cost
- Tighter temperature control
- Lower operating costs
- Reduced noise
- Control information available from the VSD

- ***Chemical Treatment and Cleaning***. Cooling towers are notorious for requiring high maintenance. The use of cooling towers has been linked with the outbreak of legionellosis (Legionnaires' disease). Unfortunately, cooling towers are very good air scrubbers. A 200-ton open cooling tower can remove 600 pounds of particulate matter in 100 hours of operation. Because they are open to the atmosphere, the water is oxygen-saturated which can cause corrosion in the tower and associated piping. Towers evaporate water, leaving behind calcium carbonate (hardness) that can precipitate out on the tubes of the condensers and decrease heat transfer and energy efficiency.
  - Towers must be cleaned and inspected regularly. Well-maintained and regularly cleaned cooling towers have generally not been associated with outbreaks of legionellosis. It is best to contract with a cooling tower chemical treatment specialist. The following are some of the strategies to consider in a good chemical treatment program:
  - Blowdown: To control dissolved solids a portion of the flow of the tower should be discharged into the sewer. A rule of thumb is that for a build-up of no more than 2 to 4 concentrations of hardness, the blowdown rate should be about 0.5 to 1.0% of the total flow rate.
  - Scale Prevention: Control of the pH (acid levels) is extremely important. Usually acids, inorganic phosphates or similar compounds are commonly used to control pH.
  - Corrosion Control: Corrosion can be caused by high oxygen content, carbon dioxide (carbonic acid), low pH, or high dissolved solids. Blowdown is the most practical solution.
  - Biological growth: Slime and algae are handled with shock treatments of chlorine or chlorine compounds. It is best to alternate between two different compounds so that organisms do not develop a tolerance to the chemicals.
  - Foam and Scum: Usually caused from excess organic material. Cleaning the machine is the best remedy.

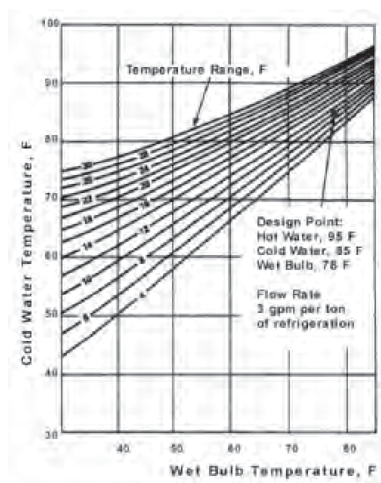
New technologies are being introduced for the treatment and cleaning of cooling towers. One treatment is the introduction of ozone (O_3) into the cooling water. Ozone is a very aggressive oxidizer and when properly applied can be effective at reducing biological growth. One pitfall in the use of ozone is that if left unchecked, large concentrations of ozone will cause runaway corrosion of piping and cooling tower basins. Another promising non-chemical method, the Dolphin System, employs pulsed electromagnetism to remove dissolved solids and inhibit biological growth.

- ***Makeup Water***. Wind can induce cooling tower drift. As a rule of thumb, drift reduces blowdown; thus, make-up water is generally estimated at 2 percent of tower flow rate considering evaporation, drift and blowdown. Drift can be significantly greater if the tower is csited in an open and windy location. Also to be considered is sump capacity and how quickly one would want the sump filled after a maintenance cleaning. Often make-up water lines include meters, useful information in monitoring and maintaining a tower.

Performance of Cooling Towers

Given a fan selection, flow rate, range, entering wet bulb, and fill volume, cooling towers have a wide range of performance characteristics. Typical performance curves show the relationship between these variables at different operating conditions. In reviewing the typical performance curve, one feature not well understood is that for a given range, as the entering wet bulb (EWB) decreases, the approach increases. As EWB drops, it is likely that the load (range) will also decrease for the same flow rate. Yet even at this condition, the approach still increases over design condition. This is particularly important when considering the selection of cooling towers for use with waterside economizers. To obtain the maximum effectiveness at low wet-bulb temperatures, a cooling tower used in a waterside economizer system should be larger than a tower selected just for maximum peak duty.

Figure 3-24: Typical Performance Curves



Cooling towers are relatively inexpensive when compared to the total cost of a chiller plant, and incremental increases in tower size and energy efficiency can be purchased at a very low cost. Skimping on the cooling tower is penny-wise and pound-foolish. Every effort should be made to optimize the selection of the tower. Matching larger fill volumes with lower fan capacities is a very good investment.

For a given design of a cooling tower the manufacturer will normally attribute a maximum and minimum flow condition to the tower. The maximum flow is usually based on the capacity of the water distribution system within the tower to adequately distribute the water over the fill. Too much flow will overflow the tower distribution pans and create a situation where the tower does not get adequate mixing of air and water to perform properly. At minimum flow the water may not distribute evenly across the entire fill. This creates voids where there is no water in the fill. When this happens the air stream will tend to travel through the fill area with no water and will not mix properly with the fill area that has the water. This creates a significant decrease in the expected performance of the tower. Another drawback to operating under the minimum flow is that at the boundary where the water and high velocity air meet, a condition is created where the water is carried up through the fans and the tower "spits" water. Prolonged operation below the minimum water flow can also cause scaling to occur on the fill where the water is missing.

ASHRAE Standard 90.1-2004 and Title 24-2005 carry provisions that establish energy performance requirements for heat rejection devices. Currently, these standards require >38.2 gpm/hp for axial fan towers and >20.0 gpm/hp for centrifugal fan towers at the design conditions of 95°F condenser water return, 85°F condenser water supply and 75°F outdoor air wet-bulb temperatures. Title 24-2005 also carries a prescriptive restriction on centrifugal fan cooling towers where the combined capacity of the cooling towers are 900 gpm or greater. In general, propeller fan cooling towers use ½ of the energy of centrifugal fan towers for the same duty and have a lower first cost. Exceptions are provided for installations with external static pressure such as ducted inlet or discharge or the need for sound traps. If acoustical criteria is important the reader is encouraged to investigate low-noise draw through towers with propeller fans. These towers have the following features:

- Heavier gauge of metal on the fans
- Slower fan speeds
- Low pressure sound traps

These low-noise propeller fan towers are generally as quiet as a centrifugal fan tower without sound traps, are less expensive and have better energy performance.

Cooling Tower Accessories and Options

The following is a list of accessories and options that should be considered when purchasing a cooling tower:

- Filters: Side stream filters include either sand filters or centrifugal separators. Do not use swimming pool sand filters for cooling towers. Side stream filters generally circulate about 10% of the system flow.
- Fan breaks or stops: These are devices that prevent the fan from rotating backwards. Consider these options if multiple cells are used and backflow air flow through a down fan may cause it to rotate in reverse. Starting a reverse-rotating fan can damage the motor. As conditions for reverse rotation seldom occur, these options are rarely applied.
- Vibration Switch: This stops the fan if vibration exceeds a certain limit. It could prevent catastrophic failure of fan. Codes in some areas require the installation of a vibration switch.
- Ladders and Access Platforms: Any area where maintenance personnel need to inspect, repair, or replace equipment should have adequate access. Without easy access, towers may not be maintained to the degree that protects the chiller plant investment.
- Vortex Breaking Inlet Screens: These prevent air from being drawn into the pump suction. This is an essential accessory.

Air-Cooled Refrigerant Condensers

Types

Another method of heat rejection commonly used in chiller plants is the air-cooled refrigerant condenser. This can be coupled with the compressor and evaporator in a packaged air-cooled chiller or can be located remotely. Remote air-cooled condensers are usually located outdoors and have propeller fans and finned refrigerant coils housed in a weatherproof casing. Some remote air-cooled condensers have centrifugal fans and finned refrigerant coils and are installed indoors in what amounts to an air-handling unit. Indoor condensers are only used on small chillers and will not be discussed further here. Air-cooled condensers, whether remote or packaged within an air-cooled chiller, normally operate with a temperature difference between the refrigerant and the ambient air of 10 to 30°F with fan power consumption of less than 0.08 hp/ton (> 69 COP). Maximum size for remote air-cooled refrigerant condensers is about 500 tons, with 250-ton maximum being more common. Air-cooled chillers are available up to 400 tons.

Figure 3-25: Packaged Air-Cooled Chiller

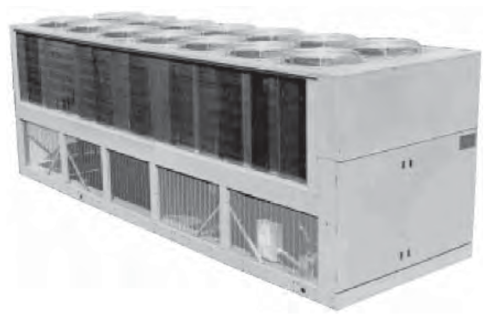
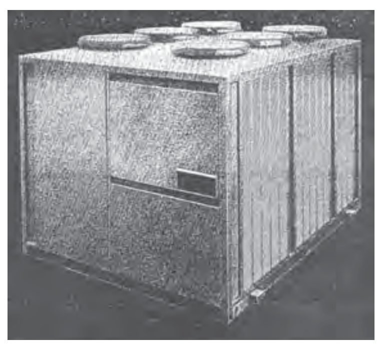


Figure 3-26: Remote Air-Cooled Condenser



There are a number of reasons air-cooled chillers are used. These include:

- Water shortages or quality problems
- Lower cost than water-cooled equipment
- With packaged air-cooled chillers, no need for machine rooms with safety monitoring, venting, etc.
- Less maintenance required than cooling towers

Air-cooled chillers are not as energy efficient as water-cooled chillers. When comparing the energy efficiency of air-cooled to water-cooled chillers, care must be taken to include in the water-cooled chiller the energy consumed by the condenser water pump and cooling tower. Air-cooled chillers have very good part-load performance; as the air temperature drops the

COP improves significantly. Remote air-cooled refrigerant condensers in chilled water plants are very seldom used because of the physical size for the larger tonnage machines. Air cooled chillers are more often used in smaller chiller plants, (generally below 200 tons), as space, water treatment and the additional maintenance cost associated with cooling towers or evaporative condensers outweighs the energy benefit.

Application, Selection, and Maintenance of Remote Air-Cooled Condensers

When selecting remote air-cooled condensers, it is important to match the total heat rejection (THR) from the compressor with the THR of the condenser. The smaller the condenser, the higher the condensing pressure and the higher the energy bills. Computer selections are used to match the compressors and condensers. A graphical method can plot the THR of the compressor against the THR of several condenser selections. When producing graphs of THR of the condenser, the pressure drop of the hot gas line and the condenser itself must be taken into account. This is done with refrigerant temperature drop and is usually kept at 2°F or less.

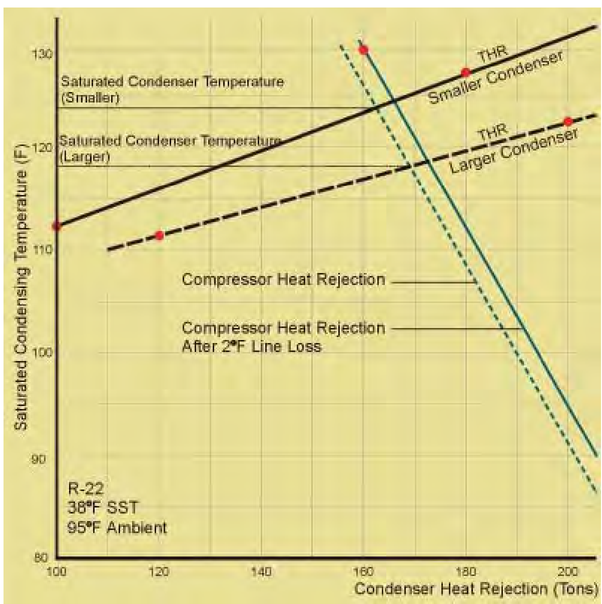


Figure 3-27: Graphical Method (THR of Compressor vs. THR of Condenser Selections)

Designing the piping for remote air-cooled condenser application is somewhat tricky. Care must be taken to properly size the hot gas line to limit pressure drop but also assure that oil is carried by the refrigerant and does not accumulate in the piping. The minimum oil carrying capacity of the hot gas piping needs to account for the lowest load (considering compressor unloading). Although very seldom used, double hot gas risers may be necessary. If the condenser is located below the evaporator, the liquid line must be carefully sized to prevent flashing caused by the pressure drop not only by the pipe friction but also by the change in elevation of the fluid. Additional subcooling may be required in this instance. When the condenser is located above the compressor, care must be taken to prevent the liquid refrigerant and oil from flowing backward by gravity into the compressor. This usually means that the hot gas line runs to the floor before rising and that there is a check valve at the top just upstream of the condenser. Using remote refrigerant condensers greatly increases the likelihood of refrigeration leaks from the piping.

Air-cooled chillers require little maintenance but they do need to have coils cleaned regularly, they require standard lubrication, and the refrigerant charge needs to be periodically checked. If excessive leaves from trees or other debris become a problem, permanent air filters are available to protect the coils. However, air filters slightly degrade the performance of the units and require additional maintenance.

As with siting cooling towers, air-cooled chillers can potentially recirculate the warm discharge air, especially when multiple condensers are located adjacent to one another or condensers are located within a pit or screen wall. Consult the manufacturer's location guidelines for multiple machines or pit locations.

Controls

When air-cooled condensers operate, typically the fan runs continually in conjunction with the compressor. When the outside temperature falls, it is possible to decrease the liquid refrigerant pressure too much to adequately overcome the thermal expansion valve (TXV) pressure drop. In this case controls are required to limit the heat rejection. These controls include:

- Flooded coil: Control valves back up liquid refrigerant into the condenser to limit the heat transfer surface. This requires a receiver and a large refrigerant charge.
- Fan cycling: Usually need multiple fans with one or more cycling on and off to maintain minimum head pressure.
- Dampers: Discharge dampers on condenser fan restrict airflow.
- Variable-speed fans: Fan speed modulates airflow.

For systems not intended to run at cold temperatures (less than 40°F), fan cycling is usually the most appropriate choice for control. For systems intended to run at temperatures down to 0°F, fan speed control or dampers are used.

Evaporative Condensers

Evaporative condensers are not unlike closed circuit fluid coolers. A pump draws water from a sump and sprays it on the outside of a coil. Air is blown (drawn) across the coil and some of the water evaporates causing heat transfer. The hot gas from the compressor condenses inside the tubes. Evaporative condensers are a cross between a cooling tower and an air-cooled refrigerant condenser. These devices are primarily used in the industrial refrigeration business and have little application in the HVAC industry. Some manufacturers produce small packaged water chillers with evaporative condensers as an integral component.

The effectiveness of the evaporation of the water and the refrigerant in the heat transfer process means that for a given load, evaporative condensers can have the smallest footprint of any heat rejection method. The evaporative condenser causes lower condensing temperatures and, as a result, is far more efficient than air-cooled condensing. Maintenance and control of evaporative condensers is similar to the closed circuit fluid cooler. Like cooling towers, the style of the tower can significantly impact fan energy power.

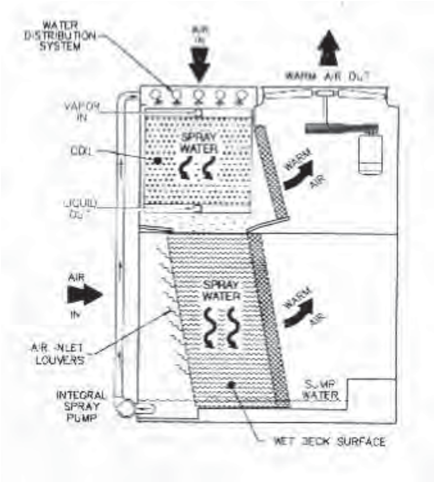


Figure 3-28:
Evaporative Condenser

Pumps

In the chilled water plant centrifugal pumps are the prime movers that create the differential pressure necessary to circulate water through the chilled and condenser water distribution system. In the centrifugal pump a motor rotates an impeller that adds energy to the water after it enters the center (eye). The centrifugal force coupled with rotational (tip speed) force imparts velocity to the water molecules. The pump casing is designed to maximize the conversion of the velocity energy into pressure energy. In the HVAC industry most pumps are single stage (one impeller) volute-type pumps that have either a single inlet or a double inlet (double suction). Axial-type pumps have bowls with rotating vanes that move the water parallel to the pump shaft. These pumps are likely to have more than one stage (bowls). The vertical turbine pump is an example of an axial-type pump and is sometimes used in a cooling tower sump application. Double suction pumps are more likely to be used in high volume applications but either a single inlet or double inlet pump is available with similar performance characteristics and efficiencies.

Most pumps in the HVAC industry are available in bronze-fitted or iron-fitted construction. Usually the pumps have a bronze impeller and wear rings, a bronze or stainless steel shaft sleeve, stainless steel shaft, and a cast iron casing. Centrifugal pumps come with mechanical seals (most common) or packing gland seals. Packing gland seals are sometimes used in condenser water systems, where an accumulation of dirt can damage mechanical seals. Manual petcocks are sometimes used to vent air from the volute. If a system has significant air accumulation, an automatic air vent is used in place of a manual vent.

Manufacturer Data

The pump manufacturers are continually modifying their product offerings. The reader is encouraged to browse the manufacturers' websites for the most current information on technologies and product ratings. The major manufacturer websites are:

- Bell and Gossett: http://www.bellgossett.com/BG-Catalog-English.asp#Commercial
- Paco Pumps: http://www.paco-pumps.com/Catalog/SelectProduct.asp?CompanyID=3&NoState=1
- Taco Pumps: http://www.taco-hvac.com/en/products/Water+Circulation+Pumps+%26amp%3B+Circulators/products.html?current_category=18&PHPSESSID=06bed23c46b2d228a7f599f7283afd41

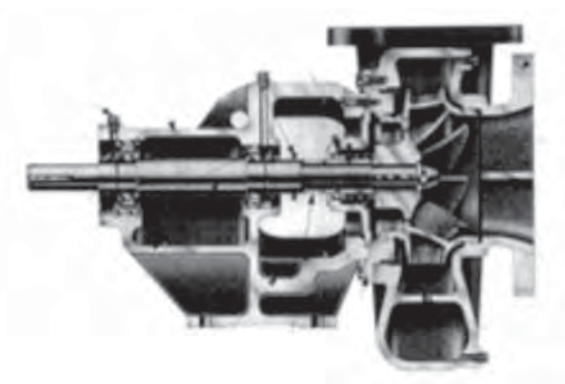
Pump Types

The following is a brief discussion of the various types of pumps used in the chilled water plant.

Single Suction

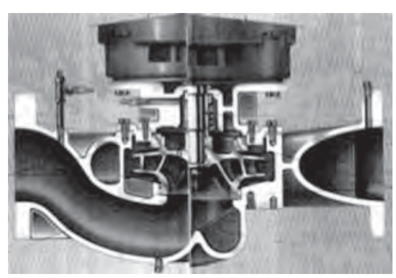
Base-mounted single suction pumps can be either close-coupled or flexible-coupled. Close-coupled pumps use a special motor that has an extended shaft to which the pump impeller is directly connected. The motor and pump cannot be misaligned and they take up less floor space than flexible-coupled pumps. However, replacement motors can have a long lead time and be difficult to get after a breakdown. Flexible-coupled pumps allow the motor or pump to be removed without disturbing the other. The flexible coupling requires very careful alignment and a coupling guard. The flexible-coupled pump is usually less expensive than the close-coupled pump. Usually single suction pumps are preferred for use up to 1,000 gpm but are available up to 4,000 gpm.

Figure 3-29: Base-Mounted Single Suction Pump



In-line pumps have the suction and discharge connections arranged so that they can be inserted directly into a pipe or they can be mounted on a base like other pumps. In the past these pumps were used almost exclusively for small loads with low heads, but now they are available in the full range of sizes. Because of the restricted inlet condition, these pumps are not as efficient as single suction pumps. These pumps can save considerable space but extra care must be taken to assure that pipe stresses are not transferred to the pump casing.

Figure 3-30: Inline Pump



Double Suction

In the double suction pump the water is introduced on each side of the impeller and the pump is flexibly connected to the motor. These pumps are preferred for larger flow systems (typically greater than 1,000 gpm) because they are very efficient and can be opened, inspected and serviced without disturbing the motor, impeller, or the piping connections. Typically, the pumps are mounted horizontally but can be mounted vertically. The pump case can be split axially (parallel to shaft) or vertically for servicing. This pump takes more floor space than end suction pumps and is more expensive.

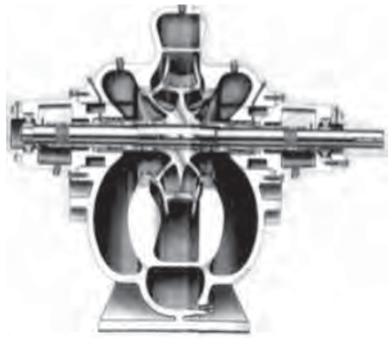


Figure 3-31:
Double Suction Pump

Vertical Turbine

Vertical turbine pumps are axial-type pumps that are used almost exclusively for cooling tower sump applications. These pumps can be purchased with enclosures or "cans" around the bowls when not sump-mounted.

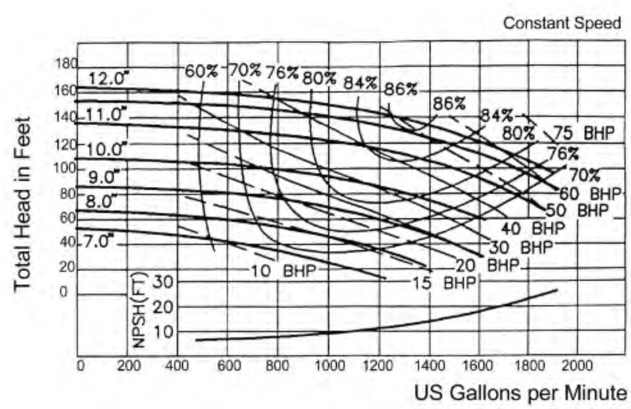
Figure 3-32:
Vertical Turbine

Pump Performance Curves

For a given impeller size and rotational speed, the performance of a pump can be represented on a head-capacity curve of total developed head in feet of water versus flow in gallons per minute. Total dynamic head (TDH) is the difference between suction and discharge pressure and includes the difference between the velocity head at the suction and discharge connection. Starting from zero flow, as the pump delivers more water, the mechanical efficiency of the pump increases until a "best efficiency point" (BEP) is reached. Increasing the flow further decreases the efficiency until a point where the manufacturer no longer publishes the performance (end of curve). Pump performance curves are a family of curves for different size impellers. Notice as the impellers get smaller, the pump efficiency decreases. The power (horsepower) requirements are also shown on the performance curve; notice that the power lines cross the pump curve until one value does not cross. This value is called "non-overloading" horsepower because operation at any point on the published pump curve will not overload the motor. Finally, information on the "net positive suction head required" (NPSHR) is shown on the pump curve. This will be discussed in greater detail below.

Pump curves are also rated as "steep" or "flat." The definition of a flat curve pump is when the pressure from "shut-off head" (head at zero flow) to the pressure at the BEP does not vary more than 1.1 to 1.2 times the pressure.

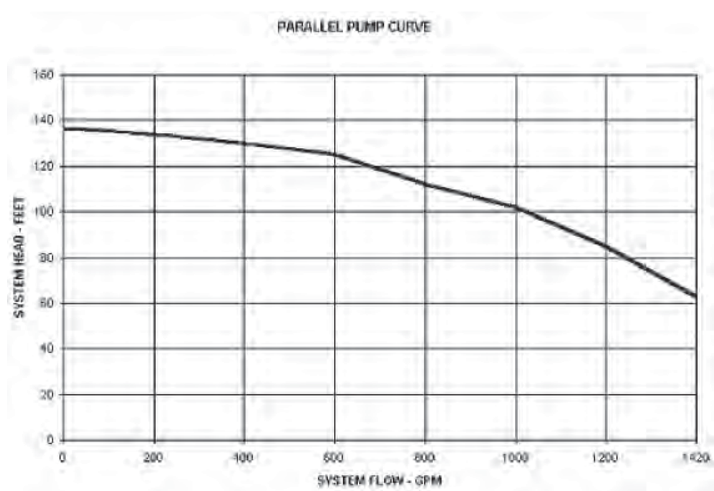
Figure 3-33:
Head Capacity Curve



Parallel and Series Pumping

When two or more pumps are operated in parallel, a combined parallel pump curve can be drawn which holds the head constant and adds the flow. Similarly, a series pump curve can be drawn which holds the flow constant and adds the head. Pumps are rarely placed in series but depending on the system piping practices can actually operate in a series mode.

Figure 3-34:
Parallel Pump Curve



Variable-Speed Pumping

For a given impeller size a family of curves can be drawn to represent the variable-speed performance of a pump. Notice that the BEP follows parabolic curve that looks surprisingly like a system curve (this will be discussed in greater detail below). Also notice that the NPSHR lines follow fairly closely with the published end of curve lines for the various speeds. The power lines decrease rapidly as the speed decreases, which graphically demonstrates the potential power savings of variable-speed operation in variable-flow systems. For a more detailed example of variable-flow applications, refer to Chapters 4 and 6. Some designers have placed variable-speed pumps in parallel with constant-speed pumps with unexpected results. The constant-speed pump will always overpower the variable-speed pump until the variable speed is increased sufficiently high to meet the pressure created by the constant-speed pump. One unexpected result is that as the flow and pressure in the system decreases, the flow in the constant-speed pump increases and the operating point moves steadily down the pump curve. This can result in the constant-speed pump operating beyond the end of its published curve with resultant increase in radial thrust forces and potential cavitation.

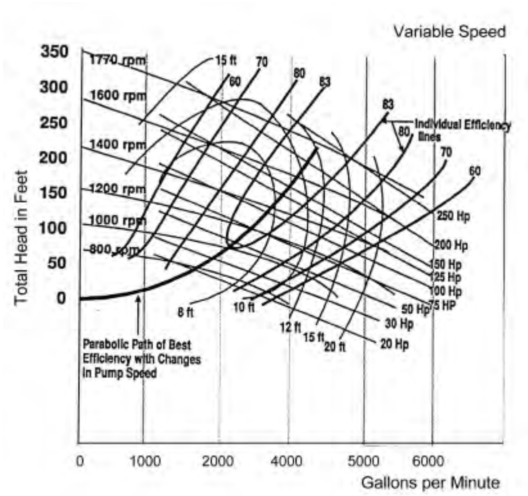


Figure 3-35: Variable-Speed Performance Curve

Title 24-2005 requires variable flow design for all chilled water systems with more than 3 control valves. It also requires variable speed drives on all variable flow systems with pump motors greater than 5hp.

Selecting Pumps

In general, a constant-volume pump should be selected +25% of BEP and a pump with a variable-speed drive should be selected to the right of the BEP. Selecting a pump too close to shut-off head or too near the end of the curve presents problems with radial thrust and potential cavitation. This will be discussed further below. Chilled water pumps serving terminals with two-way valves (variable flow) generally should be flat-curved pumps, and condenser water pumps should be steep-curved pumps, if constant speed. Motor size should be selected so that the power curve does not cross the pump curve at any point (non-overloading).

When applying pumps, the actual pump head (as measured in the field) is often different from the calculated head used to select the pump. For constant-flow pumps with the actual head lower than the calculated head, trimming the impeller to match the actual requirements is energy efficient and cost effective.

System Curves

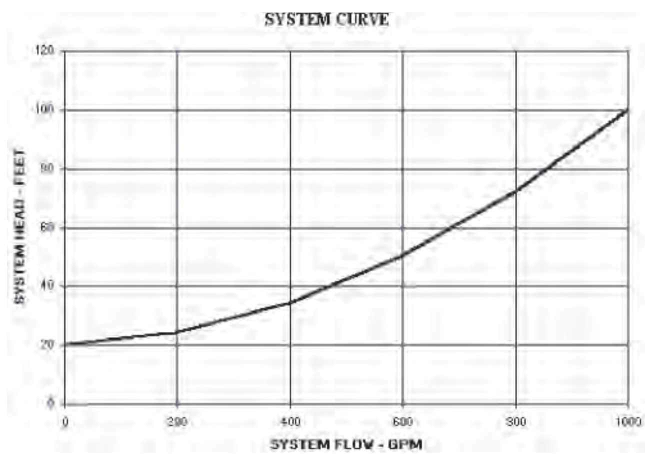
The affinity laws govern the performance of a pump under varying conditions of flow and head pressure. The most important law is:

Equation 3-2

$$\frac{Q_1^2}{Q_2^2} = \frac{H_1}{H_2}$$

Simply stated, the head pressure varies to the square of the flow. The head pressure and flow characteristics of a system can be predicted using the affinity law by plotting a “system curve.” A subtlety is that the system curve is actually a representation not of the pump performance, i.e., affinity law, but of pipe friction which follows slightly different formulas. Many times the actual velocity in chilled water piping system causes less than fully developed turbulent flow. Using 1.85 instead of 2 in the exponent is more accurate. In open systems (cooling towers) the static head is a constant, as is the head pressure in a variable-flow system that has a constant element, such as the minimum pressure maintained at one point in the system by a differential pressure controller. These constant pressures are represented by raising the starting point of the curve at the zero flow line to the pressure that remains constant.

Figure 3-36:
System Curve



The system curve can be used to predict the flow characteristics at a variety of points in a system, and are useful in determining corrective actions that need to be taken should the actual flow and head not match the design condition. For example, the system curve can be used to size a new impeller for a pump when field-testing reveals that the actual flow is greater than the design flow. Another example of using system curves is to predict the performance of parallel pumps. In one case turning on the second pump produces almost no additional flow. In another example turning the second pump off results in a flow from the first pump that is at the end of the published pump curve.

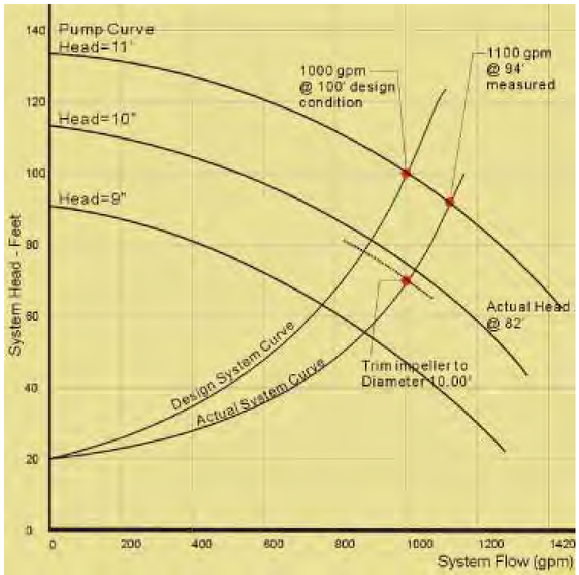


Figure 3-37:
System Curve (for Sizing New Impeller)

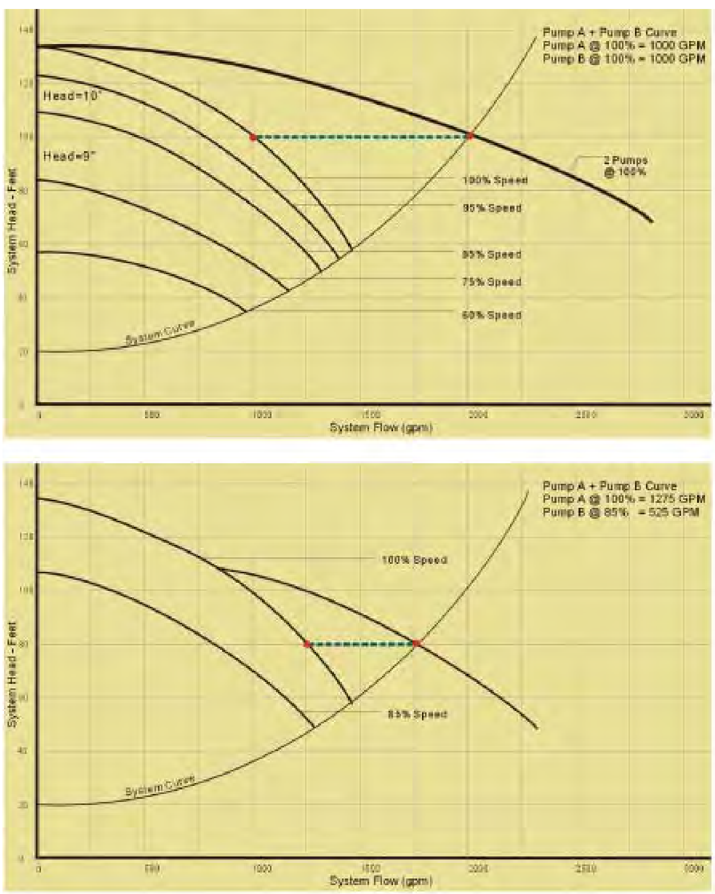
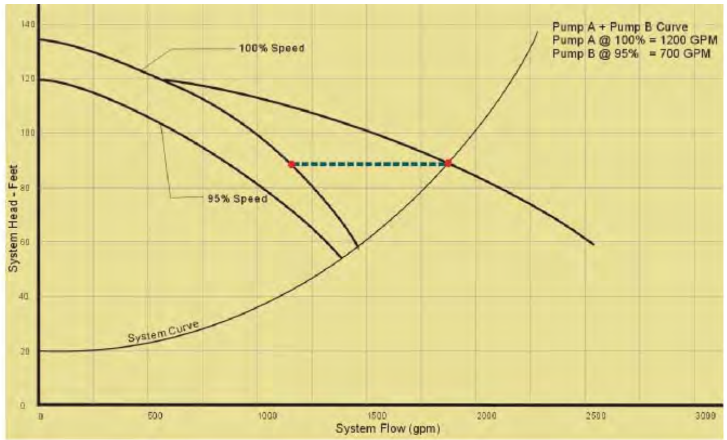
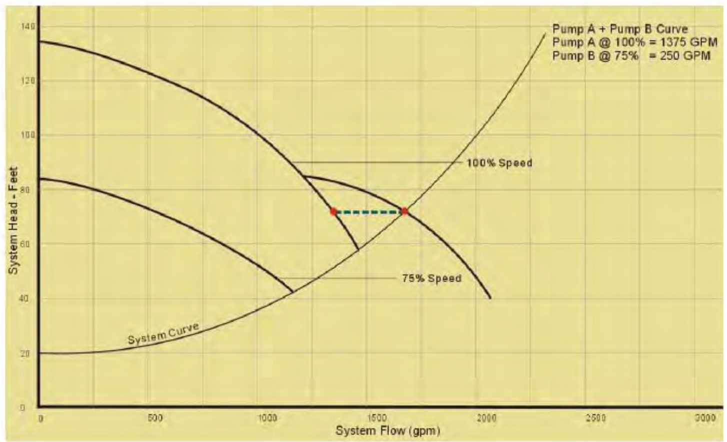


Figure 3-38:
Pump Curves

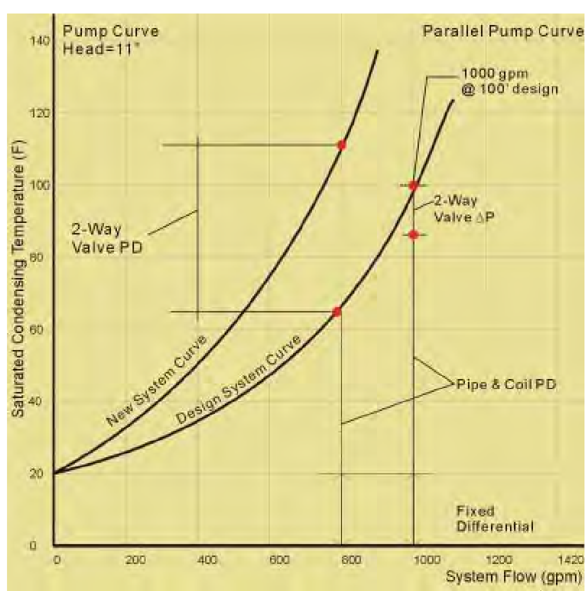
Figure 3-38: (cont'd)
Pump Curves





The system curve is accurate so long as nothing in the system changes. When two-way valves are incorporated into the system, the variable pressure drop created changes the system curve. This is important to understand because in a variable flow system with multiple two-way valves, there are many system curves that may represent the condition at various times.

Figure 3-39:
System Curve with Two-Way Valves Incorporated



Depending if the active loads in a piping system are near to or far from the pump, a range of system curves best describes the conditions that may occur at any given time. As the flow decreases, the velocities of the water in the pipe can develop laminar flow or a mixed laminar-turbulent flow. This has the affect of decreasing the exponent in the system curve equation above and is another reason to expect a range of system curves in an operating system.

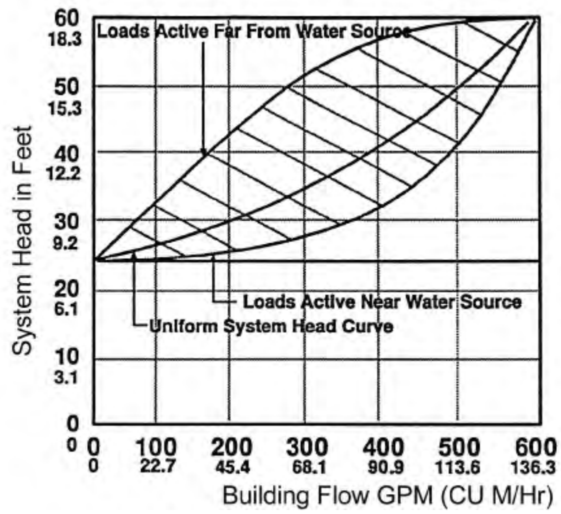


Figure 3-40:
Range of System Curves

Pump Inlet Limitations

The boiling temperature of water is a function of the absolute pressure surrounding the water. In a pump the pressure at the eye of the impeller can be the lowest in the system and depending on the temperature, the water can boil (vaporize). As the liquid moves through the impeller and gains pressure, the water vapor collapses back into liquid. This process is called "cavitation" and can be very harmful to the impeller. The pressure at the inlet of the pump must be high enough to prevent the water from boiling. The higher the velocity of water into the eye of the impeller, the lower will be the pressure and the more likely cavitation will occur.

Manufacturers publish "net positive suction head required" (NPSHR) curves with the pump curve. Notice that the NPSHR increases dramatically when the flow gets higher. The designer must ensure that the system will have enough "net positive suction head available" (NPSHA) to prevent cavitation. In closed systems with minimum inlet pressure control (usually not less than 12 psig), cavitation is rarely a problem. In open systems, i.e., cooling towers, cavitation is a very real concern and all efforts must be made to ensure that the NPSHA is greater than the NPSHR. This usually means that the sump levels are substantially higher than the pump inlet (4 to 6 feet minimum) and that the pressure drop of the suction line from the tower to the pump is not excessive. In many installations, the strainer in the condenser water system is located on the discharge side of the pump to avoid excessive pressure drop.

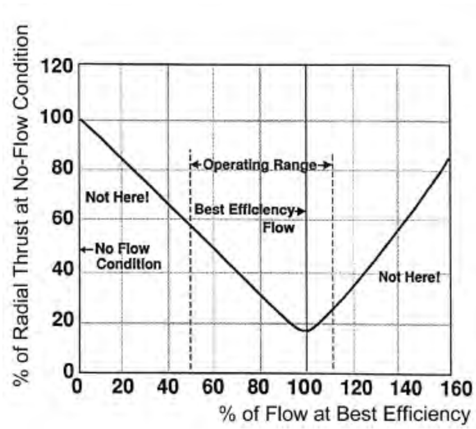
Another pump inlet problem to be avoided is "vortexing" or air entrainment. Any time water is drawn from an open tank or sump there is a potential that a vortex will occur. Vortexing will cause air to enter the pump suction line and will decrease the effectiveness of the pump. Vortexing will occur even with very deep sumps. Any time water is drawn from a sump or open tank, anti-vortexing devices should be installed.

When using axial pumps (i.e., vertical turbines), care must be taken to assure that manufacturer recommendations are followed to maintain a minimum submergence distance above the inlet bell. Also, adequate clearance must be maintained from the tank's bottom to the pump's inlet.

Radial Thrust

When pumps operate at points on the pump curve other than BEP, non-uniform pressures can develop on the impeller. This is called radial thrust and can cause severe shaft deflection, excessive wear on pump bearings and even shaft failure. Radial thrust occurs when pumps are operated at or near the shut-off pressures or near the end of the curve.

Figure 3-41:
Radial Thrust



Pump Installation and Operation

The following are installation and operation guidelines that will help ensure proper operation of pumps:

- The minimum flow through a pump should be sufficient to remove the heat of compression (motor input power) with no more than a 10°F temperature rise.
- Careful attention must be paid to the conditions at the pump inlet; a minimum of 4 to 6 diameters of straight pipe upstream or a "suction diffuser" is recommended for field-installed pumps.
- Variable-flow pumps should never have balance valves installed on the discharge, as flow balancing can be easily accomplished by varying the speed of the pump.
- When using a combination duty balance and check valve, install an additional shut-off valve downstream so that the check valve can be maintained.

Variable-Speed Drives

One of the greatest improvements in the design of chiller plants is the result of the variable-speed drive (VSD). The advent of a cost effective means to vary the speed of chiller rotors, impellers, and pump impellers has meant that greater operating efficiencies are now possible, and systems are inherently self-balancing with lower maintenance. The adjustable-frequency drive (AFD) is the electronic device that gives us the ability to vary the speed of the motors that drive the equipment. This device works by converting a fixed 3-phase voltage and 60 Hz frequency source into a variable voltage and frequency source. Frequency of the source to the motor controls the speed. In order to keep the required torque of the motor, the voltage and frequency relationship must be maintained. This is called Volts to Hertz relationship. By maintaining a constant Volts to Hertz relationship, the motor can develop full torque at all speeds, except at very low speeds (0 to 20 HZ). Most HVAC applications require variable torque because as the speed of the motor decreases, the load (torque) also decreases. This requires that the drives have a variable Volts to Hertz relationship.

There are three different types of VSDs currently on the market. These include VVI (Variable Voltage Input), CSI (Current Source Inverter), and PWM (Pulse Width Modulation). The pros and cons for each type of drive are too numerous to discuss in this guide. The majority of the drives currently in HVAC system use are PWM. The PWM drive has a fixed diode rectifier that converts the AC input voltage to fixed DC voltage. The DC voltage is filtered and sent to the Inverter section that changes the fixed DC voltage to variable AC voltage and changes fixed frequency to variable frequency. The Inverter uses power transistors to chop the DC voltage to create the variable output. The transistors are turned on and off at a variable rate (carrier frequency) to create the variable output voltage and frequency. PWM variable-speed drives have very high efficiency with little motor heating, have constant input power factor, run at low speeds, and have reduced audible motor noise (because of high carrier frequency).

VSDs have obvious benefits, particularly in improved energy efficiency, but do have some disadvantages, including a possible negative impact on power quality, motor noise, electromagnetic interference (EMI), radio frequency interference (RFI), and nuisance tripping. In applying VSDs, one must consider the efficiency of the motor and drive combined, the type of motor being connected, the distance of the drive to the motor, and numerous accessories.

Manufacturer Data

The variable speed drive manufacturers are continually modifying their product offerings. The reader is encouraged to browse the manufacturers' websites for the most current information on technologies and product ratings. The major manufacturer websites are:

- ABB: http://www.abb-drives.com/
- Danfoss (formerly Graham): http://www.namcdanfoss.com/products/index.html
- Hitachi: http://www.hitachi.us/Apps/hitachicom/content.jsp?page=Inverters/ACVariableSpeedDrives/index.html&level=2&section=Inverters&parent=ACVariableSpeedDrives&nav=left&path=jsp/hitachi/forbus/powerequipmentsystems/&nId=iD
- Saftronics: http://www.saftronics.com/
- Square D: http://www.squared.com/us/products/drives.nsf/unid/6A62B823134D194685256A1F005F1E13/$file/ACDrivesFrameset.htm
- Toshiba: http://www.toshiba.com/ind/

Power Quality Issues

Harmonic Distortion

Because the drives are based on static switches (power transistors), they represent a non-linear load to the electric supply. The switches cause distortion to occur in the other loads connected to the same supply. This is called Harmonic Distortion. When a load designed to expect a smooth sinusoidal voltage receives a distorted voltage, the result can be overheating of wiring, motors, and transformers, or malfunction of the equipment. The distortion of the voltage combines with the system impedance frequency response characteristic with the result of a harmonic distortion. The harmonic voltages and currents can cause spurious operation of relays and controls, capacitor failures, motor and transformer overheating, and increased power system losses. The problems can be compounded by the application of power factor correction capacitors that can create resonance conditions that magnify the harmonic distortion levels.

Harmonic distortion from an individual VSD is seldom critical for a distribution network, but the problem can be severe if multiple non-linear loads are involved. PWM drives have much lower harmonic distortion than other types of drives. Harmonic distortion can be avoided with proper cabling and grounding of the VSD and motor. Refer to manufacturer instructions. Line reactors or isolation transformers can reduce harmonic distortion in the other loads connected to the same supply. The best approach (and most expensive) for reducing harmonics is the tuned harmonic trap. These filters absorb practically all of the harmonic currents generated by the drive. The filter must be sized properly. The proper sizing of the filter requires a special study of the electrical distribution system to which the variable-speed drives are connected. Most modern drives can meet the project harmonic distortion requirements using a 3% input line reactor and high carrier frequency. The local utility companies often have minimum requirements for VSD design as part of their incentive programs.

EMI and RFI

All VSDs produce electromagnetic emissions to some degree. EMI is similar to a radio wave. If the EMI signal is strong enough, it will cause unwanted reference signals or “noise” in other electronic equipment. The easiest way to correct problems associated with EMI is proper routing of the drive conductors in separate metallic conduits, and even separate raceways, if practical, and as remote as possible from any other conductors or suspect equipment. RFI can also cause operating problems in other electronic equipment, but the effects are more profound in equipment that is not properly grounded or in inductive devices like solenoids that do not have noise suppressors. To contain RFI through the media from the VSD, complete shielding using a metallic enclosure is required. This will contain most of the radiated RF to a reasonable distance. EM/RF filters, which are expensive, can be engineered for a system to trap or inhibit high frequency emissions into power system conductors; however, the effectiveness of any filter is sensitive to where it is located.

Drive System Efficiency

The efficiency of the drive system takes into account electrical losses from the variable-speed drive, the connected motor, and the combination of the two devices. The VSD has losses in the form of thermal power from the inverter (60%), rectifier (30%), and leaking current and power lost in the cooling equipment (10%). The inverter and rectifier losses are proportional to the motor shaft speed and the other losses are fairly constant. Drive losses are 2 to 2.5% of the nominal power of the drive. The motor losses are from rotating losses, including friction and iron losses, and resistance losses caused by the resistance in the internal wiring in both the stator and the rotor. When a motor is married to a VSD, there is a 10 to 15% increase in the motor rotating losses, and the same in the resistance losses. The efficiency at zero speed is always zero.

The actual efficiency of the drive and motor operating together can be calculated if both the motor and drive losses are known in the entire speed range. The typical performance of the drive, motor and pump is shown in Figure 3-42. This figure takes into account the motor and drive losses and the decrease in power requirements as the pump speed is decreased.

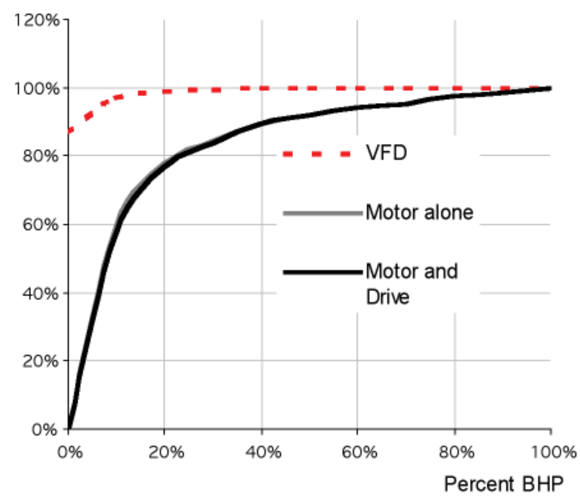


Figure 3-42: Typical Performance, Drive and Motor

VSD Accessories and Considerations

Bypass Switch

A VSD can be equipped with a manual or automatic bypass switch. Normally this is provided to allow the operation of the motor across the line (with no speed control) while the drive is being serviced. This requires two sets of contacts, one on the power side and one on the load side to isolate the drive for servicing. Care must be taken when operating the motor across the line. Larger motors (60 HP and above) are usually not started across the line due to limitations on inrush current. When operating in the bypass mode, the system may be over-pressurized causing failures in piping and control valves.

Shaft Grounding

VSDs have been shown to create voltage differences (as high as 60 volts) between the rotating shaft and the casing of the motor. Electrical current passes from the shaft through the bearings, gearboxes, tachometers, etc. into the casing. This electrical current can cause damage in the form of pitting of the bearing cases. A pattern known as “fluting” can form that will eventually cause premature failure of the bearings. The IGBT transistors in the VSD are a major source of the voltage difference. Shaft ground kits are available that are installed on the shaft, typically on the end of the motor.

Disconnect Interlock

If the motor served by a VSD is equipped with a separate disconnect switch located at the motor, the disconnect should be equipped with an auxiliary contact switch to signal the VSD that the motor has been manually turned off from a remote location. This auxiliary contact protects the VSD from accidentally starting under full-load conditions that can cause catastrophic failure of the drive. Though newer VFDs provide "Load-side Switching" safety cutouts, providing this disconnect switch with auxiliary contacts is advised.

Motor and Drive Matching

Some manufacturers insist that their VSD be carefully matched to a special motor (usually of the same manufacturer as the drive). This is not necessary. Most VSDs can serve any induction motor. The exception can be when applying a new VSD to an existing motor. In some older motors, Class B insulation windings may not be sufficient to handle the voltage surges and additional overheating caused by the VSD. In these cases the motors should be replaced with high efficiency motors or the existing motors can have a megger test that will verify the condition of the insulation on the windings. Contact the VFD supplier,: ABB, for example, maintains a database of existing motors and the expected failure rate when applying a VFD. Inverter rated motors or motors with Class F insulation permit motor operation at high winding temperatures and should be specified when used with VFDs. Newer model VFDs produce a truer sinusoidal waveform and motor overheating and failures problems are not as prevalent as they once were.

Motor heat is affected by the wire distance between the motor and the VFD. When retrofitting to an existing motor, limiting wiring lengths (each phase) to a maximum of 100 feet is a good rule. Where inverter rated motors are installed, wire lengths can be extended to 300 to 400 feet without a problem.

Critical Frequency Lockouts

When the speed of a cooling tower fan is varied, there are certain speeds in which a resonance frequency vibration can occur. he VSD can be programmed to lock out these frequencies to protect the fan. The cooling tower manufacturer should be contacted to verify the speeds at which resonance frequency vibrations occur.

4. HYDRONIC DISTRIBUTION SYSTEMS

This chapter of the CoolTools Design Guide addresses piping layouts and design issues related to chilled water distribution systems and condenser water systems. The first section addresses the chilled water (evaporator) side of the chillers, chilled water pumps and cooling coils. The second section addresses the condenser waterside of the chillers, including cooling towers, condenser water pumps, water economizers and other design issues.

Chilled Water Systems

Introduction

The chilled water distribution system consists of chillers, pumps, piping, cooling coils, controls and other components on the evaporator side of the chillers. This dynamic system, which provides cooling for many air conditioning applications, is one of the most energy intensive systems in commercial buildings. Understanding how hydronic distribution systems react to varying loads and how their components interact is essential for designing an energy-efficient and cost-effective chilled water plant.

Older chilled water plant designs circulate a constant volume of chilled water through the chiller(s) and the building, no matter if the cooling load is large or small. If loads are small, the constant volume of chilled water is diverted around the cooling coils by three-way valves. In multiple chiller systems, it is difficult to shut off the machines that are not needed. There are other problems as well with constant volume designs. As a result, variable flow systems have emerged along with a different set of engineering challenges.

This section—Chilled Water Systems—discusses appropriate applications for constant- and variable-flow chilled water systems and presents design strategies for achieving energy efficiency and operational simplicity. A brief outline follows:

- Constant-Flow Chilled Water Systems
    - Single chiller serving single or multiple cooling loads
    - Multiple chillers (in parallel or series) serving multiple cooling loads
- Variable-Flow Chilled Water Systems
    - Concerns about variable flow in evaporators
    - Primary-only variable flow design
    - Primary/secondary variable flow design
    - Distributed systems in larger plants
    - Coil pumping strategies

- Variable-Flow System Design Considerations
    - Primary pump configuration
    - The balance valve debate
    - Causes and effects of low delta-T syndrome
    - Design solutions for low delta-T syndrome
    - Techniques for connecting multiple chiller plants
    - Methods of connecting heat recovery chillers

The second section of this chapter addresses the design of condenser water systems. Optimizing the size and selection of the chillers, towers and other components, and the sequencing of the chiller plant are discussed in Chapter 6, "Optimizing Design."

Constant-Flow Chilled Water Systems

This subsection addresses how constant-flow systems can be used in the following applications:

- Single chiller serving a single cooling load
- Single chiller with multiple cooling loads
- Multiple parallel chillers with multiple cooling loads
- Multiple series chillers with multiple loads

Single Chiller Serving a Single Cooling Load

With a single chiller serving a single cooling coil (Figure 4-1), the simplest design strategy is to eliminate the traditional three-way control valve at the coil and to use a constant-volume pump to circulate water between the evaporator and the coil. Control is provided by resetting the temperature of the chilled water leaving the chiller. Constant water flow provides reliable heat transfer at both the evaporator and the cooling coil. Also, chiller performance is improved when the leaving chilled water temperature is reset to be as high as possible, subject to the limitation of minimum refrigerant head pressure differential between the evaporator and condenser as discussed in Chapter 3. When dehumidification is required at low loads, the temperature of the air leaving the coil can be set to achieve the necessary dew point temperature.

Chillers must have a sufficient volume of water in the piping system to prevent unstable temperature swings and this may be an issue with single-chiller, single-coil systems. Often a small storage tank is required if the chiller is closely coupled to the coil. The minimum water volume should be verified with the chiller manufacturer, but some general guidelines follow:

- Provide 2.4 gallons/ton for a screw compressor.
- Provide at least a 5-minute re-circulation rate for reciprocating or scroll compressors.

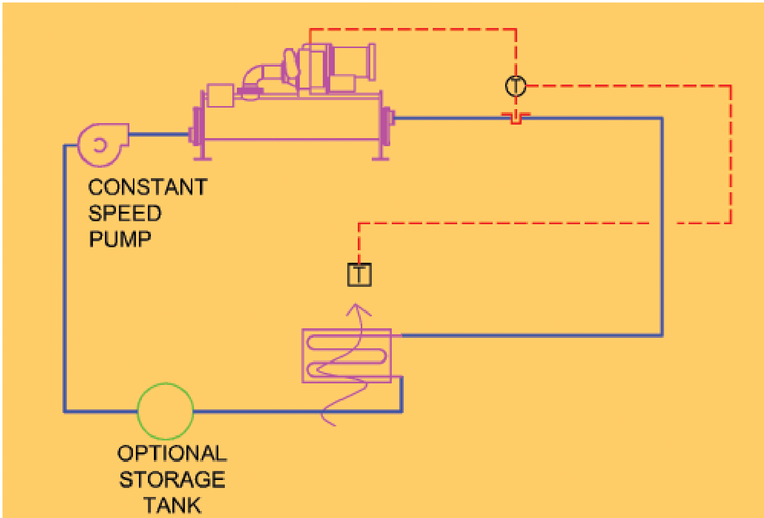


Figure 4-1:
Constant-Flow System,
Single Chiller, Single Coil

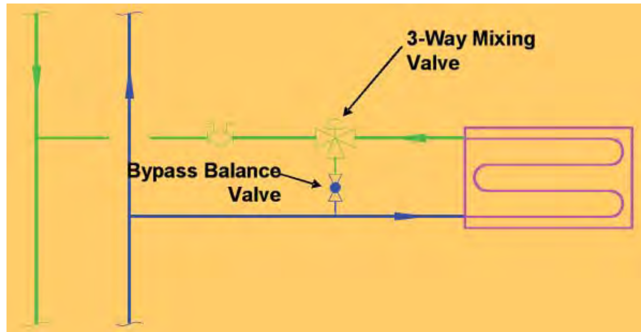
Single Chiller with Multiple Cooling Loads

A "constant-flow" chilled water system is a simple, cost-effective design for plants with a single chiller and multiple cooling coils (Figure 4-2) as long as the pumping head is not too great (less than ~50 feet). A variable-flow system may be more cost effective if either the piping between the chiller and the coils is long or the coil head is large: both cases resulting in high pumping head (over ~50 feet). In a "constant-flow" system, three-way valves are used at the cooling coils to modulate the load at each air handler. An energy-saving control strategy is to reset the chiller's leaving water temperature to satisfy the coil requiring the coldest temperature.

Title 24 requires variable flow design for systems with more than 3 chilled water valves. Under this requirement not all valves are required to be two-way, but current best practice with modern chillers is to provide 2-way valves on all coils with variable flow systems. Two-way valves save energy and first cost.

The use of 3-way valves does not actually provide a constant-flow system. This is demonstrated in Example 4-1 below. In Example 4-1, the first four rows of the table show the system pressure drop by component at a fixed flow rate of 100 gpm to the branch circuit. The resulting pressure drop at the point of connection to the branch circuit varies from 20' of head at both the full (100%) flow and no (0%) flow conditions. The bypass balance valve is provided for this reason. At the half-load (50%) flow condition the pressure drops to 11.5' of head. In the bottom row of the table we see what the flow is through the branch if the system pressure is held at 20' of head at the branch circuit point of connection. At both the full (100%) flow and no (0%) flow conditions the flow is 100 gpm (design). At the half-load (50%) flow condition the flow increases to 132 gpm, 32% over design flow. At part load conditions, systems with 3-way valves can experience staved coils at the most remote parts of the system. The solution is to provide a few 2-way valves on the system. Experience has shown that 2-way valves should be installed on coils representing approximately 20% of the design flow. Locating these 2-way valves close to the pumps ensures adequate flow for the remaining valves under part-load conditions.

Example 4-1:
Flow Variation as a
Function of Valve Position

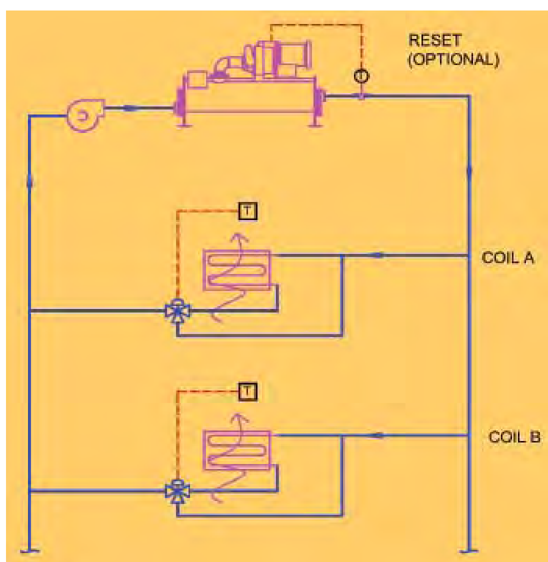


	Pressure Drop @ 100 GPM		
Item	**100% to Coil**	**50% to Coil**	**0% to Coil**
Pipe/Valves	2.0	2.0	2.0
Coil Bypass	8.0	2.0	6.0
Control Valve	10.0	7.5	12.0
Total	20.0	11.5	20.0
GPM @ 20' ΔP*	100	132	100

*Actual ΔP available may change

Using constant-flow regulating valves at each coil is another—although more expensive—way to maintain system balance. This is not recommended as described below under the system balancing discussion.

Figure 4-2:
Constant-Flow System,
Single Chiller,
Multiple Coils



Multiple Parallel Chillers with Multiple Cooling Loads

Constant-flow systems can be piped with multiple chillers in either a parallel (Figure 4-3) or series (Figure 4-4) configuration. Since staging with constant-flow systems can be dictated by flow rather than load, these systems should be limited to coils that are serving loads with the similar part-load characteristics. When the system operates near full load, performance is satisfactory as all chillers and pumps are operating. However, constant-flow systems have problems during part-load or off-peak conditions. Consider for example a system like that shown in Figure 4-4 with two equally sized chillers that serves two equally sized coils, each coil in turn serving a hotel meeting room. If there are functions in both rooms (i.e., both rooms are at or near full-load) the system operates well: both chillers with their associated pumps are running and each function space is receiving its design flow. Now consider that only one of the two function spaces is occupied (say Coil A in Figure 4-3) and the other (Coil B) is vacant. Coil A still needs its design flow, so in theory one chiller with its pump could satisfy it. Coil B will also take its design flow, although it will merely bypass this flow from the supply to the return. If the plant operates with only one chiller and pump, it has sufficient load capacity but it cannot meet the flow demands. One half of the water will flow through Coil A (which is less than it needs to meet the load), and the other half of the water will flow through Coil B. Since Coil A will be starved, both chillers will have to operate at 50% load to satisfy the load at Coil A. This problem of flow dominated staging is solved by variable flow design.

If both Coil A and Coil B are serving similar loads (for instance, different floors on an office building), the story is different. Similar to the previous example, both chillers and pumps run when the loads are at or near full load. In this case each coil needs and receives nearly the full load and flow. When the loads drop below 50% on both coils, one chiller and pump should be sufficient to satisfy both coils. Both coils will receive the same reduced flow and the valves should be at or near 100% open. Since water is not bypassed, nearly 100% of the chiller's capacity is used by the coils and the loads are satisfied in an efficient manner. Note that this scenario is only true if the coil loads are similar; if the coils serve different internal loads or facades, the problems of our previous example are likely to arise.

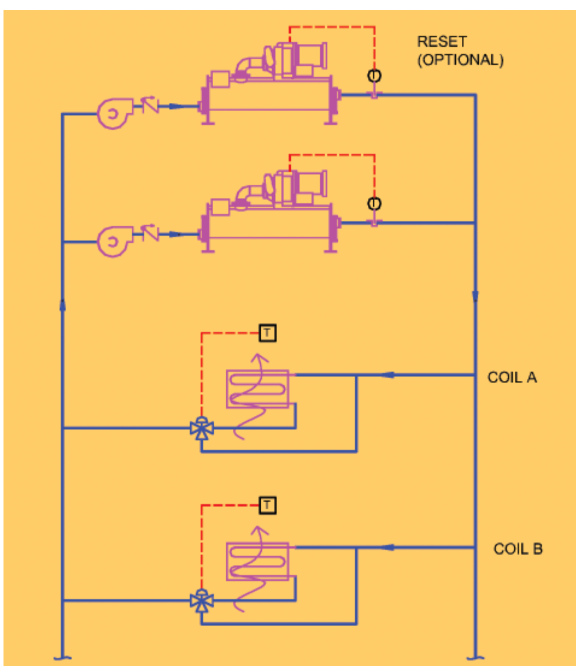


Figure 4-3:
Constant-Flow System,
Multiple Parallel Chillers,
Multiple Coils

Multiple Series Chillers with Multiple Load

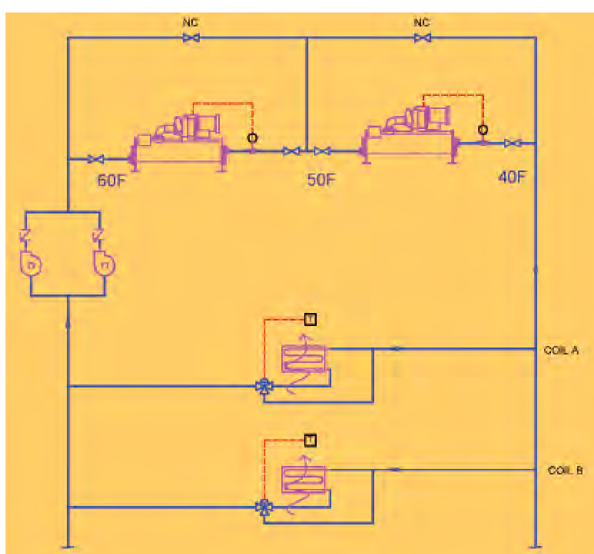
One solution to using a constant-flow system with multiple chillers is to put the chillers in series. In this configuration, the entire flow passes through each machine. This method is effective for systems designed with a high delta-T (15 to 20°F). During off-peak periods, the *lag machine* (second in series) is turned off and the *lead machine* (first in series) continues to deliver chilled water at the correct temperature.

Although this system works well, chilled water pump energy savings are not realized during periods of low load. This is partially offset by the lower flows required with high delta-T designs. Series chillers also become very cumbersome when the plant has more than two chillers. One problem is that the pumping head (water pressure drop) through series chillers can be excessive. The pump head through the evaporator can be minimized by the judicious selection of the chiller and/or the number of passes through the evaporator can be reduced from two (or three) to one with some reduction in the downstream chiller efficiency.

Another solution that solves the high pressure drop problem (although more expensive) is to pipe the chiller to the distribution loop in a primary/secondary fashion as shown in Figure 4-4. This design also allows more than two chillers to be used in series. This piping arrangement is very effective for systems designed for high delta-T (over 20°F) between the entering and leaving chilled water temperatures.

One application for piping chillers in a series primary/secondary fashion is where an absorption or engine-driven chiller is coupled with an electric chiller. The arrangement allows the operator flexibility to choose which machine to load based on utility rates or other criteria. In addition, allowing absorption machines the opportunity to operate at higher inlet and outlet temperatures (in a series configuration) increases the energy efficiency of these types of machines. The series configuration also allows the chillers to be unequally loaded (for example if you wanted to preferentially load the thermal machine during on-peak times). For this to work the preferred chiller would be controlled to maintain its leaving supply water temperature at the desired plant temperature and the other chiller would be reset to maintain the plant leaving supply water temperature at the same setpoint. With this control, either the upstream or downsteam chiller can be preferentially loaded.

FIGURE 4-4:
CONSTANT-FLOW SYSTEM, MULTIPLE SERIES CHILLERS, MULTIPLE COILS



Variable-Flow Chilled Water Systems

Introduction

Variable flow has many advantages in large chilled water systems with multiple chillers and multiple loads or coils. Significant pumping energy can be saved because the plant can effectively modulate during periods of low load. This section discusses many important aspects of variable-flow chilled water systems, including:

- the effect of variable flow through the evaporator;
- energy efficiency opportunities with distributed pumping techniques in both large and small central plants;
- how (and when) to balance variable flow systems;
- the causes and effects of low delta-T syndrome; and
- techniques for interconnecting multiple central chiller plants for added energy efficiency and redundancy.

Variable Flow in the Evaporator of a Chiller

Flow in the evaporator can be dynamically varied but not without some risk. If a chiller is operating in a stable condition and flow in the evaporator is reduced, the leaving chilled water temperature will drop. If the flow reduction occurs slowly, the controls will have adequate time to respond and the system will remain stable. But a rapid change in flow will cause the leaving water temperature to drop quickly. If the controls react too slowly the chiller may shut down on low temperature safety. This is a significant nuisance since someone must manually reset the safety control and the chiller must remain off for a minimum period of time before restarting. Some manufacturers (although not all) have adopted modern controls that account for the rate at which the leaving chilled water temperature drops. These controls will prevent inadvertent shutdown of the chiller.

Another issue is avoiding laminar flow in the evaporator tube. A fluid velocity of at least 3 feet per second is recommended to maintain good heat transfer. In chilled water plants with higher delta-Ts (lower flow rates), the variation between the design flow and the minimum flow may be limited. For example, on a system with a two-pass evaporator and a 12°F delta-T, the minimum flow could vary down to about 50% of design, and with a three-pass evaporator could vary down to about 30% of design. Given the fluctuations and accuracy of controls, a good designer will choose a minimum flow rate that is not too close to the published minimum. Consult the manufacturer's literature for maximum and minimum flow rates.

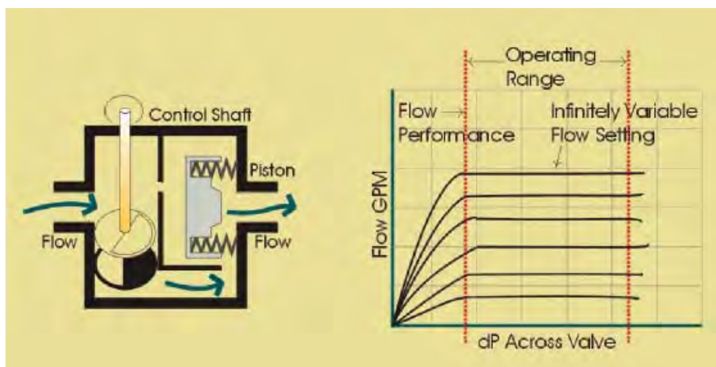
Because of these risks, designers traditionally chose constant flow rates through the evaporators. However, due to the operational advantages (discussed below) coupled with the success of the new chiller internal control strategies, variable flow through the evaporator is rapidly becoming standard practice.

Primary-Only Variable Flow Design

Primary-only variable flow systems have many advantages over common primary/secondary designs. Primary-only variable flow systems consist of single or multiple chillers with system pumps that move the water through the chillers and distribution system to the cooling load. The cooling output at each coil is controlled with two-way valves. A bypass line with control valve diverts water from the supply into the return piping to maintain a minimum (or constant) flow through the chiller(s). The simplicity of this approach makes it attractive, but it has some drawbacks. For instance:

- The bypass valve (when located near the pumps) can act against a relatively high-pressure differential, with the result that it is very susceptible to wear, cavitation, and unstable operation at low loads. Note this is probably not a problem on systems with variable speed drives on the primary pumps as they will back off on the pressure across the bypass when the system is in low demand conditions where the bypass must operate.
- If the bypass valve maintains a constant flow through the chiller(s), there will be no pump energy saved as the loads vary (except when pumps are shut off as chillers are disabled).

Figure 4-5:
Delta-P Pressure-Independent Control Valve



Variable-speed drives can be added to the primary pumps so that as demand goes from maximum to minimum, the speed can be adjusted downward, thus saving pump energy. In this case, the bypass valve should be sized to operate at a low flow condition with the coils unloaded and less than the full pump design pressure across the valve. The pump bypass valve is a very good application for a pressure independent control valve such as the one presented in Figure 4-5. These valves (sold by Griswald, Delta-P and Belimo) are highly recommended on systems with constant speed pumps. They are very expensive but have excellent control authority over a wide range of operating pressures.

- ***Single chiller with multiple coils*** (Figure 4-6). Single chiller systems are usually constant flow, except for conditions with a large pumping head (over 50 ft). In a single chiller system with two-way valves on each coil, the demand for flow will decrease as the load decreases. Controlling the bypass valve between the supply and return mains for a constant flow at the chiller has the same result as using three-way valves at all the coils: there are no pump energy savings. Pump energy savings can be realized, however, if the bypass valve is controlled only to maintain the minimum flow rate recommended by the chiller manufacturer, which typically is 25% to 50% lower than the design flow rate. Today the most common approach is to use a variable-speed drive on the pump that is controlled from a remote differential pressure controller or the cooling coil valve position of the most open valve. In this configuration the bypass valve would be controlled to maintain the minimum flow through the chiller using a flow meter on the chiller side of the bypass line. In addition to pump energy savings, the variable-speed drive will also improve controllability at the coil valves and, perhaps more importantly, at the bypass valve by reducing the pressure differential across the valves. The two control loops (pump speed and bypass position) need to be tuned carefully to prevent hunting.

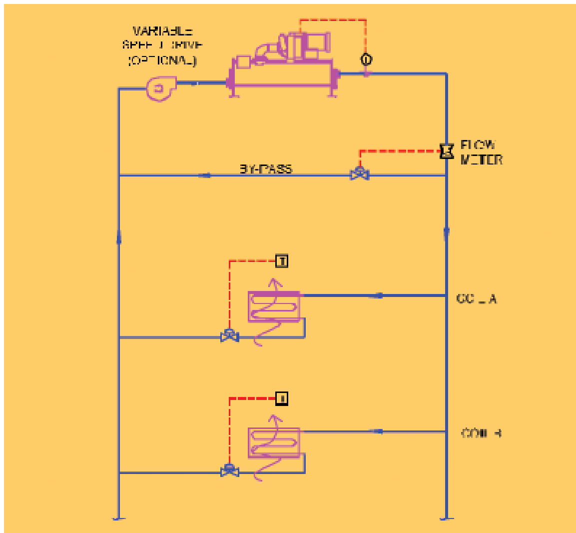


Figure 4-6:
Primary-Only Variable Flow Piping, Single Chiller, Multiple Coils

- ***Multiple chillers with multiple coils*** (Figure 4-7). A primary-only variable flow system is a good choice for multiple chillers that are coupled with multiple coils. When two-way valves at the coils modulate toward closed and the load and/or flow are sufficiently low, one or more of the chillers can be shut down. It is important to eliminate flow through the "down" chillers by either closing a two-position isolation valve at each chiller or by shutting off a dedicated pump (pump check valve prevents backflow).

  Control of the primary pumps and bypass valve are the same as discussed in the one chiller system with two added complexities:

  1. The bypass setpoint changes with the operating chiller(s). For a system with two equally sized chillers there are just two setpoints (one for a single chiller and one for both chillers in operation). For systems with unequally sized chillers, with more than two chillers or with different chillers, a look-up table is usually required to map minimum flows based on the sum of the minimums from all of the chillers that are operating.
  2. The staging of the pumps must also be considered (see Chapter 5, Controls).

- ***Other Issues***. The bypass valve can be located either adjacent to the chiller or far out in the system. Locating it close to the chillers provides the best energy performance since it reduces flow in the distribution piping, hence reducing pump energy. It also allows the chiller flow meter and the bypass valve control to be from the same control panel, ensuring that DDC system network delays or failures do not affect control of the bypass valve. On the other hand, a remote location results in more consistent pressure drop across the valve, which makes control more stable. For campus situations, a more remote location also ensures that the water in the distribution system is kept cold—for a large campus loop, the time required to cool down the mass of water in the system can be substantial. But keeping the loop cold can be accomplished by other means (e.g. a 3-way valve at the end of the system); because of the energy advantages of the close location, we generally recommend it.

The selection of the flow meter (see Chapter 5, Controls) must be accurate at the low end of the flow range since it is under low flow conditions that the bypass will be engaged. Since this meter (or meters) are used to protect the chiller, a high quality flow meter or meter that can be easily calibrated are recommended. We have had luck with both insertion dual-turbine (or magnetic) meters on the main chilled water line or individual full-bore magnetic meters on each chiller. In all cases the meter(s) must be on the chiller side of the bypass line to measure the combined flow through the bypass and coils.

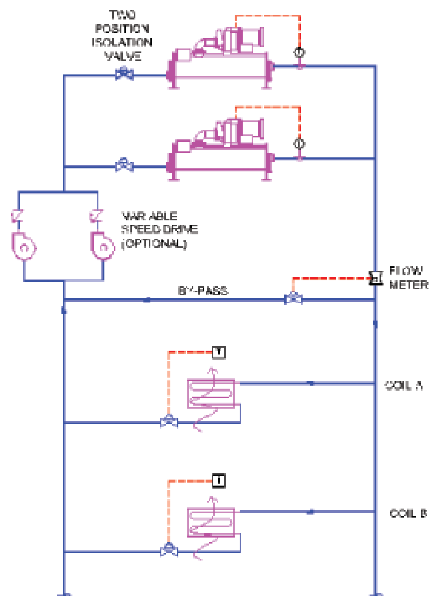


Figure 4-7: Primarily-Only Variable Flow Piping, Multiple Parallel Chillers, Multiple Coils

Primary/Secondary Variable Flow Design

Until recently, primary/secondary was the standard design for central chilled water plants with multiple chillers and multiple cooling loads. The beauty of the primary/secondary variable flow design is that the piping loop for chillers (the primary loop) is hydraulically independent (decoupled) from the piping loop for the system (the secondary loop). The key to this design is that two independent piping loops share a small section of piping called the “common pipe.”

When the primary and secondary pipe loops operate at the same flow rate, there is no flow in the common pipe. Depending on which loop has the greater flow rate, the flow direction in the common pipe is subject to change. Typically, the number and flow rates of the primary pumps match each chiller. The primary pumps are typically constant volume, low head pumps intended to provide a constant flow through the chiller’s evaporator. The secondary pumps deliver the chilled water from the common pipe to coils then back to the common pipe. These pumps are variable-speed pumps controlled from differential pressure sensors located remotely in the system or from cooling coil valve position.

Normally it is desirable to have the flow rate in the primary loop equal to or greater than the flow rate in the secondary loop. This means that some of the cold supply water is bypassed through the common pipe to the return side. The cold bypass water mixes with the return water from the secondary system, dropping the temperature accordingly. This water is then pumped back into the chiller. When the secondary flow exceeds the primary, return water from the system flows back through the common pipe and mixes with the supply water from the chillers. This increases the temperature of the supply water to the secondary system, sometimes with dire consequences. The warmer supply temperature causes the valves at each cooling coil to open even more, creating an ever-increasing demand for secondary system flow. This phenomenon (often referred to as the “death spiral”) is well documented in literature as it has plagued many large campuses. To address this problem, there are two options: 1) stage chillers by flow so that the primary flow is always equal to or greater than the secondary flow; or 2) insert a check valve in the common leg to put the primary and secondary pumps in series (thereby increasing the flow through the operating chillers).

If the secondary system return water temperature is lower than the design temperature, the chillers cannot be loaded to their maximum capacity. This is called “low delta-T syndrome” and it results in greater pump, chiller, and cooling tower energy consumption, as well as a reduction in cooling plant capacity. In most cases, the capacity control and control valve of the air handling units are the cause of low delta-T. (See the low delta-T syndrome section for a more detailed discussion of the causes and remedies.)

Staging a chiller by flow requires one or more flow meters that can be used for the controls. A single bidirectional flow meter can be placed in the common leg and the chillers will be staged as required to keep the flow in the common leg from the primary chilled water supply to the primary chilled water return. The difficulty in this scheme is that it is hard to know when to stage the chillers down again. A better scheme is to have flow meters on both the primary and secondary loops. Pump or chiller staging can be done on the primary side as required to keep the primary flow greater than the secondary flow.

A check valve in the common leg breaks the hydraulic independence of the primary and secondary loops. A low pressure drop “swing” check valve should be employed and oriented to prevent warm secondary chilled water return flowing back to the secondary chilled water supply. If the primary pumps are dedicated to the chillers (see discussion on Primary Pump Arrangements below), the isolation valves on the chillers must be automated to prevent “ghost” flows from occurring through chillers that are off. When the check valve seats (due to the secondary pumps out pumping the primary pumps), the chilled water return can get pushed through an inactive chiller and pump. The only way to prevent this is having an isolation valve closed. If the primary pumps are headered (see discussion on Primary Pump Arrangements below), it is theoretically possible to “deadhead” the secondary pumps if the secondary pumps are running when all of the chillers are off line. This latter problem can be prevented by logically interlocking the secondary pump operation to the status of one or more chiller.

Figure 4-8:
Primary/Secondary
Variable Flow Piping,
Multiple Parallel Chillers,
Multiple Coils

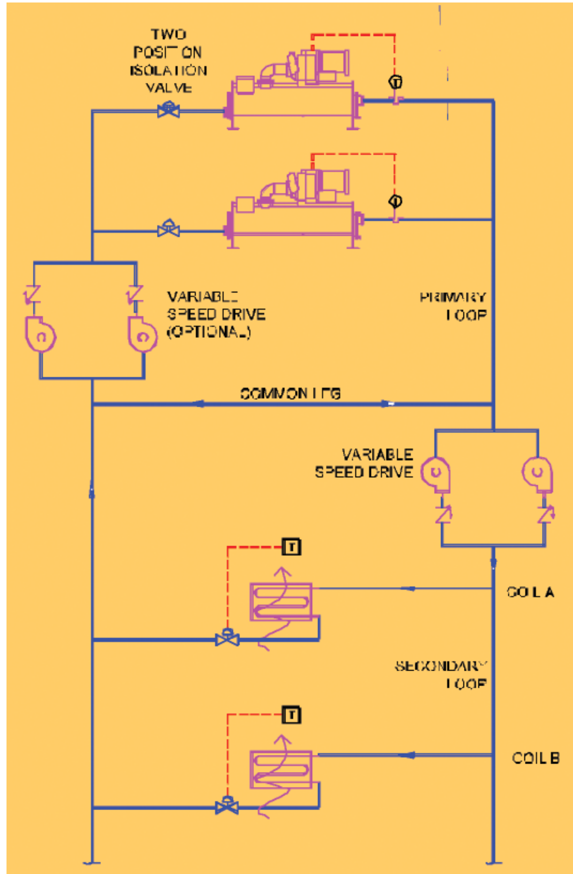
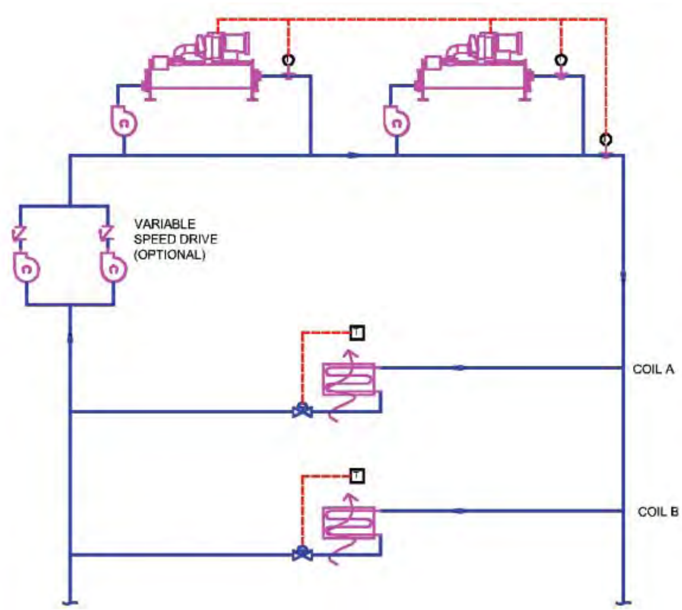


Figure 4-9:
Primary/Secondary
Variable Flow Piping,
Multiple Chillers, Multiple
Coils, Multiple Series
Chillers, Multiple Coils



Another issue in primary/secondary piping is the size of the common piping (bypass pipe). In a multiple chiller system that is properly controlled, the maximum flow in the bypass should not exceed 110% to 115% of the flow from one chiller. However, in most cases the bypass line should be the same size as the supply and return header. There are two reasons for this:

- The first cost of saving material by decreasing the size of the bypass line can be more than offset by the labor for the extra fittings required.
- By keeping the primary piping system's pressure drop as low as possible, flow variations are minimized when operating with one chiller instead of multiple chillers.

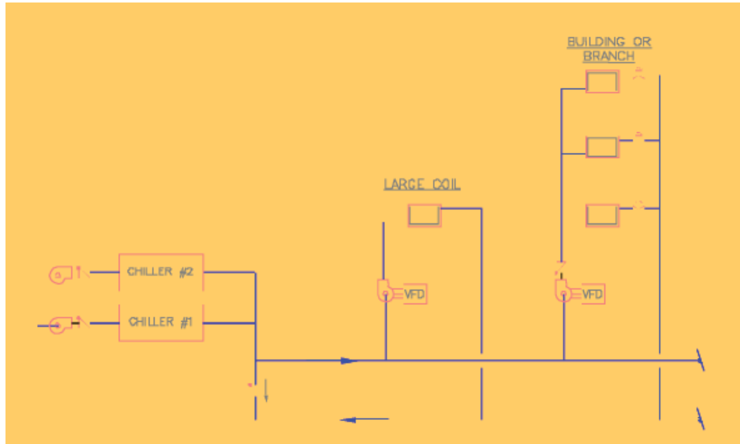
In very large systems (say 5,000 tons or pipes larger than 20 inches), the cost savings achieved by downsizing the common pipe should be considered.

Large Plant or Building Distribution Systems

- ***Distributed pumping system*** (Figure 4-10). The primary/secondary pumping arrangements described above have the secondary pumps located near the common piping (within the central plant) and serving all of the secondary system. While this strategy is reasonable for some smaller systems, it uses more energy and often costs more than a systems with distributed distribution pumps. Distributed distribution pumps as shown in Figure 4-10 should be considered as a design alternative for distribution systems with high pumping head in both a single building and in large campus-like central plants (for instance, those that serve hospitals, airports, and university campuses). In a traditional primary/secondary arrangement, pressure created by the secondary pumps must be sufficient to deliver the chilled water to the most remote load or coil. As a result, the coils located closest to the secondary pumps operate at high differential pressures. In a smaller system this may not present a problem, but in larger plants this over-pressurization not only represents energy waste but it strains the ability of the control valve to accurately modulate water flow and may even cause valves to lift off their seats.

  A distributed pumping system moves the secondary pumps from within the plant and locates them remotely nearer to the loads they serve. In a single large building these pumps could be located either at the coils or on distribution branches that serve a group of smaller loads. On a large system the distributed secondary pumps would be located in each building being served. In either case, the distributed secondary pumps are sized for the pressure drop needed to move the water from the common pipe at the plant to their most remote coil within the building and back to the common pipe. Significant pump energy and installed cost savings can be achieved with this design strategy. Because the secondary pumps are distributed, they need to be sized for the peak demand required in the building. This strategy may lose some of the benefit of diversity provided by centralizing the secondary pumps with the result that the pumps may be larger and more expensive.

Figure 4-10: Primary/Secondary Variable Flow Piping, Distributed Pumping



- Tertiary pumping system (Figure 4-11). A derivation of the standard primary/secondary pumping strategy is to provide standard secondary pumps within the central plant and locate tertiary pumps remotely within the building, at the loads. Traditionally, a crossover bridge with a two-way valve would be used at the connection point between the main distribution piping connections and the tertiary pump. The crossover bridge hydraulically isolates the tertiary pumping system in the building from the main secondary system. The two-way valve ensures that the secondary flow through the crossover bridge was equal to or less than the building flow. This valve is typically controlled to provide building chilled water supply temperatures 1°F warmer than the secondary loop supply temperature (to ensure that all of the secondary flow goes to the building).

  With modern variable-speed technology and control, however, the crossover bridge is not necessary. Without the crossover bridge the closest buildings can use available differential pressure, with the tertiary pumps simply supplementing this when needed (see 4-11). When using a tertiary pump without the crossover bridge, it is sometimes prudent to add a bypass valve or check valve across the tertiary pump so that during periods of high differential pressure in the secondary piping, the tertiary pump can be de-energized. In buildings close to the secondary distribution pumps, the tertiary pumps may seldom if ever run. Direct coupled tertiary pumps (without a cross over bridge) need to be controlled with variable speed drives and control algorithm that stages the bypass and pump speed as required to maintain the build coil pressure. With the tertiary pump connected directly to the secondary return it is possible for the pumps to disrupt the flow in the tertiary loops of adjacent buildings. With demand based reset of building pressure (refer to the controls chapter), this issue is seldom a problem. This elimination of the cross over bridge has been successfully retrofitted on a number of large campuses and shown to significantly decrease pumping energy.

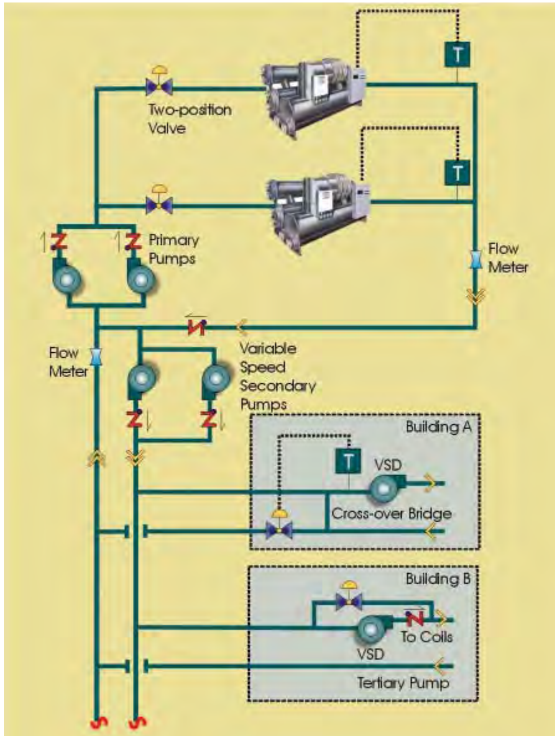


Figure 4-11: Primary/Secondary Variable Flow Piping, Tertiary Pumping

Coil Pumping Strategies

Sometimes variable-speed secondary pumps can be located directly at the cooling coils thereby decentralizing the pumping and eliminating the need for two-way control valves. This strategy works best when applied to large air handling units in a distributed design approach. Four coil pumping strategies are discussed here and shown in Figure .

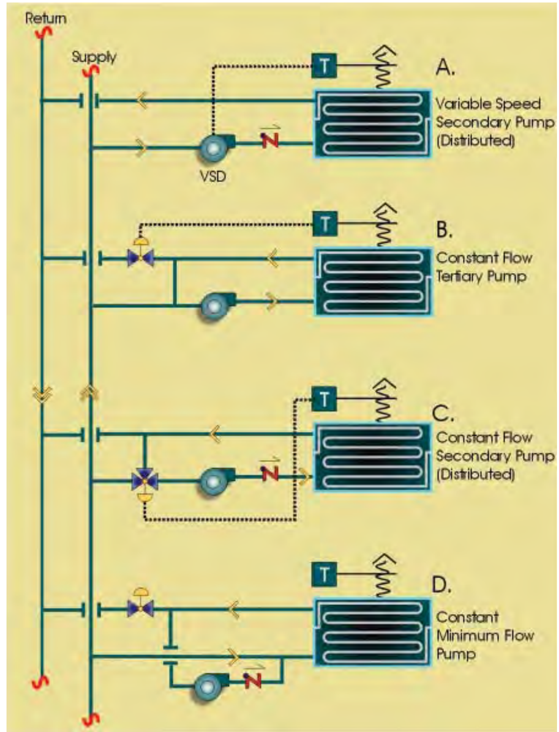
If it is desired to keep constant flow within the coil, a three-way valve can be added to the coil pump circuit that will allow variable flow between the coil and the common pipe (see C). Caution must be applied to this approach because the flow through the pump and coil will vary as a function of the position of the three-way valve. Flow variations can be minimized if the pressure drop between the three-way valve and the common pipe are kept very low. This approach does not save pumping energy and because of the potential for poor selections with low head coil pumps, the pumping energy may actually increase compared to other approaches.

A constant-volume pump can be added to the coil and piped in a primary/secondary configuration with a two-way valve located in the primary loop (see B).

A small coil pump can be added to the coil circuit to assure a constant minimum flow through the coil during periods of low loads (see D). The minimum flow can be based on maintaining a velocity above laminar flow conditions. During periods of higher demand the pump would be off.

For large coils (~100 gpm or larger) the variable speed coil pump in Figure 4-12A has proven to be both lower installed cost and lower energy than the traditional secondary pumps. If the building contains a mixture of both large coils and smaller coils, this scheme can only be used if the smaller coils are collected on one or more dedicated branches such that they can be served by a dedicated distributed secondary pump.

Figure 4-12:
Coil Pumping Strategies



The primary reason for the coil pump schemes in 4-12B, 4-12C and 4-12D is to maintain a high delta-T across the coil. The theory was that coil performance (delta-T) would degrade as the flow through the coil becomes laminar. Research on coils shows that this does not necessarily happen with coiling coils. In fact with fully circuited coils, delta-T appears to improve (not degrade) at lower flows (See Figure 4-14 below). Furthermore simulations have shown constant speed coil pumps (4-12B, 4-12C and 4-12D) use more energy than they could possibly save even if the delta-T is assumed to degrade drastically at low flows. These configurations should be avoided as they increase both installed and operating costs.

Variable-Flow System Design Considerations

This section discusses several design considerations for configuring the primary pumps. It also addresses the contentious issues of balancing valves and low delta-T syndrome and discusses several options for connecting separate chilled water plants and for connecting heat recovery chillers.

Primary Pump Arrangements

There are two basic options for locating the primary pumps (see Figure 4-13) on a chilled water system:

- Option A – Dedicate a pump for each chiller; or
- Option C – Provide a common header for the pumps and two-way isolation valves for each chiller.

The primary advantage of dedicating a pump for each chiller is that the pump can be custom-selected for the chiller it serves. Pump selection can then take into account variations in evaporator pressure drop and flow rates. Adding a standby pump (shown in Figure 4-14B) is cumbersome, however, because it requires extensive piping and manual isolation valves. This extra piping and the associated valves can be extremely expensive.

The other method is to pipe the primary pumps into a common header, then distribute the flow to each chiller. When a chiller is "off," a two-position valve at the chiller closes. Adding a standby pump is simple with this piping arrangement. In this arrangement with headered pumps, accurate balancing of flow between the chillers is more important.

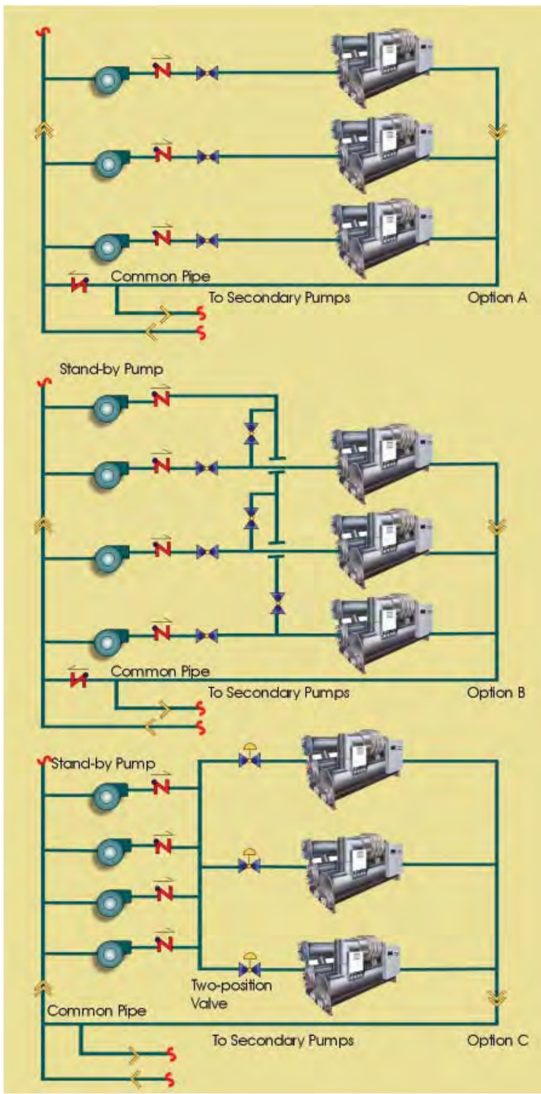


Figure 4-13:
Options for Primary Pumps

Balancing Considerations

This section summarizes the findings from the article "Balancing Variable Flow Hydronic Systems," by Steve Taylor and Jeff Stein. To write this article, the authors took a real design of variable flow chilled and hot water systems and used both hydronic and energy analysis programs to evaluate the following design options:

1. No balancing (relying on 2-way control valves to automatically provide balancing)
2. Manual balance, most commonly using calibrated balancing valves (CBVs) to measure and adjust flow
3. Automatic flow limiting valves (AFLVs)
4. Reverse-return
5. Oversized main piping
6. Undersized branch piping
7. Undersized control valves

A copy of that article is posted at http://www.taylor-engineering.com/downloads/articles/ASHRAE%20Journal%20-%20Balancing%20Variable%20Flow%20Hydronic%20Systems-Taylor%20&%20Stein.pdf. The summary of the findings follows.

Table 4-1:
Valve Maximum Operating Pressures and Flow Variations

	Balancing Method	Maximum pressure drop of control valve required for design flow, feet		Percent of design flow (percent of design coil sensible capacity) with all control valves 100% open			
				Maximum flow through closest coil		Minimum flow through most remote coil	
		CHW	HW	CHW	HW	CHW	HW
1	No balancing	20.5	44.4	143% (106%)	212% (119%)	73% (89%)	75% (96%)
2	Manual balance using calibrated balancing valves	0	0	100% (100%)	100% (100%)	100% (100%)	100% (100%)
3	Automatic flow limiting valves	20.5*	44.4*	100% (100%)	100% (100%)	100% (100%)	100% (100%)
4	Reverse-return	1.2	10.4	103% (100%)	150% (109%)	99% (100%)	85% (97%)
5	Oversized main piping	7.0	20.9	122% (103%)	173% (112%)	94% (99%)	82% (97%)
6	Undersized branch piping	19.5	NA	142% (106%)	NA	73% (100%)	NA
7	Undersized control valves	8.0	NA	120% (103%)	NA	86% (89%)	NA

Table 4-1 summarizes the transient performance of the balancing methods studied in the article.

The columns with maximum pressure drop data are for the condition of all valves at design flow. They indicate how much pressure the worst case control valve had to shed to get design flow. This is an indirect indication of valve controllability (the higher the pressure the more closed that the valve is under design conditions). It should be noted that a standard control valve has a 50:1 turndown ratio.

The last four columns indicate the condition of a transient warm-up or cool-down scenario where all the control valves are completely open. The numbers in each column indicate the percentage design flow. The numbers in parenthesis indicate the percentage sensible coil capacity at this flow. For the closest chilled water coil in the 1 no balance case, the flow was 212% of the design flow but this represented only 119% of design coil capacity. The worst case cooling coil had only 73% of the design flow but 89% of its design capacity.

In this and in the following table pressure independent valves are not included. They were not considered in the article, however their performance can be extrapolated from the costs and performance of the other options. For small valves, pressure independent valves cost the same as automatic flow limiting valves (option 3) and they have similar but slightly higher pressure drop. They will have the same transient performance as AFLVs.

Table 4-2:
Energy and Installed Cost Performance of the Design Alternatives

	Balancing Method	Pump head, feet		Annual Pump Energy, $/yr		Incremental First Costs vs. Option 1			
						$		$ per design gpm	
		CHW	HW	CHW	HW	CHW	HW	CHW	HW
1	No balancing	58.5	82.7	$1,910	$3,930	—	—	—	—
2	Manual balance using calibrated balancing valves	60.3	83.6	$1,970	$3,970	$7,960	$47,530	$6.60	$88.00
3	Automatic flow limiting valves	66.6	90.8	$2,170	$4,310	$11,420	$50,750	$9.50	$94.00
4	Reverse-return	55.3	80.0	$1,810	$3,800	$28,460	$17,290	$23.70	$32.00
5	Oversized main piping	45.0	59.3	$1,470	$2,820	$12,900	$7,040	$10.80	$13.00
6	Undersized branch piping	58.5	NA	$1,910	NA	($250)	NA	($0.20)	NA
7	Undersized control valves	58.5	NA	$1,910	NA	($2,340)	NA	($2.00)	NA

Table 4-2 summarizes the pump head, annual pump energy costs and incremental first costs for each of the options. The first costs are relative to the no balancing option.

As you can see in this table, automatic flow limiting valves and calibrated balancing valves have a very large incremental cost and provide very little benefit compared to the no balancing case. The prime advantage for automatic flow limiting valves is the performance during transients. For a similar cost you could use pressure independent control valves and get better control as a bonus.

It is also worthwhile to compare option 4 reverse return with option 5 oversized main piping. The reverse return costs more in both installed and operating costs.

Balance valves serve to equalize flow to multiple hydronic circuits. Typically, balance valves are located at each coil and on major branches of distribution piping. Balance valves are also sometimes located on the discharge of pumps. In many cases the balance valves also serve as flow meters.

The recommendations for balancing of variable flow systems are as follows:

- For other than very large distribution systems, option 1 (no balancing) appears to be the best option. This has a very low first cost, excellent energy performance and minimal or insignificant operational problems.
- For systems with long hours of operation, the added cost of reverse-return piping at the floors and oversized mains in the risers appears to be the best option based on pump energy savings. For floors with a loop distribution, reverse return can be achieved at a minimal cost penalty by looping the chilled water supply in one direction (e.g. clockwise) while looping the chilled water return in the other (e.g. counter clockwise).
- Undersizing piping and valves on the non-critical runs can reduce first costs but require significant additional engineering time. If the branches are reduced too far, they could become the index runs and cause increased energy usage.

- Automatic flow limiting valves and calibrated balance valves are not recommended on chilled water distribution systems under any circumstance. They increase first cost, construction labor and energy usage. They can also introduce coil performance problems under certain operating conditions (refer to the article for a detailed discussion).
- For systems with high design pressure consider using pressure independent control valves on the coils nearest to the distribution pumps (those that have to spill the largest system pressure).

These observations apply to variable flow systems. Balancing valves are required on constant flow systems.

Low Delta-T Syndrome

In most variable-flow chilled water plants, it is assumed that "delta-T" (the difference between return and supply chilled water temperature) will remain relatively constant. Because load is directly proportional to flow rate and delta-T, the following relationships apply:

Equation 4-1

$$Q = \dot{m} c_p \Delta T$$
$$= 500 \text{ GPM } \Delta T \qquad \text{(IP units)}$$

If delta-T is constant, it follows that the flow rate must vary proportionally with the load. Most variable-flow systems are designed based on this assumption and fail to perform well if the delta-T does not stay relatively constant.

But in almost every real-world chiller plant, delta-T falls well short of design levels, particularly at low loads. The result is higher pump and chiller energy usage. Plants that have been designed to accommodate high delta-T can fall woefully short of capacity when it is discovered that the return water is not as warm as it should be. That translates to wasted first cost for unused plant capacity.

This section addresses the causes of low delta-T and offers mitigation measures. The next section, Designing Chiller Plants to Accommodate Low Delta-T, explains why low delta-T is almost always present in chilled water systems and how to design efficient chiller plants that accommodate low delta-T.

Either a flow-based or a load-based strategy usually controls the sequencing of chillers and pumps in primary/secondary variable flow chilled water plants:

- The flow-based strategy operates enough chillers and pumps to ensure that the secondary system flow to coils is adequate and that the primary system flow is equal to or greater than the secondary flow. If the temperature of the return water from the secondary system is below design (low delta-T) then the chiller entering water temperature is also below design and the chiller cannot be fully loaded. For example, if a system were designed for a 14°F delta-T, but at 50% load the actual delta-T was only 7°F, both the primary and secondary pumps would be at full flow. There would be no opportunity to shut down chillers, pumps and cooling towers for energy efficiency even though the load did not justify this level of operation.

- The load-based control strategy does not start a new chiller until the operating chillers are loaded. But with low delta-T, there is no opportunity to load the chiller (except to increase flow through the chiller). The result is that the secondary flow quickly becomes greater than the primary flow, with the consequence that supply temperatures to the coils rapidly rise. This creates an unstable control cycle in which the warmer supply water temperature causes the control valves to open, which in turn creates a demand for even greater flow.

The performance (or lack thereof) of the cooling coil and control valve creates the low delta-T problem. The solution is to design for maximum delta-T at the coils as much as possible. Even then, however, low delta-T is inevitable, particularly at low loads. Therefore, the plant should be designed to accommodate the low delta-T that will occur. The following are causes of and mitigation steps for low delta-T syndrome:

Low Delta-T Cause 1 – Improper Setpoint or Controls Calibration

Imagine this scenario: the boss complains to the maintenance technician that it is too warm in his office. The technician responds by lowering the setpoint on the air handler a few degrees, thinking that he has solved the problem. Unfortunately, however, the setpoint is so low that the controller commands the control valve to be 100% open, but even that may not satisfy the setpoint. (There would be a similar result if the discharge controller was set, for example, at a 55°F leaving air temperature but the calibration was off several degrees.) Table 4-3 shows how even a modest drop in supply air temperature setpoint from 54°F to 51°F can cause coil flow rate to more than double and delta-T to drop in half. This is probably the greatest single cause of low delta-T syndrome.

Mitigation. Check setpoints and recalibrate controls regularly. Use pressure independent delta-P control valves or automatic flow control valves on each coil.

Leaving Air Temperature (LAT) Setpoint	GPM	Delta-T	% of Design GPM
54	80	13	100%
53	104	11	130%
52	143	8.5	179%
51	208	6.5	260%
50	327	4.3	409%
49	Cannot be attained		

Table 4-3: Coil Performance (at full load) for Low LAT Setpoints

Low Delta-T Cause 2 – Laminar Flow in Coils

The heat transfer coefficient in a straight tube is primarily a function of flow turbulence, which is described by the Reynolds number, a dimensionless value defined as:

$$Re = \frac{VD\rho}{\mu}$$

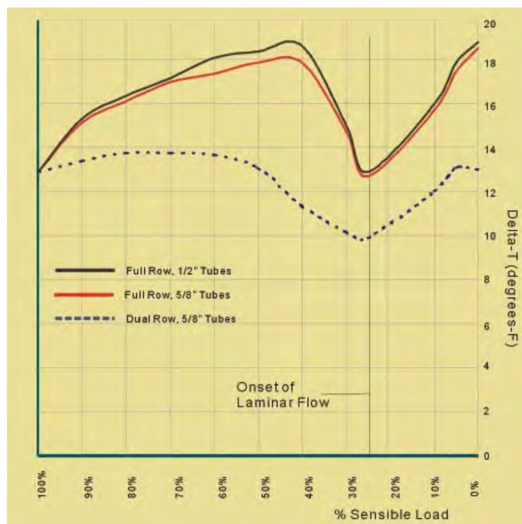
Equation 4-2

where V is the tube velocity, D is the tube diameter, ρ is density of the water, and μ is the viscosity. When the Reynolds number is above 10,000 (2.9 fps in 5/8" OD tube), flow is turbulent. When the Reynolds number is below 2,000 (0.6 fps in 5/8" OD tube), flow is laminar. Between these velocities, flow is in transition, resulting in unpredictable heat transfer coefficients.

Because the performance of a cooling coil is somewhat unpredictable for flow with Reynolds number below 3,100 (about 1.0 fps in 5/8" OD tube), ARI does not certify performance below this number. Conventional wisdom is that the delta-T will drop dramatically, as the flow becomes laminar and that there will always be a point where decreased flow will cause the heat transfer coefficient to increase sharply. Actual performance of cooling coils may be different than conventional wisdom. Some manufacturer's coil selection programs show a slight degradation of the delta-T at low flows (less than 1.0 fps) but not to the extent one would expect from the heat transfer textbooks. Testing from several manufacturers (unpublished) corroborates the results of the computer program. One likely explanation for this phenomenon is that the bends at the ends of the rows creates turbulence that enhances the thermal performance of the coil.

The idea that low delta-T is caused by laminar flow in the coils may be overstated. Yet prudence dictates that extending the range of fully turbulent flow within the coil should be a benefit and worth the modest efforts suggested below.

Figure 4-14:
Delta-T at Part Load



Mitigation. Designers specifying cooling coils should not only schedule the maximum performance of the coil but should also specify a "low load" coil condition so that coils are selected with low delta-T in mind. There are three ways to mitigate this problem:

- The first solution is to size coils for a high initial tube velocity. Selecting partially circuited coils (½ or ¼ circuits) can accomplish this. The consequence of using this approach is accepting a higher coil pressure drop. Accepting coil pressure drops of 15 to 20 feet is not uncommon.
- Another way to delay the onset of laminar flow is to use coil "turbulators" or "turbo spirals." These are spring-shaped spirals of wire that fit inside the tube to increase turbulence. One manufacturer claims that turbulators prevent laminar flow for a Reynolds number as low as 450 (versus 2,000). The downside is increased pressure drop through the coil, additional cost, and greater potential for fouling.

- Finally, coil pumps can be added to the design to maintain either a constant flow or a constant minimum flow through the coil. Constant flow pumps may increase pump energy costs because of their small size and low head, and they are inherently less efficient than larger secondary pumps. Constant minimum pumps are sized to ensure that the minimum flow through the coil is always above the laminar velocity and only runs when the control valve has reduced the flow to a predetermined minimum. This tends to be a smaller horsepower pump with limited run times so the pumping energy impact is reduced.

Low Delta-T Cause 3 – Tertiary Connections and Control

Figure 4-11 shows a typical connection from a secondary piping system to a tertiary connection. This type of connection is used at the interface of a building when the building needs more differential pressure than is available from the secondary pumps, or when the supply temperature of the building is intended to be greater than that supplied by the secondary system. In this situation the pump is sometimes referred to as a "blend pump." The two-way valve is modulated to maintain the supply water temperature in the building, which must be higher than the supply water temperature available from the secondary loop. If the building setpoint temperature is lower than the secondary water supply temperature or if the control is out of calibration, the two-way control valve will open in an attempt to satisfy the load. The result will be full flow through the crossover bridge.

Mitigation. Calibrate the controls frequently and add either a pressure independent delta-P valve or an automatic flow control valve. Consider deleting the crossover bridge and connecting the tertiary pump in series with the secondary pump.

Low Delta-T Cause 4 – Using Three-Way Valves

Three-way valves by their nature bypass supply water into the return to control temperatures at the load. As described above in the constant flow discussion, at part load, three-way valves actually increase the flow to their branch circuit because of the drop in pressure from the variation of flow in the coil. This exacerbates the low delta-T problem. An acceptable design practice is to carefully place a few three-way valves or two-way bypass valves in the system to ensure a minimum flow for the variable speed secondary pumps.

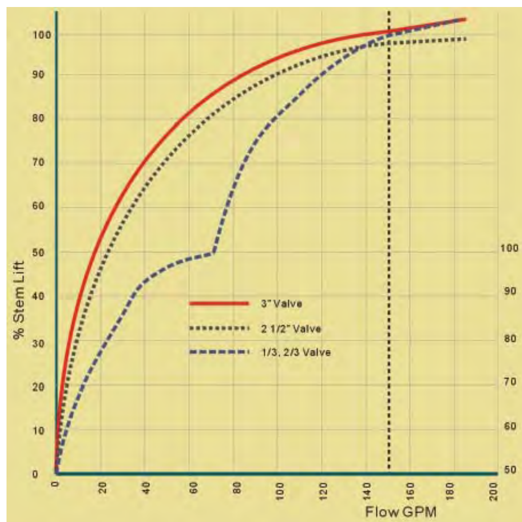
Mitigation. Do not use three-way valves in variable-flow systems. One way to maintain minimum flow in the system is to strategically place two-position bypass valves across the supply and return with flow control valves and open these valves only when the flow at the pumps approaches the minimum flow.

Low Delta-T Cause 5 – Poor Two-Way Valve Selection

Properly sizing the control valves is important in any system, but in variable-flow systems it is particularly critical. Manufacturers usually recommend that wide-open control valve pressure drop be equal to or greater than the pressure drop of the coil plus the pipe and fittings connecting them to the supply and return mains. Typical engineering specifications for control valves call for sizing them at a 3 to 5 psi pressure drop. They also recommend the use of equal percentage globe valves. As noted in Chapter 5, full ported ball valves have excellent equal percentage flow characteristics.

Figure 4-15 shows the flow variation as a function of valve stem position for a typical equal percentage valve. Note that double-seated valves do not provide tight shut-off. Using high pressure drop cooling coils for low flow conditions means that the typical control valve pressure drop moves to 7 to 9 psi. A control valve has a "rangeability," which is the ratio of the maximum controllable flow to minimum controllable flow. Typical rangeabilities are 40 or 50 to 1. But the design flow for a cooling coil is nowhere near the "maximum controllable flow," with the result that in practice the actual rangeability may be only 4 or 5 to 1. Larger valves typically have less rangeability than smaller valves. Oversized valves do not control well at low flows. During low flow operation there is a tendency to "hunt," which means that the valve alternately opens and closes, causing over- and under-shooting of the setpoint. This results in very unstable operation. See Chapter 5, "Controls and Instrumentation," for a more detailed discussion of control valves.

Figure 4-15:
Flow Variation vs.
Percent Stem Lift



Undersizing the actuator is another common problem. Manufacturers recommend that the actuator on a control valve be able to close the valve tightly with a pressure of up to three times the differential across the supply and return mains at design conditions. Where valves are located close to pumps on large systems, the requirement may even be higher. If a valve does not have the correct close-off pressure rating, high differential pressures in the system may cause the valve to open when that is not intended. This results in excess water flow and low delta-T. In a retrofit application some designers may be tempted to close off the bypass side of a three-way valve to create an inexpensive two-way valve. The problem with this is that the three-way valve actuator most likely will not have the close-off rating to shut the valve.

Two-position (on-off) control valves such as those used to control small fan coil units are often blamed for low delta-T problems. If these valves are not equipped with flow control valves, or piped in a reverse return arrangement, they may consume more water flow when open than the design calls for. With full flow through the coil, at partial loads the delta-T will invariably be lower than design. But since the air temperature entering the fan coil unit is fairly constant and is usually not subject to outdoor air conditions, the delta-T will not degrade significantly.

Mitigation. Control valves must be selected with consideration for the pressure drop of the load served and the available differential across the supply and return mains. For larger loads it is more effective to use two smaller control valves instead of one larger valve. This can extend the rangeability considerably, especially when the valve operation is staggered. If multiple control valves are used they can be split equally (or sometimes a one-third/two-thirds split will work best). Using pressure independent delta-P valves is a very good (but expensive) option, as these valves automatically compensate for excessive differential pressure and have excellent control of flow over the entire range of operation. It is important to check the shut-off pressure of the valves selected. Normally closed chilled water valves may require "industrial" grade actuators with large force springs not only to close against the available head pressures but also to offer stabilized flow control over the entire flow range. If two-position valves are used, provide flow control valves or use a reverse-return piping system.

Low Delta-T Cause 6 – No Control Valve Interlock

Sometimes when a pump is shut down, the control power and the control valve remains operational. The controller will futilely try to achieve the desired space or discharge temperature until the control valve eventually fully opens.

w. Interlock the controls so they are disabled during "off" times. In a DDC control system this is a programming issue. In a pneumatic control system, pressure electric (PE) switches can be added to interlock the controls.

Low Delta-T Cause 7 – Reduced Coil Effectiveness

Coil heat transfer effectiveness is reduced by waterside fouling (e.g., slime, scale, or corrosion on the inside of coil tubes), airside fouling (e.g., dirt build-up on coil fins), airside deterioration (e.g., deteriorating fins), non-uniform air distribution across the cooling coil, and coil bypass air. Any reduction in coil effectiveness increases the flow rate of water required to deliver the desired leaving water temperature, thus reducing delta-T.

Mitigation. Waterside fouling is easily controlled by proper chemical treatment. Since the chilled water piping is most often a closed system, water treatment need not be an ongoing expense. Reducing airside fouling is a very good reason to consider increasing the filter efficiencies. Given the proximity of fans to the cooling coil and the need to change directions quickly within air handling units, it is sometime difficult to achieve uniform air distribution, but every effort should be made to ensure this is done. Testing for proper air distribution across the cooling coils is an important component of the commissioning process. (See Chapter 8 on commissioning.)

Low Delta-T Cause 8 – Outdoor Air Economizers and 100% Outdoor Air Systems

One issue very often overlooked as a cause of low delta-T on systems designed for high delta-T (that is, above 14°F) is the impact of integrated outdoor airside economizers and 100% outdoor air systems. With these systems, when the weather is cool but not cold enough to provide 100% of the system's cooling load, these systems deliver 100% outdoor air but need a small amount of chilled water to meet cooling demands. Under these conditions, the air temperature entering the coil is low, causing correspondingly low return water temperatures.

For instance, a coil might be designed for 80°F entering air temperature with a chilled water return temperature of 60°F. When the outdoor air temperature is 60°F, it is clearly impossible to maintain a 60°F return water temperature. A coil on a VAV system designed for 44°F chilled water and an 18°F delta-T would only be able to achieve an 11°F to 15°F delta-T at 55°F to 65°F outdoor air temperatures.

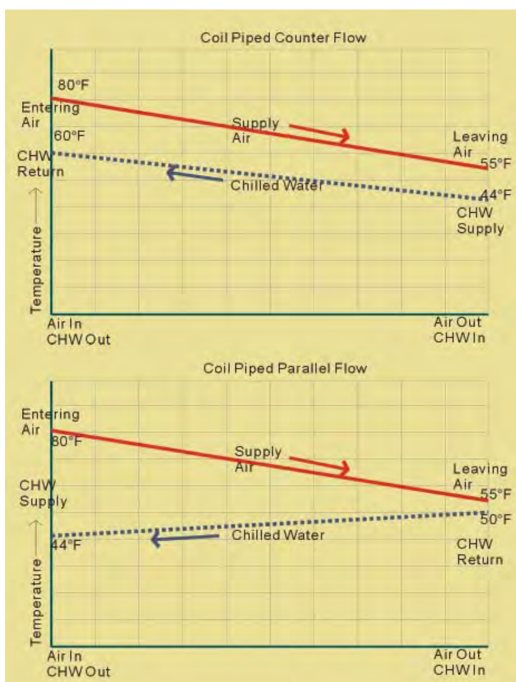
Mitigation. In these cases the only alternative available is to accept that there will be times when low delta-T is inevitable and to design the central plant to accommodate this. This is discussed further in the section, "Designing Chiller Plants to Accommodate Low Delta-T."

Low Delta-T Cause 9 – Improperly Piped Coils

It is not uncommon to find chilled water coils piped backwards. Instead of being piped in a counter-flow arrangement, they are piped in a parallel-flow arrangement with water entering the coil on the same side as the entering air. A coil-piped counter flow can achieve "overlapping" temperature ranges with the supply air. For example, the leaving water temperature can enter at 44°F and leave at 60°F while the supply air enters at 80°F and leaves at 55°F. Figure 4-16 shows the relationship between air and water temperatures with coils piped in both parallel and counter flow. With parallel-flow piping, the leaving water temperature will always be a few degrees cooler than the leaving supply air temperature. Thus if 55°F is maintained, flow must be much higher and the return water temperature will only be in the low 50s. The desired delta-T would be impossible to attain.

Mitigation. Re-pipe coils in counter-flow arrangement.

Figure4-16: Impact of Piping Coil Backwards



Low Delta-T Cause 10 – Chilled Water Reset

Chillers are more efficient at higher leaving water temperatures, so when loads are low, setting the chilled water temperature higher can be an effective energy-saving strategy. However, high chilled water temperature will reduce coil performance. As a result, coils will demand more chilled water and delta-T will be lowered.

Mitigation. The best chilled water reset strategy will vary depending on the plant design, the chiller performance characteristics, and the nature of coil loads. Smaller plants with low pumping distribution losses will also usually benefit from chilled water reset. For large plants with high pumping distribution losses, raising chilled water temperature will increase pumping energy more than it reduces chiller energy, resulting in a net increase in plant energy usage. During mild weather when delta-T is bound to degrade because of low entering air temperature, resetting chilled water supply temperatures down will have the effect of increasing the delta-T. Large plants may benefit from lowering chilled water setpoint even below design levels in mild weather.

Low Delta-T Cause 11 – Uncontrolled Process Loads

A chiller plant in an industrial environment may serve process loads in addition to cooling coils. Some process equipment have no flow control devices and hence use as much chilled water when they are "on" as they do when they are "off." The system designer may not be aware of this. He or she is directed to deliver a certain amount of chilled water to a process device, but may not understand how the device operates (for instance, whether it includes any modulating or shut-off controls). When there are no controls, chilled water delta-T falls whenever these process systems are not at full load.

Mitigation. The designer should work with the process equipment supplier to determine if controls are present, or if not, whether external, field mounted shut-off valves may be installed.

Designing Chiller Plants to Accommodate Low Delta-T

The previous discussion shows that low delta-T syndrome is caused by many conditions, most of which can be avoided by careful design practices (including the proper selection of cooling coils and control valves) and attentive operation. There are, however, situations where low delta-T syndrome is inevitable. For instance:

- Putting constant minimum pumps on all coils may not be cost effective.
- Units with economizers or 100% outdoor air will at some point produce low delta-T.
- It is inevitable that coils will become fouled to some degree.
- Perfectly uniform air distribution on the face of all coils may not be achievable.

Low Delta-T Strategy 1 – Choosing Design Delta-T

When designing variable flow systems to accommodate some degree of low delta-T, a three-tiered design strategy is recommended.

1. Select cooling coils for the highest delta-T the designer believes is practical. For example, the coils could be selected for a 15°F to 18°F delta-T. The entering water temperature at this level could be slightly higher than what is expected at the plant, such as 46°F for the coil versus 44°F for the plant. The designer must ensure that the coil will perform the dehumidification duty when selecting the entering water temperature. This will result in a slightly more massive coil with a somewhat higher water pressure drop than otherwise.
2. Select the delta-T of the secondary (or tertiary) system so that the system accommodates the connected load (minus diversity as may be appropriate) and the design delta-T is reduced 1 or 2°F. For example, if the coil selections are based on 16°F delta-T, the secondary (or tertiary) system flow rate should be based on 14°F delta-T (that is, greater flow capacity).
3. Finally, select the primary system at the chillers for a 1°F delta-T lower than the secondary system and for a leaving water temperature at least 2°F lower than the supply temperatures used in the coil selections.

This somewhat conservative design results in pumping capacities and possibly pipe sizes that are somewhat larger than might otherwise be specified. But this strategy acknowledges the inevitable deterioration of the cooling coil performance over time. It also allows the system to continue to deliver the full capacity of the plant under real world conditions, which are invariably harsher than what many designers expect. The advantage of this approach is that if everything does perform according to design, the system will respond accordingly, and the actual flows and delta-T will achieve the goals of the design.

Low Delta-T Strategy 2 – Over-Pump Primary Chillers and Add VSDs

When a chiller plant has multiple machines piped in parallel and the delta-T is lower than design, the result is that one or more additional chillers must be activated (not because of load but because of flow). One method of minimizing this phenomenon is to pump more flow through the evaporators on the primary side. With a lower entering water temperature but a higher flow than design, the machine can continue to perform. Pumping more flow through the evaporators involves increasing the size of the primary pumps. This will also increase the head and horsepower of the pumps.

In primary-only systems, this is a very modest proposal. In primary/secondary systems, providing a variable-speed drive on the primary pump allows the flow through the evaporator to vary from a minimum (say, 40% of design) to a maximum (say, 125% of design). The speed of the pump(s) can be varied to match the flow of the secondary system so that the chillers always get the maximum water temperature available and the flow in the common pipe is always near zero. (For more information, refer to the previous discussion on variable flow in the evaporator of a chiller.) Another way to accomplish the variation in primary flow through the evaporator is to install a two-way control valve in the primary circuit and modulate the valve to provide the variable flow. A life-cycle analysis could be performed to determine the best approach versus utilizing variable-speed drives.

Low Delta-T Strategy 3 – Add Check Valve in Common Pipe

Figure 4-17 shows how adding a check valve to the common pipe results in a secondary flow that can never be greater than the primary flow. If a low delta-T situation occurs and the secondary flow increases above the primary flow of one chiller, the check valve will cause the secondary pump to drive the water through the primary pump. The two pumps will be in series. Normally, the primary pump and evaporator can accommodate additional flow. When the secondary pump flow increases the primary pump flow, the primary pump rides on its pump curve to the right. The amount of additional flow through the primary pump depends on where the primary pump was operating on its pump curve and on the steepness of the pump curve. The evaporator generally does not have a problem accommodating a greater flow. Additional pump flows of 25% to 40% are not uncommon. Adding a check valve is an excellent retrofit opportunity for existing plants that suffer from low delta-T syndrome.

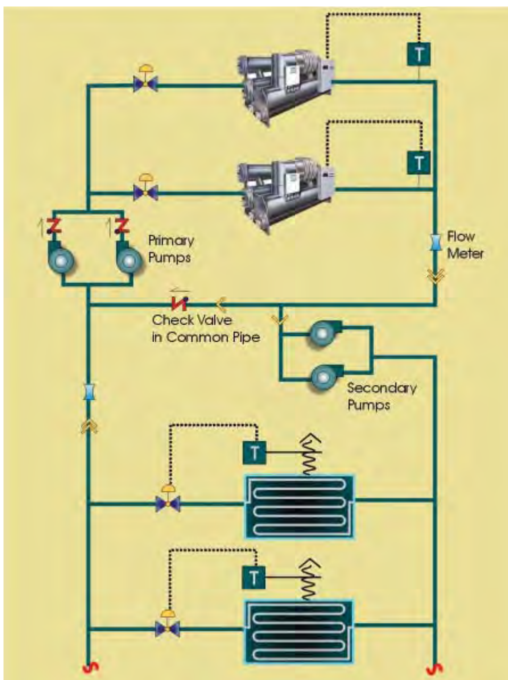


Figure 4-17:
Check Valve in Common Pipe

Connecting Multiple Chiller Plants

Many campuses and very large buildings have been constructed with a central chilled water plant in each building or major addition. Frequently, one or more of these plants have extra capacity. Also, there are many hours when two or more plants are operating at very low loads. If these plants were combined, energy savings could be achieved by running only one plant (or at least fewer machines). Energy savings are available if plants are combined in such a way that new, more efficient, chillers are operated as lead machines, with the older machines used only for peak loads. Although there are many design options for combining chiller plants, two methods are shown in this section.

One caution that must be recognized is that when connecting multiple chiller plants, each plant may have an expansion tank and fill valve for pressure control. Since the expansion tank represents the "point of no pressure change," having multiple expansion tanks in the combined system can change the pressure characteristics at the expansion tanks. This may result in unexpected relief valve discharges. A thorough analysis of system pressures should be made during the design process.

Figure 4-18 shows a typical piping arrangement for connecting two chiller plants from the primary pumping loop. This method provides a transfer pump with automatic valves piped in such a way that the excess capacity from either plant can be transferred to the other plant. The flow rate of the transfer pump varies as a function of the flow demand of the secondary loop that is receiving the excess chilled water. This method is particularly useful when the differential pressure requirements (in the secondary circuit) of each chiller plant are significantly different.

Figure 4-18:
Connecting Two Chiller Plants on Primary Side

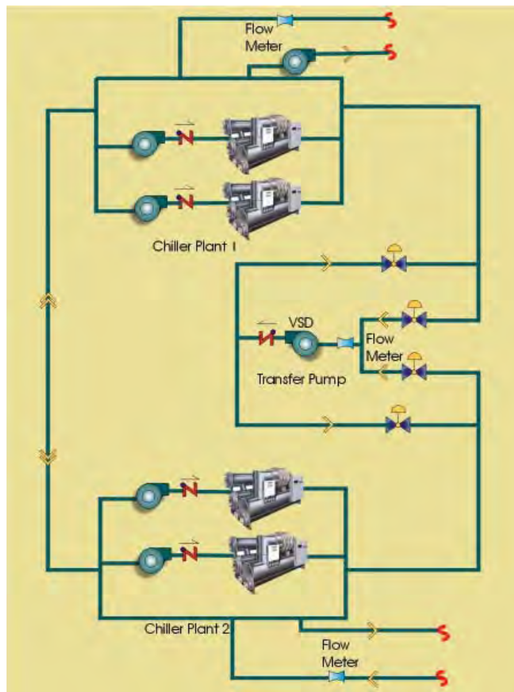


Figure 4-19 shows a typical piping arrangement for connecting several chiller plants from the secondary pump loop. In this diagram several buildings have chillers and one does not have a chiller. In this method, the differential pressure sensors located in the remote piping of each loop are used to control the variable-speed drives of the secondary pumps. Sufficient differential pressure is maintained to ensure adequate flow to remote cooling coils. Series booster pumps can be integrated into remote piping loops, as shown in the diagram for the building without the chiller (see Figure 4-19). The bypass around the pump is used when there is sufficient differential pressure in the mains to satisfy the building coils without the tertiary pumps.

Experience with these types of systems indicates that the most successful approaches are those that limit the differential pressure in the interconnecting piping. This may involve the addition of secondary or tertiary pumps.

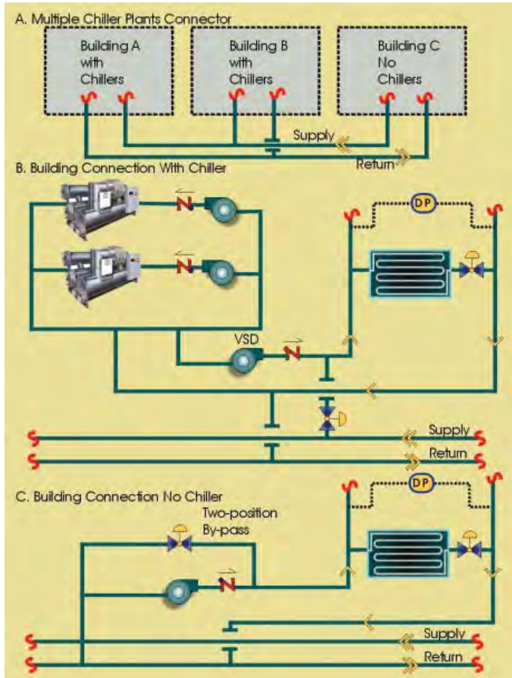


Figure 4-19: Connecting Multiple Chiller Plants on Secondary Side

Connecting Heat Recovery Chillers

The heat rejected from the condenser of a chiller can be used for many purposes, including domestic water preheating, process heating, and building heating. Heat recovery chillers are usually sized for a small portion of the total cooling load because of the need to have a simultaneous mechanical cooling load and heating load, and because of the lower cooling efficiency of heat recovery chillers. An unsatisfactory strategy for incorporating a heat recovery machine into a chiller plant is to pipe the chiller in parallel with the other chillers in the primary loop. The problem with this approach is that constant-flow primary chillers will almost always have a percentage of the cold supply water bypassed into the return, thereby decreasing the temperature of the water entering the chiller. This decreased entering temperature can diminish the heat recovery potential (cooling load) of the machine. In primary-only variable flow systems where the flow through the evaporator is allowed to vary, this is not as much of a concern.

Another method of dealing with a heat recovery chiller is to pipe it for "preferential" loading. Figure 4-20 shows a heat recovery machine piped in parallel with other chillers, but the location of the heat recovery primary pump suction pipe is such that it "sees" only the warmest return water from the system. Any bypass flow will go to the other chillers unless the heat recovery machine is the only one "on." A problem with this approach is that there is no way to effectively unload the heat recovery machine during times when the heating load is low. The "preferentially" loaded machine will be required to cool its full volume of warm return water and because the need for recovered heat is low, most of the heat will be rejected out the cooling tower. The COP of the heat recovery machine operating at elevated condenser pressures is relatively low, resulting in inefficient operation for cooling purposes.

Figure 4-20:
Preferentially Loaded Heat
Recovery Chiller

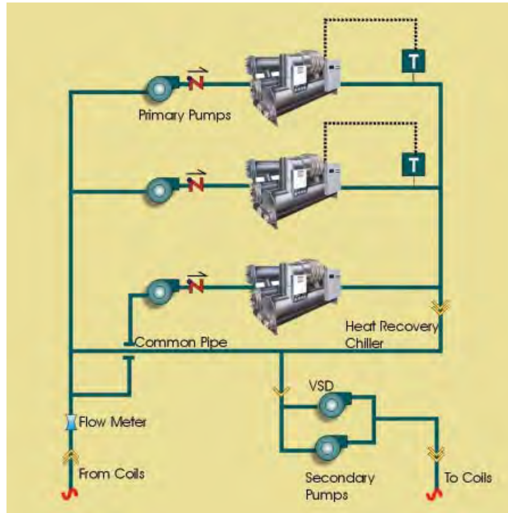
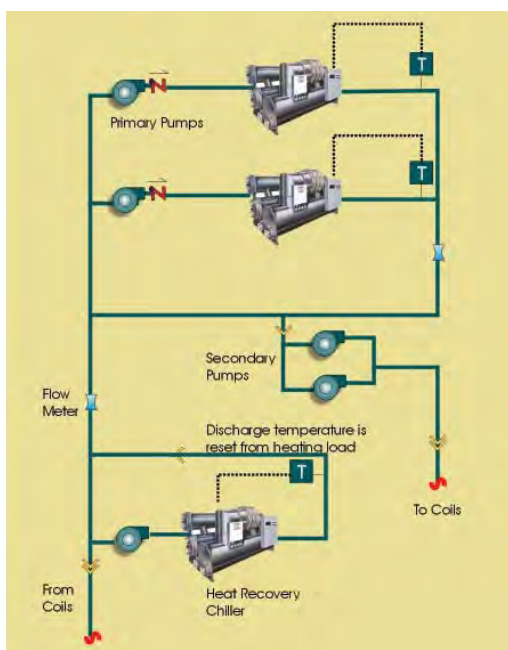


Figure 4-21 shows a configuration for a heat recovery machine that essentially puts the machine in series with the remaining chillers in the plant. Warm return water is pumped to the chiller then back into the return, thereby precooling the inlet water to the other chillers. The heat recovery machine can remove as much or as little heat as is needed for the heating load.

Heat recovery systems are sometimes designed to collect as much "waste heat" as possible during periods of high heating demand. This waste heat may include heat from exhaust air streams, transformer vaults, or even return air that would otherwise be exhausted through the relief vent. Sometimes this is called "false loading" of the machine. One strategy to control the amount of heat collected (cooled in the evaporator) is to provide a "load shed economizer." With this strategy, a temperature sensor in the hot water heating system signals the need to either reject heat out the cooling tower or to stop collecting it. As the heating water temperature rises, global signals are sent from the central DDC computer to close off exhaust cooling coil valves, and to open (or allow to open) outside air dampers (economizer dampers) on air handling units serving various loads. As much cooling load as possible is shed before the heat is rejected out the cooling tower.

Figure 4-21:
Heat Recovery Chiller
in Series



Introduction

In designing energy-efficient central chilled water plants, it is extremely important to select the proper condenser water system. The efficiency of the chillers is affected not only by the operation of the cooling towers and associated pumps, but also by the temperature and quality of the condenser water. In this section the following aspects of condenser water systems are discussed:

- Selection of the proper style of cooling tower
- Location of condenser water pumps
- Options for piping multiple chillers and cooling towers
- Integration of waterside economizers
- Integration of auxiliary condenser water circuits
- Piping practices for heat recovery chillers

Choosing the Style of Cooling Tower

As shown in Chapter 3, there are numerous commercially available styles of cooling towers, including:

- Packaged induced draft, axial fan
- Field-erected induced draft, axial fan
- Forced draft, axial fan
- Forced draft, centrifugal fan
- Closed circuit evaporative cooler, axial fan
- Closed circuit evaporative cooler, centrifugal fan
- Spray towers

When choosing which cooling tower is most appropriate for a particular application, the following factors should be considered:

Packaged versus Field-Erected Cooling Towers

The type of cooling tower selected will be determined to a great extent by the required capacity.

- Packaged cooling towers are manufactured to be cost effective and to ship on standard-size carriers. Typically, a single cell of a packaged tower will handle a maximum cooling capacity of 650 to 1,000 tons at ARI conditions. Larger plants will require multiple cells.
- If the chilled water plant is large, field-erected cooling tower cells may be more cost effective than a packaged cooling tower. Field-erected cooling towers also offer a greater degree of energy efficiency because the design flexibility makes it possible to match lower horsepower fans with larger fill volume. Although field-erected cooling towers offer greater flexibility when the site has physical constraints, they may have longer procurement times than packaged cooling towers.

Direct vs. Indirect Cooling Towers

As discussed in Chapter 3, the direct cooling tower is the most prevalent type of cooling tower used in the HVAC industry.

The indirect cooling tower is at a disadvantage due to its higher first cost, additional energy cost, and larger physical size. The indirect cooling tower could be used:

- where the condenser water pumps are located remotely from the tower;
- where the cooling tower is located below the condensers; or
- where it is necessary to keep the condenser water free from contamination with dirt or impurities.

Centrifugal vs. Axial Fans

In Chapter 3, the comparison of centrifugal versus axial fans in cooling towers demonstrates that the axial fan is significantly more energy efficient in cooling tower applications. The use of centrifugal fans in cooling towers should be limited to situations where:

- low profile towers are required;
- the site is constricted so that clearances for intake air is only practical with centrifugal fans; or
- sound traps or other acoustical considerations make the centrifugal fan the only choice.

Spray Towers

Spray towers are seldom used in the HVAC industry because of maintenance and control limitations. Spray towers could be used where very quiet operation is required.

Locating the Condenser Water Pumps

When using a direct cooling tower, the water falls by gravity into the collection basin and sump. (The sump is a depressed chamber below the collection basin into which the water flows to facilitate pump suction.) Typically, on a packaged cooling tower the collection basin and sump are an integral part of the tower. At the outlet to the pump suction there should be a screen for collecting larger debris and a device for breaking the vortex created by the suction. From the cooling tower outlet, the water flows into the suction of the pump. Typically, there is either a *Y-strainer* or a *basket strainer* located upstream of the pump suction.

When considering the location of the condenser water pump, great care must be taken to ensure that *cavitation* does not occur. This usually means that the pump must be in relatively close proximity to the cooling tower and at a lower elevation.

EXAMPLE 4-2:
NET POSITIVE SUCTION HEAD

To prevent cavitation at the pump, the net positive suction head available (NPSHA) must be greater than or equal to the net positive suction head required (NPSHR), as shown in the following relationship:

$$\mathbf{NPSHA \geq NPSHR}$$

The NPSHA is given by the following equation:

EQUATION 4-3

$$\mathbf{NPSHA = H_a + H_s + H_{vpa} + H_f}$$

Where,

H_a atmospheric pressure on surface of liquid that enters pump, ft

H_s static elevation of liquid above center line of pump, ft

H_{vpa} absolute vapor pressure at pumping temperature, ft

H_f friction and head losses in suction piping, ft

Case 1. Consider an open cooling tower located at sea level. It has three pumps, each delivering 1,670 gpm at 50 ft of head. The temperature of water leaving the cooling tower is 85°F. The NPSHR is 8 ft, which is taken from the manufacturer's pump curve. The NPSHA is calculated as shown below:

$$\mathbf{NPSHA = H_a + H_s - H_{vpa} - H_f = 34 + 4 - 1.4 - 10.5 = 26.1}$$

This value of 26.1 ft is greater than the 8 ft required so there will be no cavitation.

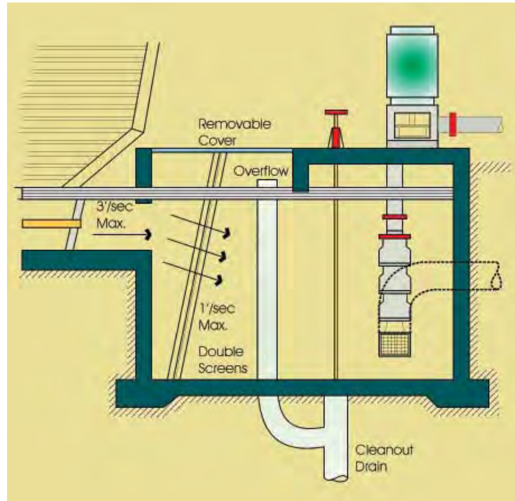
Case 2. Consider the same conditions as Example 1 except that the tower is located at 3,000 ft. The value of Ha for 3,000 ft is 30.5 ft, which is taken from a look-up table. The NPSHA is 22.6 ft as calculated below. There will still be no cavitation.

$$\mathbf{NPSHA = H_a + H_s - H_{vpa} - H_f = 30.5 + 4 - 1.4 - 10.5 = 22.6}$$

If the cooling towers are a significant distance from the chillers, the pumps are usually located adjacent to the cooling towers. This can present difficulties if the building is in a freezing climate. Freezing can be avoided by:

- providing a concrete sump (Figure 4-22) below the cooling tower collection basin (a concrete collection basin below the cooling tower greatly simplifies the suction piping and eliminates the need for "*equalizer piping*"); and
- using vertical turbine pumps to move the water back to the machine room ("*canned*" *vertical turbine pumps* with an integral barrel have been used successfully in this application).
- In cold regions electric heaters are added to the cooling tower sumps and exposed piping is heat traced to provide freeze protection.

Figure 4-22:
Typical Cross-section of Concrete Sump Pit



Piping Multiple Chillers and Cooling Towers

Maximum and Minimum Flow Rates

When water enters the cooling tower, it is distributed uniformly across the fill by means of spray nozzles or a gravity distribution basin with properly sized nozzles. Each cell of a cooling tower has a maximum and a minimum flow rate.

With a gravity distribution basin:

- If the flow is greater than the maximum rate, the water will overflow the distribution basin.
- If the flow is below the minimum flow rate, the water will not be distributed evenly in the basin.

In the minimum flow condition, the portions of the fill without water will have less air pressure drop. This causes most of the air to flow through the dry media. The result is that the cooling tower will not be able to cool the water as effectively as if it were wet. A side effect is that water droplets will become entrained in the air and will spit out the air discharge. Some distribution basins are bisected longitudinally with a weir several inches high to help distribute water evenly during low flow situations.

Because similar problems can occur when using spray nozzles, some manufacturers have patented spray nozzles that have excellent flow distribution patterns at low flow.

In plants with multiple cooling towers and chillers, the maximum and minimum flow conditions may create a need for automatic isolation valves on the inlet and/or outlet piping.

Equalizer Piping and Maintaining Sump Levels

When piping multiple cooling towers, the water flow rate drawn from the sump is never exactly equal to the amount distributed into the inlet. This can lead to either an overflowing collection basin or air entrainment into the suction piping. Design strategies to help prevent these problems include:

- making every effort to balance the flow into and out of each tower cell; and
- providing an "equalizer line" between sumps.

The equalizer line allows flow by gravity from one basin to the next. Since the force that moves the water through the equalizer line is the difference in water level between the sumps (which is sometimes just several inches), it is essential that the equalizer line be sized for a very low pressure drop. The equalizer line should be independent of the suction piping due to the various pressure differentials in the suction piping.

When just two cells are used, an equalizer flume weir gate is usually installed between towers. A removable cover simplifies maintenance on the sumps. When more than two cells are used, an external equalizer line with isolation valves is necessary.

Automatic isolation valves are required on the suction pipe of the cooling tower when the pressure drop of the equalizer line causes too great a fluctuation in water level between cells. Since this is difficult to accurately calculate, automatic isolation valves (or provisions for them) should be installed when three or more cells are used.

Start-up Conditions

After a condenser water system has been shut down, water will drain by gravity from the inlet piping into the cooling tower basin. Depending on the size of the lines, this volume could be enough to overflow the sump, thereby wasting valuable treated water.

Conversely, when the pump starts, there needs to be enough water in the sump to fill the empty piping without drawing the volume so low that air is entrained into the suction piping. The amount of water moved back and forth during start-up and shutdown must be minimized by:

- the judicious selection of pipe elevations;
- the use of automatic valves; and/or
- the selection of the proper size volume for the sump.

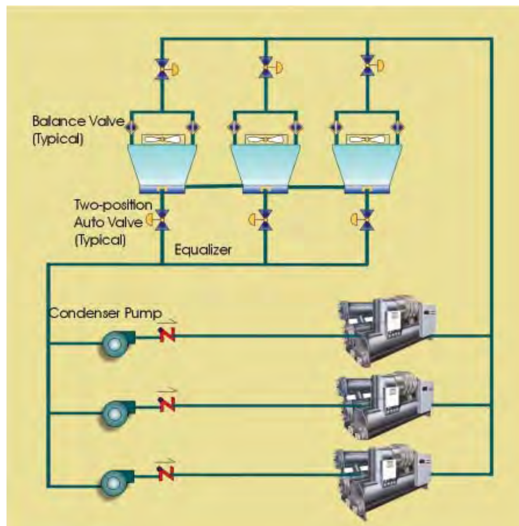
Another condition common at start-up is that the water temperature in the sump is too cold for the chiller. Chillers cannot operate at very cold inlet temperatures. When the machine requires warmer inlet temperatures than are available, three-way bypass valves are sometimes installed in the piping going to the tower. The bypass water is diverted into the cooling tower sump or into the suction piping. Care must be taken to install balance valves in the bypass line to create the pressure drop equal to the height of the cooling tower. To determine if a bypass valve is required, consult with the chiller manufacturer about minimum temperature requirements. Another option for cold start-up is to utilize a variable-speed drive on the condenser water pump.

Piping Option: Dedicated Condenser Pump per Chiller

One method of piping multiple condensers and cooling towers is to dedicate an individual pump for each chiller (Figure 4-23). This method has the advantage of closely matching the condenser water pump to the chiller, taking into account that the condensers may have different flow requirements or pressure drop characteristics. (In this configuration a dedicated cooling tower could be used for each chiller and pump combination, but the amount of piping to and from the cooling tower becomes prohibitive.)

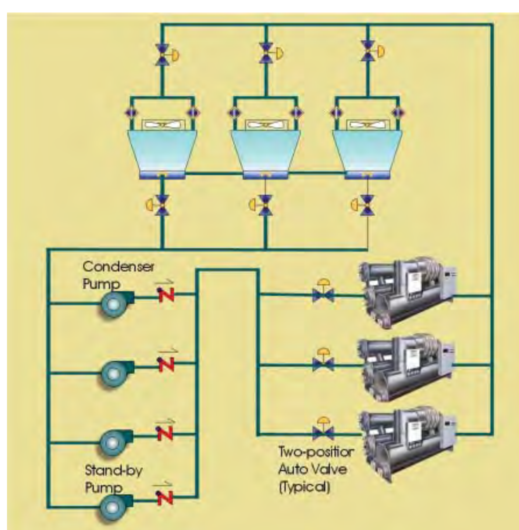
Since each condenser water pump shares the common piping to and from the cooling tower, the flow through the chiller will vary depending on how many pumps are operational. An analysis of pump curves and pressure drop characteristics may be necessary to determine if the flow variation is acceptable. This is one of the reasons that Chapter 3 recommends steep curve pumps for condenser water duty. If the flow variation is excessive due to long runs of piping (i.e., high piping pressure drop- see Example 4-3 for piping pressure loss calculation), flow control valves (constant volume regulators) are sometimes used in the condenser water circuits. But using flow control valves in this application may not be effective, as they can become fouled over time from dirt and debris in the condenser water.

Figure 4-23: Dedicated Pump per Condenser



In a multiple chiller arrangement a standby pump (Figure 4-24) is sometimes provided for redundancy. In this arrangement, adding the standby pump is cumbersome and requires a number of added valves.

Figure 4-24: Dedicated Pump per Condenser with Standby



Description	Flow (gpm)	Pipe Size	"K" Factor	Quantity or Length	Cv Rating	Delta P or Loss / 100 ft	Total
Entrance	1670	10				2.00	2.0
Piping	1670	10		6		2.58	0.2
Tee	1670	18	0. 44	2		0.04	0.1
Piping	1670	18		20		0.18	0.0
Tee	3340	18	0.07	2		0.03	0.1
Piping	3340	18		20		0.68	0.1
Tee	5010	18	0.07	2		0.05	0.1
Piping	5010	18		175		1.42	2.5
Piping	3340	18		10		0.68	0.1
Piping	1670	18		10		0.18	0.0
Piping	1670	10		8		2.58	0.2
Elbows	5010	18	0.22	3		0.18	1.8
Butterfly valves	1670	10		2	5750		0.4
Strainer	1670	10		1	1800		2.0
Reducer	1670	10	0.18	1		0.89	0.9
Total							10.5

Example 4-3: Piping Pressure Loss Calculation

Piping Option: Pump to Common Header with Isolation Valves

Another option for piping condenser water systems is to arrange the pumps to discharge into a common header (Figure 4-25) and to provide isolation valves at each condenser. As a chiller comes on-line another pump is energized and the isolation valve is opened. This works particularly well when more than three condensers are piped in parallel or the cooling towers and pumps are located remotely from the chillers. As in the previous arrangement, if the piping to and from the condensers is extensive (that is, a high pressure drop situation), the use of flow control valves at each condenser should be considered. One advantage of this approach is that any pump can serve any chiller. Therefore, if a pump and chiller are inoperative at the same time, the remaining operative chillers and pumps can be matched for greater efficiency.

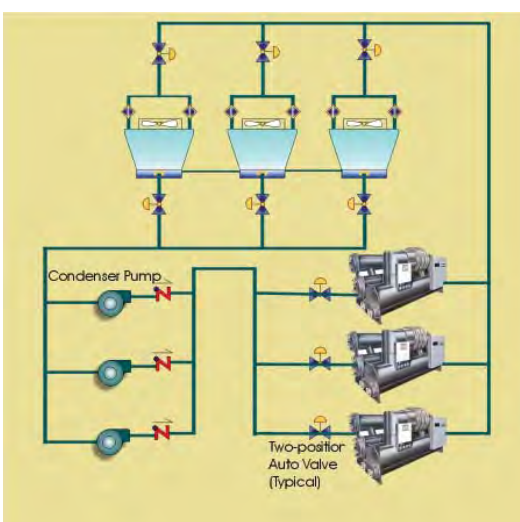
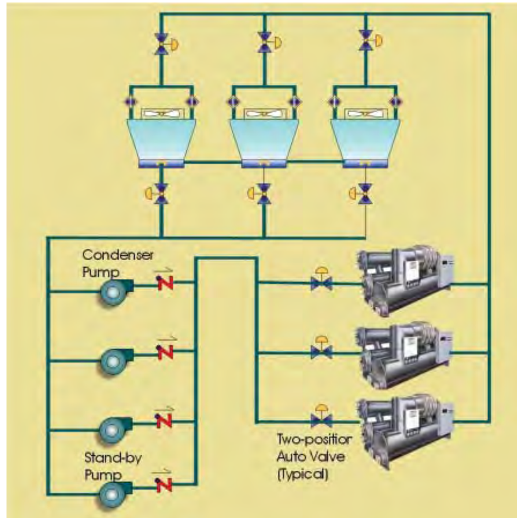


Figure 4-25: Headered Condenser Water Pumps

Adding a standby pump (Figure 4-26) is very simple with this configuration. For large plants with more than three chillers, the number of condenser pumps can be reduced (for example, one pump for two chillers), as long as variable-speed drives are added to the pumps and valves are controlled to maintain a constant differential pressure at the chiller.

Figure 4-26:
Headered Condenser Water Pumps with Standby



Piping for Waterside Economizers

A waterside economizer uses cold water generated at the cooling tower to produce chilled water or process chilled water. This is accomplished by running the cooling towers to produce water temperatures typically 45°F and less during periods of low ambient wet bulb temperatures. The cold water is pumped through a high effectiveness water-to-water heat exchanger, usually a plate and frame type, to produce chilled water at temperatures of 50°F or less. The heat exchanger protects the chilled water system from the corrosion, dirt and debris typical of condenser water.

For systems that do not require the simultaneous operation of the chillers and waterside economizer, the heat exchanger can be piped in parallel to the condensers similar to another chiller (Figure 4-27). But to improve performance at no additional cost, the waterside economizer should be piped to allow simultaneous operation of the chillers and economizer. (When the economizer is able to operate at the same time as the chillers, it is called an "integrated" economizer.) Figures 4-28A and 4-28B show how the economizer is piped for integrated operation. Integrated economizers are more efficient for two reasons:

- The economizer can provide some pre-cooling of the return chilled water temperature even if it cannot provide all of the cooling. With the non-integrated design (Figure 4-27), either the economizer provides the entire cooling load or none of it.
- The chilled water entering the heat exchanger is warmer in the arrangement shown in Figure 4-28A than that in Figure 4-28B, particularly for variable-flow systems. This improves the effectiveness of the heat exchanger.

The design in Figure 4-28A is used when the secondary pumps (or primary pumps on a primary-only system) have variable-speed drives. These pumps must be sized for the distribution pressure drop plus the pressure drop through the heat exchanger. When the economizer is not required or the towers cannot generate cold enough water to make the economizer useful, the valve labeled V-1 is opened, and the secondary pumps can slow down as the heat exchanger pressure drop is removed from the circuit.

The design in Figure 4-28B is used when the secondary pumps do not have variable-speed drives. In this case, the heat exchanger has a dedicated pump piped in a primary/secondary manner so that the pressure drop of the heat exchanger is not seen by the secondary pumps. The heat exchanger pump runs when the economizer is able to provide free cooling and is off otherwise. This design can also be used when the secondary pumps have variable-speed drives, but it will cost more than the design in Figure 4-28A and the effectiveness of the economizer can be reduced when the secondary circuit flow is less than the heat exchanger pump flow since that reduces the entering water temperature to the heat exchanger.

In both the integrated and non-integrated designs, the heat exchanger is generally not provided with its own condenser water pumps. Since the load will be low when the weather is cold enough for the towers to deliver cold water, it should not be necessary to run both chillers, so one of the chiller pumps can serve the heat exchanger.

When using waterside economizers, some type of chiller head pressure control is required because of the cold water coming off the cooling tower. This is true even for non-integrated economizers: even though the chiller and economizer will never operate simultaneously, the chillers will still have to operate with cold tower water at the changeover point where the economizer can no longer handle the load and the chillers are started. At that time, there is usually too much mass of cold water in the system for the chiller to warm it up before it trips on low head pressure.

There are several ways to provide head pressure control, but the least expensive and most effective is generally that shown in Figures 4-28A and 4-28B. The automatic isolation valve (which is used to shut-off flow to the chiller when it is off) doubles as a head pressure control valve, throttling flow through the chiller to maintain head pressure. (This is usually done by taking a signal from the chiller control panel directly, but head pressure can be controlled indirectly by controlling the water temperature leaving the chiller.) Another option for head pressure control is to vary the flow through the condenser by modulating the speed of the condenser water pump.

In most climates, the airside economizer will be more efficient than the waterside economizer since it does not require the operation of tower fans and chilled and condenser water pumps. Also, the approaches of the many heat exchangers in the waterside economizer system (condenser water temperature to outdoor air wet-bulb temperature, chilled water temperature to condenser water temperature, and supply air temperature to chilled water temperature) usually result in fewer hours of waterside economizer operation than airside economizer operation. Outdoor air economizers can also provide improved air quality by increasing outdoor air rates. On the other hand, waterside economizers can be more efficient than airside economizers in hot, dry climates (e.g., Palm Springs) due to the very low wet-bulb temperatures, and they are ideal for applications requiring wintertime humidification (e.g., data centers).

Figure 4-27:
Water Side Economizer,
Non-integrated

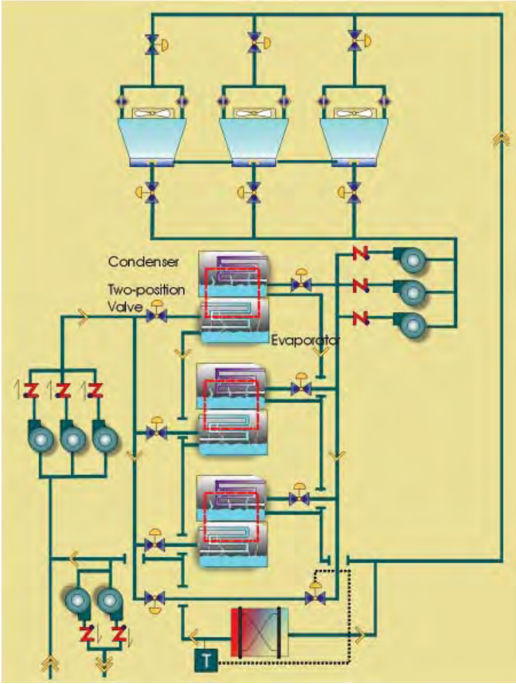
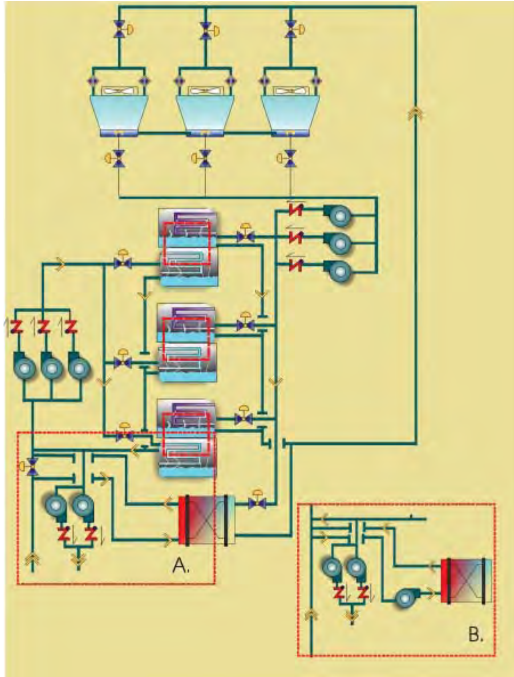


Figure 4-28:
Water Side Economizer,
Integrated



Auxiliary Condenser Water Circuits

In some facilities with chilled water plants, condenser water is used for purposes other than the chillers, such as process cooling water. Because of the poor quality of the condenser water, it must be determined whether a heat exchanger should be incorporated into the process cooling water circuit or whether the condenser water can be used directly. If a heat exchanger is used, it can be piped into the condenser water circuit in parallel with the chiller condensers, similar to another chiller.

If the water can be used directly in the process loop, a primary/secondary piping arrangement may be most suitable. In this configuration the secondary loop can be constant or variable flow. The pressure drop is independent of the primary loop. In cases where the requirements for the auxiliary water circuit are fairly low, care must be taken in selecting the flow through the primary circuit to ensure that the flow is equal to or greater than the minimum flow of one cooling tower.

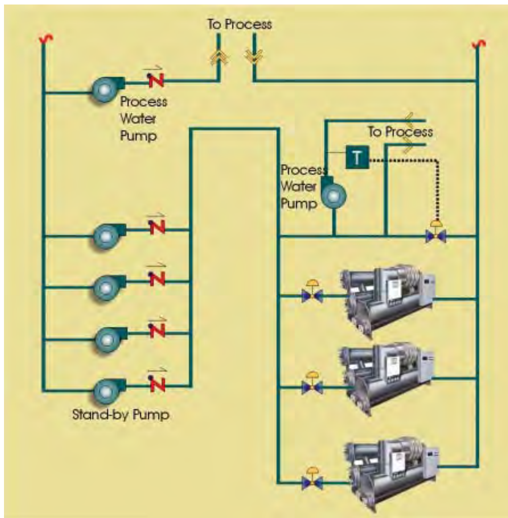


Figure 4-29: Auxiliary Condenser Water Loop

Piping Heat Recovery Options

Heat rejected from chillers can be used in numerous ways, including preheating domestic hot water and—with the use of double-bundle heat recovery chillers—heating buildings. In the case of preheating domestic hot water, the condenser water is routed through a double-wall heat exchanger that is either an integral part of a storage tank or is remotely located with a circulation pump to the storage tank.

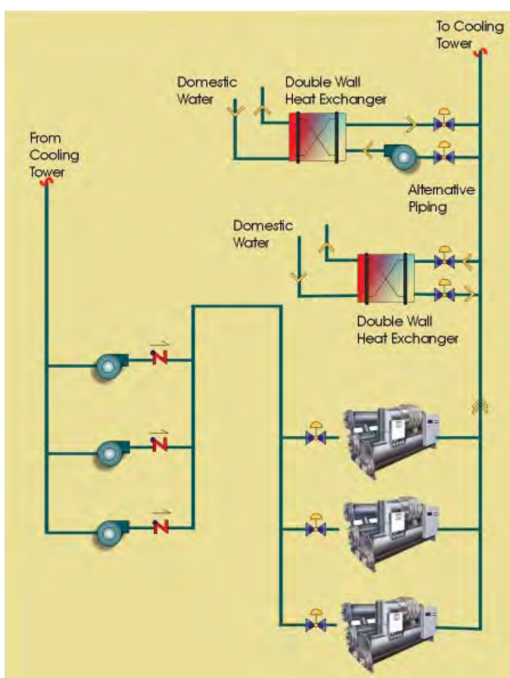
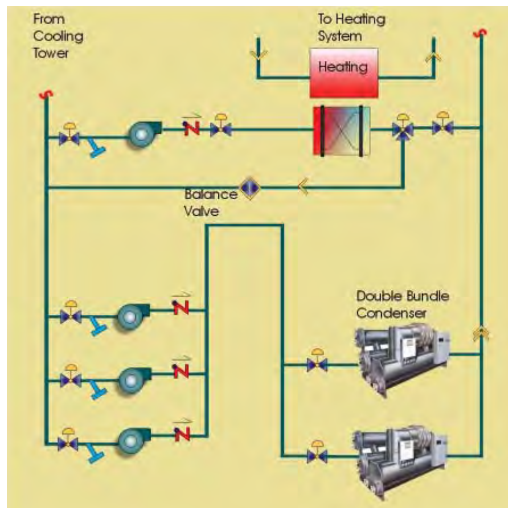


Figure 4-30: Preheat of Domestic Hot Water

For heat recovery chillers, the piping of the cooling tower side of the double-bundle condenser involves using a three-way valve that controls the water temperature leaving the heat recovery side of the double-bundle condenser. In a double-bundle heat recovery condenser, the hot gas from the compressor first enters the heat recovery side of the condenser where the building's heating system removes the heat at a suitable temperature (105°F to 130°F). If all of the heat from the chiller is not rejected in the heat recovery bundle, the leaving heating water temperature (and refrigerant pressure) will rise above set temperature. This will cause the temperature controller to modulate the three-way valve on the cooling tower side of the condenser to maintain set temperature. A balance valve must be provided on the bypass line that goes back to the pump suction.

FIGURE 4-31:
PIPING FOR DOUBLE-BUNDLE HEAT RECOVERY CHILLER



Heat recovery chillers have limited unloading capability when in the heat recovery mode due to the high condensing temperature level. If the cooling load is small relative to the design chiller capacity, hot gas bypass must be used to prevent surge, severely increasing energy usage. Therefore, where heat recovery chillers are used, it is important to install one or more non-heat-recovery chillers. The more efficient non-heat recovery chillers can be operated at low loads when there is insufficient load to keep the heat recovery chiller on without hot gas bypass. They can also be run when the cooling load exceeds the capacity of the heat recovery chiller.

Using chiller heat recovery for space heating and using economizers (air or water) are generally mutually exclusive because the economizers will keep the chillers from operating in cold weather so there is no condenser heat to recover. During cold weather when the heating load is equal to or greater than the amount of heat rejected from the chillers, it can be shown that using heat recovery chillers will be more energy efficient than airside economizers and gas-fired heating systems. In most commercial buildings, the cooling load will be small in cold weather since only the interior zones need cooling and their loads are usually relatively small. In this case, heat recovery will be more efficient than economizers. But as the weather gets milder, heating loads get smaller while cooling loads can get larger as some sunny perimeter zones switch from requiring heating to requiring cooling. At this point, economizers begin to outperform the heat recovery system. On an annual basis, economizer systems tend to be more energy efficient in mild climates because:

- The heating season is relatively short and mild.
- Integrated economizers reduce energy usage even when heating is not required. For instance, in mild weather (55°F to 65°F), integrated economizers will reduce the cooling load, which can be substantial since both interior and perimeter zones will require cooling in this case, while a heat recovery system will do little since heating loads are very small or nonexistent. In California's mild climate, a substantial number of building operating hours fall into this temperature range. (In very mild climates like San Francisco's, more than 85% of the operating hours fall into this range.)

As noted above, heat recovery chillers have limited unloading capability when in the heat recovery mode, often requiring the use of hot gas bypass. When doing so, the chiller is essentially acting like a costly electric resistance heater.

Heating systems using recovered heat must be designed for low temperatures (e.g., 110°F) and low temperature differences because of the limits of the heat recovery condenser. This will increase hot water flow rates and may require larger heating coils, increasing both air and waterside pressure drops and thus increasing fan and pump energy.

If there is a large constant heating load, such as that for domestic hot water in a hotel, heat recovery will probably outperform economizer systems. A detailed computer analysis would be required to evaluate the two design options in this application. It is important to include maintenance costs in the analysis since heat recovery systems require the chiller to operate all day long, all year long, increasing the maintenance costs and reducing the service life of this expensive machine.

It is possible to combine economizers and heat recovery to maximize energy savings. The heat recovery mode is used, with economizers locked out, when the heating load is large enough (e.g., when OAT<55°F) and the economizer mode is used during milder weather. However, the first costs of this design can be prohibitive so it is seldom used.

5. CONTROLS AND INSTRUMENTATION

Introduction

Chapter 5 explores the many design and performance issues that design engineers face when selecting *controls* and instrumentation for chilled water plants. This chapter begins with a discussion of some general factors that should be considered when designing and installing *control systems*, including:

- The effect of the plant's operating environment on control and instrument performance;
- The importance of finding the optimal level of instrumentation; and
- Guidelines for selecting control and monitoring points.

To maintain stable and effective plant operation, designers must ensure that the instruments are of high quality and are sufficiently accurate and reliable. Also, the chilled water plant's control system must have sufficient programming and data manipulation capabilities. This chapter covers in detail the following topics related to controls and instrumentation in chilled water plants:

- Types of sensors available for energy monitoring and control
- Styles of and selection criteria for control valves
- Controller requirements and interfacing issues
- The importance of performance monitoring
- Types and configuration of local instrumentation
- Control sequences for chilled water plants

Choosing Instrumentation Control and Monitor Points

If the *control* and *monitor* instrumentation points are not carefully chosen, the system may suffer from either under- or overinstrumentation. Either of these can be a serious problem, though they may affect different aspects of the system's operation. An underinstrumented system may be difficult to control optimally, whereas an overinstrumented system may be confusing to the operations staff, expensive, and difficult to maintain. Although designers may be tempted to provide more rather than fewer control and monitoring points, it's important to remember that additional points do not guarantee that the system's performance will be acceptable.

Point Justification

The selection of *control* and **monitor points** should be based on a careful analysis of the chiller plant's control and operating requirements. Each point must meet at least one of the following criteria:

- It must be necessary for effective control of the chiller plant as required by the sequence of operations established for the plant;
- It must be required to gather necessary accounting or administrative information such as energy use, efficiency, or run time; or
- It must be needed by operations staff to ensure that the plant is operating properly or to notify staff that a potentially serious problem has or may soon occur.

Because each plant has individual operating requirements for staffing, accounting, operations and maintenance, there is no single standard for determining exactly what instrumentation is required for any plant. Consensus in the industry is that a metric of total plant water-to-wire efficiency is desirable. ASHRAE is working on a guideline (GP 22P) GPC Instrumentation for Monitoring Central Chilled Water Plant Efficiency to detail how to achieve this. The purpose of each point should be considered. For example, points such as supply and return chilled water temperature would require a higher accuracy if they are used to track chilled water production, than if they were just used to control chiller operation. Other points may require certain additional equipment such as power transducers that are needed to calculate and trend power use. All aspects of each point should be carefully considered to ensure desired performance. Work is currently underway to develop guidelines for accuracy requirements for chilled water plant instrumentation used for various functions. Some of the initial results of that work for chilled water plant operation and control are presented in this Guide.

Some equipment, such as chillers and variable-speed drives, have built-in controls. It may be possible to access the built-in control and monitoring points and avoid the cost of installing redundant control points as part of the energy management system (EMS). For instance, the chillers have controls that will include chilled and condenser water temperature sensors. If data from these sensors is accessible to the EMS system, the installation of additional sensors can be avoided. Accessing these built-in points generally requires an interface or "*gateway*" between the EMS and the built-in equipment controls. In most instances, the cost of the gateway is lower than the cost of installing redundant sensors, particularly when multiple equipment of the same type (e.g., chillers) are installed, since only one gateway is usually required. See the later section, Interfacing with Chillers and Chiller Networks.

Operating Environment for Controls and Instrumentation

Chillers and other plant equipment generate heat and vibration that may adversely affect controls and instrumentation. Also, routine maintenance may expose certain areas in the plant to substantial water and debris. The controls and instrumentation should be located away from areas where they might be subject to temperature or mechanical damage during plant operation and maintenance activities. If such locations are not available, then the equipment must be protected from adverse environmental conditions.

Network Connections to Equipment Instrumentation

There is a strong trend for plant equipment to incorporate factory-installed instrumentation that can be accessed through network connections. This enables control and monitoring functions to occur over the network. In general it is good practice to limit these network connections to monitoring and not control of critical components as network connections often drop out or experience momentary loss of connection. This is particularly important for plants that serve mission critical loads like manufacturing, hospitals, and data centers. To ensure that network connections meet the requirement for data accuracy and time response, the specification must clearly state that the requirements apply to control/monitoring point to the network connection. The specification should also be specific as to which points should be hardwired and which can be transferred via network. The three most critical points for equipment are start/stop, setpoint (speed or temperature) and summary alarm. It is prudent to consider hardwiring these points. With all network connections it is important to make sure to coordinate the work of the equipment manufacturer and the EMCS contractor in the specifications. Critical details include the network communication language (e.g., BACnet), physical link (e.g., RS 485) and points to be mapped across from the network device to the EMCS. Be aware that many gateways only allow a limited number of the available points to be mapped to external EMCS.

Sensors

Sensors are used in chilled water plants for a variety of measurements, including:

- Temperature
- Humidity
- Liquid Flow
- Pressure
- Electric Current
- Electric Power
- Gas Flow

The sections below discuss the end-to-end accuracy requirements and the types of sensors available for each type of measurement, their measuring characteristics, and special factors to consider. Installation and calibration information is also provided.

Typical End to End Accuracy Requirements

Typical and suggested end-to-end accuracies for common chiller plant control points are shown in Table 5-1. These accuracies are readily achievable with common commercial DDC systems and are usually all that is required for acceptable plant performance. Increased accuracy may be achieved but usually at added cost. Additional discussion on accuracy requirements is included in specific sensor sections.

Table 5-1: Typical End-Use Accuracy

Measured Variable	Reported Accuracy
Outside Air Dry-Bulb Temperature	±1°F
Chilled and Condenser Water Temperature at Central Plant Mains	±0.2°F
Chilled and Condenser Water Temperature Elsewhere	±0.5°F
Water Delta-T (supply to return)	±0.15°F
Relative Humidity	±5% RH
Water Flow	±1% of full scale
Water Pressure	±2% of full scale
Electrical power	1% of reading, 3kHz response for VSD driven equipment

Temperature Sensors

Types of Sensors

Thermistors and Resistance Temperature Detectors (RTDs) are the most common devices used for temperature measurement in chiller plants. Both are actually RTDs, although thermistors typically have a negative resistance/temperature characteristic with respect to temperature. Thermistors and RTDs each have advantages and disadvantages, but when properly applied either may be used to measure temperature. The following issues may impact the choice of temperature sensor:

- ***Direct digital control (DDC) interface***. An interface is usually required to link RTDs to most DDC controls. This is because the resistance change in response to temperature for RTDs is too small to input directly to typical digital controls. The interface is an external transmitter circuit that is included with most RTDs. The output of this circuit is linear and usually available as an industry standard such as 4-20 ma or 0-5 vdc. The interface reduces uncertainty when the sensing instrumentation and the control systems are supplied by different manufacturers. Some DDC systems are able to take data directly from RTD devices, obviating the need for a transmitter. Such DDC systems usually work with certain "standard" thermistors. The connection is easy, inexpensive, and accurate, because the preset scaling ranges on these systems can automatically adjust for the non-linear scale. For such DDC systems, thermistor temperature sensors are very cost effective.
- ***Cost***. RTDs generally are more expensive than thermistors.
- ***Precision***. Since the resistance change in response to temperature change is generally higher for thermistors than RTDs, they are slightly more precise. However, the added precision is well beyond the limits of most air- and water-sensing applications. For chiller plants, both RTDs and thermistors provide adequate precision.
- ***Long-term stability***. RTDs are more stable in the long-term than thermistors. However, the external transmitter circuit on many RTDs can cause readings to drift. At elevated temperatures, the stability of thermistors is reduced. However, at temperatures typically found in chilled water plant systems, both RTDs and thermistors are very stable.

The following table compares the features of RTD and thermistor temperature-sensing devices:

Table 5-2: Comparison of RTD and Thermistor Temperator-Sensing Devices

Type	Range	Cost	Stability	Sensitivity	Linear
RTD	-260 to 650°C	Moderate	High	Moderate	Yes
Thermistor	-80 to150°C	Low	Moderate	High	No

Measuring Characteristics

Both *accuracy* (the ability to measure an actual value) and *resolution* (the ability to sense changes in the value) need to be considered together in selecting temperature sensors. If accuracy is considered without looking at resolution, or vice versa, the measurement objective may not be satisfied. The temperature-measuring device is among the strongest factors in determining the overall integrity of the measurement requirement. Other factors are:

- the means of transferring the signal from measurement device to control system;
- the signal span and number of *bits* used in the control system's *analog-to-digital* converter; and
- the *change of value threshold* and/or timing required for the networked point reporting or for the control system to notify programs, displays, etc. that a value change has occurred.

Whenever possible, it is useful to specify "end-to-end" measurement property requirements. When using temperature measurements for control functions, accuracy is based on the sensitivity of the plant's operating cost to potential errors in the measurements. When using temperature measurements for load or efficiency calculations, measurements would generally need to be more accurate, in order to get meaningful results. For example, to determine cooling load, supply and return chilled water temperatures would need to be about 0.1°F accurate, in order to calculate cooling load within 2%. For many control points, however, 0.5°F to 1.0°F end-to-end overall accuracy is quite adequate.

Indirect well temperature sensors are recommended for measuring water temperature, so that the sensor may be removed from the well for calibration or testing. The wells should be stainless steel. Here are recommended guidelines for installing well temperature sensors:

- Place the temperature sensor in a well that penetrates the pipe by the lesser of half the pipe diameter or four inches.
- Install the sensor in the well with a thermal-conducting grease or mastic.
- Use a *closed-cell* insulation patch that is integrated into the pipe insulation system to isolate the top of the well from ambient conditions but allow easy access to the sensor.
- Locate wells far enough downstream from regions of thermal stratification or mixing so that the fluid's temperature is uniform at the well.
- For field calibration and testing, consider using a sensor and well combination that allows insertion of a field test element alongside the sensor, or a second well or test tap adjacent to the sensor well for this purpose.

Special Considerations when Measuring Differential Temperature

Where sensors are used to measure differential temperatures, special attention must be paid to measurement accuracy. The sensors should be located near one another so that field calibration or testing can be done simultaneously using the same thermal test medium.

"Matched" temperature sensors are available with almost exactly duplicate temperature/output characteristics. Alternatively, field calibration can be used to equalize sensors that are not perfectly matched.

When measuring load, consider using a manufacturer's BTUH meter (e.g. Onicon's System-10 or System-30, http://www.onicon.com/Btu_meters.shtml). These meters sample the temperatures and flow at a much higher rate than the EMCS to reduce sampling error of the calculated heat flows. BTUH meters typically provide access to the individual sensor measurements either as hardwired points or through a gateway.

Calibration

There are two approaches to calibration and each has merit depending on how critical the measurement applications are:

- ***Pre-calibrated***. For non-critical measurement applications, it may be satisfactory to rely on the manufacturer's "*temperature tolerance*" limit for the sensor supplied. Factory-established temperature tolerance limits generally range about +/-0.5oF for RTD and thermistor sensors. Some manufacturers use the term "pre-calibrated" to refer to the *tolerance limit*. However, this pre-calibration does not account for all the factors involved in the end-to-end measurement characteristics of the sensor (see Measuring Characteristics above).
- ***Field-calibrated***. It is recommended (and *required* for critical measurement applications) that each temperature sensor be field calibrated as part of the commissioning and startup process (see Chapter 8 for information on commissioning). Because of the limited *rangeability* of most chiller plant temperature-sensing requirements, a single-point calibration is often adequate. Field calibration procedures can be combined with inspection and testing procedures. An end-to-end field calibration of all devices can enhance the overall accuracy of the system.

Humidity Sensors

Types of Sensors

A wide variety of sensors are available to measure humidity. In chiller plants, the outdoor wet-bulb temperature are sometimes used either for operating sequences or efficiency calculations. Humidity sensors are subject to error due to sensing technology, hysteresis and drift. The Iowa Energy Center did a definitive study of humidity sensors comparing their performance in both test chambers and in the field to highly accurate laboratory grade references. This study is posted on the website http://www.energy.iastate.edu/Efficiency/Commercial/nbcip.htm. Of the sensors tested, the gold standard was the Vaisala HM series transmitter. The bulk of the sensors tested did not even meet the manufacturer's printed specifications.

For outdoor humidity-sensing applications, it is important that the sensor be specifically designed for outdoor use and that it be protected from direct sunlight and other adverse effects (follow the manufacturer's recommendations). Some sensors designed for outdoor service can be purchased with a protective enclosure.

Measuring Characteristics

The recommended accuracy specification is: (% RH), NIST Traceable and certified at 77°F over 20-95% RH including hysteresis, linearity and repeatability for either 3% or 2%.

Installation

For outdoor humidity-sensing applications, the humidity sensor should be installed along with an outdoor air temperature sensor in a location where they are both always protected from direct sunlight and from the building's air exhaust. They should also be in a well-ventilated area. For certain buildings or facilities, it may make sense to use multiple sensors; logic in the control system can determine which values to use.

Calibration

When humidity sensors are used to control a cooling tower or other chiller plant equipment, the outside air humidity sensor must be field tested initially and at intervals recommended by the manufacturer to ensure that it is providing an accurate signal. It is recommended (and *required* for critical measurement applications) that the outside air humidity sensor be field calibrated as part of the commissioning and startup process (see Chapter 8 for information on commissioning). Also, a portable humidity-sensing device should be used to check the accuracy of the humidity sensor at least at the start of the cooling season each year.

Liquid Flow Sensors

Types of Sensors

Fluid flow measurement can be a difficult and costly instrumentation item for a chiller plant. Flow meters are expensive to install and set up, and their placement requires great care in order to obtain good flow readings. Also, the cost of keeping flow meters calibrated and fully operational may be high.

For chilled and condenser water applications we recommend use of magnetic flow meters: full bore for new construction in pipes 12” or smaller and insertion for pipes larger than 12”. These meters are highly accurate and low maintenance as shown in the table below. Of all of the meters, full bore magnetic meters are least susceptible to errors from variations in flow profile or the presence of particulates or air in the pipe. Full bore magnetic meters, ultrasonic magnetic meters and insertion magnetic meters have no moving parts in the water stream that can get fouled.

Orifice meters require a pressure drop, which increases energy use, and therefore they are not recommended. *Paddle*, turbine, and to a lesser extent vortex meters are subject to fouling and mechanical failure. An ultrasonic meter may provide erroneous readings at times if air or other particles pass through the meter.

Table 5-3 compares various types of liquid flow sensing devices:

Table 5-3:
Comparison of Liquid Flow Sensing Devices

Type of Flow Meter	Range of Flow (turndown ratio)	Relative First Cost	Measurement Accuracy	Maintenance Costs
Orifice	Low (5:1)	Medium	Low	Medium
Insertion paddle	Low (10:1)	Low	Low	Medium
Insertion turbine	Medium (30:1)	Medium	Medium	Medium
Vortex meter	Medium (30:1)	High	High	Medium/low
Insertion magmeter	High (50:1)	Medium	High (± 1% of reading from 0.25 to 20 fps)	Low
In-line ultrasonic	Highest (1000:1)	High	High (+/-2%)	Low
In-line magmeter	Highest (1000:1)	Medium up to 12” pipe High > 12”	Highest (+±0.5% of actual reading from 3 to 30 feet per	Lowest

Measuring Characteristics

Generally, the most important flow measuring characteristics are range and accuracy. Design specifications must include the required range of flow velocities. Also, the design engineer must fully understand the expected error at various flow levels to be certain the device chosen will meet all requirements.

Installation

The manufacturer's requirements for placement and installation must be carefully observed to ensure an accurate flow measurement. Because flow measurements often require specific upstream and downstream lengths of undisturbed straight piping, it may be necessary to pay special attention to the piping layout during plant design and construction.

Special Considerations

Where flow meters are intended to measure flow that may be bi-directional (e.g. installations in a common leg or for a TES tank), special attention must be paid to the design and equipment layout and specifications so that both the flow and direction can be determined with acceptable accuracy.

Calibration

Calibrating flow meters in the field can be very difficult and, more often than not, is impractical or impossible. Calibration of any device requires field measurement of the measured variable with a device that is substantially more accurate than the device being calibrated. With flow measurements, it is often hard to find an acceptable location for the flow meter, one with adequate length of straight piping both up and downstream of the device. It is therefore very seldom possible to locate an additional, temporary flow sensor in the system in order to calibrate the installed sensor. Strap-on ultrasonic meters are often considered, but they are very difficult to properly use and, at least with current technology, are not usually more accurate than the installed flow meter and therefore not appropriate for use in calibration. Use of pump curves, differential pressure across known devices (such as chillers), or calibrated balancing valves at pumps or coils also are not accurate enough for calibration. However, all or some of these can be used as a "reality check" to verify that the flow meter is reading in the proper range, and often that is all that can be done in the field. Therefore it is important that flow meters be accurately factory calibrated. A minimum of three-point factory calibration should be required for each flow meter. For large campuses or plants where the accuracy of the meters is important you can also use a 3rd party laboratory for independent testing of flow meters. At the time of this writing there are facilities at both the Colorado Engineering Experiment Station (CEESI, http://www.ceesi.com/), The Southwest Research Institute (http://www.swri.org/) and the Utah State College of Engineering (http://uwrl.usu.edu/facilities/hydraulics/index.html).

Field test procedures can be combined with inspection and testing or balancing and commissioning procedures to minimize costs and duplication of effort. (See Chapter 8 for information on commissioning.)

Pressure Sensors

Types of Sensors

Pressure sensors for water-sensing applications have special requirements. Because water is not compressible and is high in mass, transient pressures resulting from valves opening or closing and pumps starting and stopping can be severe at certain times and places in the system. Also, turbulence—especially near pump discharges—may cause pressure measurement fluctuation.

The two most common means of sensing pressure for fluid conditions are *capacitance* and *piezoresistive* sensors. The piezoresistive sensor employs a small solid-state device whose resistance changes with pressure, while the capacitance sensor measures pressure by the strain of a small diaphragm, which is usually stainless steel. Generally the piezoresistive sensor is less sensitive to mounting vibration or shock but less tolerant of pressure transients in the fluid it is sensing. Both types are widely used in water pressure monitoring applications.

Measuring Characteristics

Generally, the most important pressure-measuring characteristics are the sensor's pressure range and the accuracy of its pressure measurements. It is important to include the required pressure range in the specifications for each pressure-sensing device. The maximum *operating pressure* as well as *burst pressure* are also important considerations for sensors that are used with high static pressures and/or high dynamic loads. Most manufacturers offer a variety of pressure ranges for their sensors, and many permit splitting the range in differential pressure-sensing applications so that the sensor can measure bi-directional pressure changes. Standard commercial grade sensors offer excellent accuracy, usually 1% or less of the specified pressure range.

Recommended Specifications

Pressure sensors are typically used to control pump speed in variable flow applications. If the sensor is inaccurate or drifts it can either waste energy by maintaining too high a pressure or starve valves if the pressure goes too low. The following specifications are based on a fast response capacitance sensor similar to the Setra 230 or Kele & Associates Model 360C:

- Overall Accuracy (at constant temp) ±0.25% full scale (FS).
- Non-Linearity, BFSL ±0.20% FS.
- Hysteresis 0.10% FS.
- Non-Repeatability 0.05% FS.
- Long Term Stability 0.5% FS per year

Installation

The *pressure taps*, lines and sensor should be installed in a location where they are accessible but will not interfere with access to pumps, valves, chillers or other equipment. Also, the sensor should be mounted in a location that is not subject to physical harm, such as mechanical shock or water damage. Special attention should be taken to ensure that valves are installed on all taps so that each sensor can be easily isolated for maintenance or testing. Sensitive differential pressure sensors may require a *cross-feed manifold* connection to equalize the differential pressure while connecting or disconnecting the sensor.

Special Considerations

It may be desirable to monitor the "gauge" pressure (in which case the reference pressure is the ambient air) or a differential pressure (in which case the reference pressure is another wet source). The exact nature of the sensor requirements should be clearly noted in the specifications with a precise description of the location of the tap or taps. Whenever possible, the tap locations should be shown on drawings, and the *taps* should always be located as close together as practical.

When measuring the pressure across a pump it is desirable to use the pump taps provided by the pump manufacturer rather than taps on the pipe. The reason is that the pump taps correspond to the manufacturer's published pump performance and can be used to deduce the pump impellor through a "dead head" test.

Calibration

Generally, factory calibration of pressure sensors is adequate for most pressure measurement applications. It is recommended that each pressure-sensing device be field tested during startup and balancing to confirm its accuracy at both zero pressure and at least one typical pressure condition. Field test procedures can be combined with inspection and testing or balancing and commissioning procedures to minimize costs and duplication of effort (see Chapter 8 for information on commissioning).

Electric Current Sensors

Types of Sensors

The universal means of sensing AC current is the *current transformer* or CT. As current passes through this device a small voltage is generated which is proportional to the current that is being measured. There are both digital and analog versions of electric current sensors. The digital sensor (usually called a current switch) provides a binary signal (contact closure) as long as the current is above a preset value. The analog sensor (usually called a current sensor or current transducer) provides an analog signal (usually 0-5 vdc or 4-20 ma) that can be scaled to read the current draw.

Measuring Characteristics

Although an analog *current transducer* is somewhat more expensive than a digital version, the analog signal provides much better information for the operator and is recommended over the digital version for most current-sensing applications. In addition to on/off status, the analog sensor can also be used to provide alarms at overcurrent as well as undercurrent conditions. For equipment with a relatively constant power factor, analog current sensors can be used to estimate power consumption. However, for major equipment in chiller plants, it is recommended that a true rsm power transducer be used for that purpose.

While the analog sensor provides more information, the digital sensor or current switch is less expensive and very commonly used where simple on/off status is all that is needed. It is almost always a better status indicator than a differential pressure sensor or flow switch for several reasons:

- It is less expensive due to substantially lower costs for installation and wiring. DP (differential pressure) and flow switches require installation into the piping (or duct) and are usually a long way from the DDC panel. Current switches are more easily installed, particularly those with split core CTs, and are mounted in the starter panel, which is usually close to the DDC panel.
- A DP switch for a pump (or fan) may indicate pressure when in fact there is no flow. This can happen when a valve (or damper) is close but the pump (or fan) is still on. The current switch setpoint can usually be set to a current higher than the power draw when the pump (or fan) is dead-headed (no flow).
- The current switch setpoint can also be set to a current higher than the motor's no-load current so that it still can be a reliable indicator of a coupling (or belt) failure.
- Current switches are solid-state devices with no moving parts or diaphragms, and are therefore much more reliable than DP switches or flow switches. They also require no maintenance and last longer.

Because of these advantages, there is seldom a need for DP switches as status indicators in chilled water plants.

Installation

For current-sensing applications, the CT should be located in the motor control center (MCC) of the plant whenever possible. Sensing equipment should be mounted so that it is accessible and does not block access to other devices.

Special Considerations

In three-phase applications, a CT is mounted only on one leg of the power. The motor starter provides single-phase protection and will automatically shut down the motor if one phase is lost.

Calibration

Factory calibration of AC current sensors is adequate. However, many AC current sensors have multiple ranges that are set with small switches or jumpers on the device. It is recommended that field test procedures be combined with inspection and testing or balancing and commissioning procedures to ensure that the equipment is correctly connected and that each device is properly set and scaled.

Electric Power Sensors

Types of Sensors

Two major types of sensors are often used for power monitoring. The "kW demand" sensor provides an analog output (usually 0-5 vdc or 4-20 ma) that indicates the instantaneous rate of electricity use. The "kWh consumption" sensor provides a pulse signal that indicates the number of kilowatt-hours of electricity that have flowed since the last pulse. Both types of sensors have the same components. For three-phase applications, these sensors include current transformers and voltage measurement taps for each leg.

Newer sensors transmit monitored information over a network. This allows multiple meters to be connected together on the network. A single port connection links the meters to the plant monitoring and control system. Networked meters are more expensive than non-networked sensors but may reduce interfacing costs when multiple meters are needed. The *networked sensor* usually provides both kW and kWh information, and often much more (see Special Considerations below).

Measuring Characteristics

The chilled water plant designer must determine whether the application calls for a sensor that provides kW, kWh or both types of data. An instantaneous kW reading is most useful for determining the status of the equipment whose power it monitors. It is also useful if the plant operations staff will require a continuous, real-time chilled water production efficiency reading (such as a COP or kW/ton).

It is possible for the controller to integrate kW readings to develop periodic kWh data. However, if a kWh reading is required for billing purposes, it is best provided by a kWh reading taken directly from the meter or by a pulse output sent to the DDC system. Because this data contains continuous time integration, these measurements will be more accurate.

Most kW and kWh meters provide better than 2% accuracy, which is suitable for verifying the plant's energy use. However, in applications that involve monitoring the power of motors that are operated by variable-frequency drives or other wave distorting equipment, accuracy may be reduced unless the power sensor provides *true RMS* power sensing.

Installation

For power-metering or power-sensing applications, the sensing equipment should be located in the motor control center (MCC) of the plant whenever possible. Sensing equipment should be mounted so that it is accessible and does not block access to other devices. Care must also be taken to ensure that sufficient ventilation is provided so that the manufacturer's temperature limits for the equipment are not exceeded.

Special Considerations

In addition to power use data, some power-sensing devices can also provide power factor, kVAR, VA and harmonic information. This data may be very useful in many facilities, especially if power quality is a concern. If the sensor can be interfaced to the plant control and monitoring system via a standard network, the increased cost to obtain this additional information is modest. The designer should detail the exact nature of the sensor and data requirements in the specifications.

Although there is no industry standard for "true RMS" sensing, there is agreement in the community that a minimum sampling or response rate of 3kHz is required to get accurate measurement of non-linear loads like variable speed drives.

Calibration

Factory calibration of electric kW and kWh sensors is nearly always adequate. It is recommended that field test procedures be combined with inspection and testing or balancing and commissioning procedures to ensure the equipment is correctly connected and the data is properly scaled for the correct readings.

Gas Flow Sensors

Types of Sensors

There are several types of gas flow sensors that can monitor the natural gas flow of absorption or gas engine-driven chillers. The *diaphragm gas meter* is most widely used (this is the type that most utilities install as site meters). Other gas flow sensors are *rotary* and *turbine* meters; these are generally used when the maximum gas flow requirements exceed the capacity of a diaphragm meter. Another type of gas flow meter—the *orifice* meter—is occasionally used for very high rates of gas flow, but because of accuracy and *turndown limitations*, it is not recommended for monitoring chiller plant energy use.

Table 5-4 below compares various options for gas flow-sensing devices.

Table 5-4: Comparison of Gas Flow Sensing Devices

Type of Gas Meter	Range of Flow (SCFH)	First Cost	Rangeability (turndown ratio)
Diaphragm	Up to 5,000	Low	100:1
Rotary	100 to 50,000	High	40:1
Turbine	1,500 to 200,000	Medium	15:1
Orifice	5,000 up to 100 mm	Low	3:1

Measuring Characteristics

Generally, the most important factor in choosing a gas flow meter is range of the flows that the meter must measure. It is also essential that the meter be suitable for the gas pressure to be used. The required range of gas flow volumes needs to be clearly specified by the designer.

Installation

Because diaphragm and rotary gas flow meters measure volume, not velocity, their placement is far less critical than turbine and orifice meters. For turbine and orifice meters, the manufacturer's requirements for placement and installation must be carefully observed. Volumetric meters are generally recommended for most chiller plant applications. The metering application should always be discussed with the utility supplying the gas. It is sometimes necessary to monitor the pressure in order to improve the measurement accuracy.

Special Consideration

Where gas is provided to a chiller plant—for absorption or engine-driven chillers and perhaps a boiler or electric generator—consider using the utility's gas meter with an auxiliary transmitter to provide data for control and/or monitoring. This is often the most economical choice and can be used if other equipment will not operate simultaneously with the chiller plant's gas-fired equipment.

Calibration

Typically diaphragm and rotary gas meters are fully factory tested and calibrated and require no further calibration. As part of the commissioning and startup process, turbine meters should be field tested to confirm accuracy at as many flow conditions as possible. Field test procedures can be combined with inspection and testing or balancing and commissioning procedures to minimize costs and duplication of effort.

Control Valves

Valve Types

Ball Valves

Modern ball valves designed for control applications are inexpensive, effective and reliable in smaller chiller plant piping. Generally, ball valves can be obtained in sizes that fit up to two-inch or three-inch pipe. Ball valves are well suited for *isolation valves* because they can be ported for full pipe size (i.e., the opening in the ball valve is the same as the inside diameter of the pipe, reducing pressure drop). Ball valves are also well suited for modulating control because they act as *equal percentage valves* when fully ported or in special porting configurations. They offer substantially lower first costs than globe valves that have been traditionally used for modulating duty. However, designers must ensure that all ball valves specified for control applications are specifically intended for that purpose; standard ball valve designs are not adequate for the continuous movement required for modulating control duty and usually suffer seal failures in a short period of time.

Butterfly Valves

Butterfly valves are the most popular large-diameter control valves in chiller plants. Like ball valves, butterfly valves make excellent isolation valves because they offer nearly full pipe bore when open and thus have low pressure drop. Butterfly valves also have valve characteristics similar to equal percentage valves when used in modulating valve applications. However, they have very low pressure drop and thus a low valve "*authority*," usually making them inappropriate for use in two-way modulating duty at cooling coils in variable-flow systems.

Globe Valves

The use of *globe valves* in chiller plants has dramatically decreased in recent years. This is because the introduction of cost-effective, reliable electric rotating *actuators* for large butterfly valves has eliminated the need for *linear actuation*. However, globe valves continue to have some advantages: they can be manufactured with a variety of flow/position characteristics and they permit three-way valve action with a single valve. The benefits of these features are sometimes significant enough to offset the higher cost of globe valves. Globe valves are still the most common control valves for large cooling coils. They have relatively high pressure drops and provide good "authority" for improved controllability and are thus the valve of choice (along with ball valves for smaller coils) in two-way valve, variable-flow systems.

Two- and Three-Way Valves

All of the above valve types are available in a two-way configuration for variable-flow system applications. For constant-flow applications, globe valves, and more recently ball valves can be configured as a *three-way* valve in a single unit. The mixing arrangement (two ports where flow enters the valve and one port for flow leaving the valve) is less expensive than the diverting arrangement (one port entering, two ports leaving). In large valve applications, using two butterfly valves is often more economical than using a single globe valve. Also, globe valves are hard to find in valve sizes above 12-inch pipe diameter. However, designers must be careful when using multiple butterfly valves in a modulating three-way valve application. Because of their equal percentage position versus flow characteristic, the modulation of the two valves cannot be a linear relationship if a constant flow coefficient is required.

Pressure-Independent and Other Special Valves

Some manufacturers package valves with other devices to provide constant or minimum flow control and other special control features. Generally such special purpose valves are not required in typical chiller plant configurations, but may be of use in solving special problems or system features. (See Chapter 4 for a discussion of the use of pressure-independent valves.)

Valve Selection Criteria

Valve Sizing and Flow Coefficient

Valve sizing in two-position (on/off) applications is straightforward: the valve is simply the same size as the piping it is installed in. But valve sizing in modulating applications is more difficult and a fairly controversial subject. The valve size is based on its full open pressure drop, which in turn determines the valve's "authority" and the ability of the control system to function as desired and expected. It is probably intuitively clear that an oversized valve will not be able to control flow well. As an extreme example, imagine trying to pour a single glass of water using a giant sluice gate at the Boulder Dam. But undersizing a valve increases the system pressure drop, which leads to higher pump cost and higher energy costs. These two considerations must be balanced when making valve selections.

The size of a valve is determined by its pressure drop when it is full open. The question then is: What pressure drop should be used? Unfortunately, there is no "right" answer to this question and there are various differing opinions and rules-of-thumb expressed by controls experts and manufacturers (discussion of which is beyond the scope of this Guide). While there is disagreement about the exact value of the desired pressure drop among these authorities, there is general agreement that the control valve pressure drop, whatever it is, must be a substantial fraction of the overall system pressure drop in order for stable control to be possible.

With the advent of more sophisticated control algorithms such as proportional + integral + differential (PID) and fuzzy-logic, some designers have questioned the need for high valve pressure drops. However, while a well-tuned controller can certainly compensate for some valve oversizing, there is clearly a point where no control algorithm will help. For instance, getting a single glass of water out of a sluice gate will be impossible no matter how clever the control algorithm may be. Oversizing will also result in the valve operating near close-off most of the time. This can increase noise from flow turbulence and may accelerate wear on the valve seats. Therefore, relaxing old "rules-of-thumb" on valve selection is not recommended.

Once the pressure drop is determined, the valve can be selected using a rating called the *valve flow coefficient*, Cv. The valve flow coefficient is defined as the number of gallons per minute of fluid that will flow through the valve at a pressure drop of one psi with the valve in its wide-open position, expressed mathematically as:

$$C_v = Q\sqrt{\frac{s}{\Delta P}}$$

Equation 5-1

Where,

Q = flow rate in GPM

s = specific gravity of the fluid (the ratio of the density of fluid to that of pure water at 60°F)

ΔP = pressure drop in psi

Specific gravity for water below about 200°F is nearly equal to 1.0, so this variable need not be considered for most HVAC applications other than those using brines and other freeze-protection solutions. Valve coefficients, which are a function primarily of valve size but also of the design of the valve body and plug, can be found in manufacturer's catalogs.

Valve Cycles

One important criterion for selecting control valves is the frequency of movement that the valve must sustain without leaking or failing. The designer should determine the number of actuation cycles each valve is likely to be subjected to over a year. This is important, because without this information a lower quality, manual shut-off valve may be installed. If this type of valve is subjected to frequent operation, it may develop stem leakage problems and fail prematurely.

An emergency or shut-off valve may see only a few cycles each year and a two-position isolation valve may see a few cycles each day, but a modulating valve may be subjected to several thousand movements in a single day. *Pneumatic actuators* generally have several million position changes before expected failure, but as the industry moves to *electric actuation*, it is essential that designers understand and accommodate the characteristics of electric actuators. The life expectancy of high quality electric actuators is usually in the range of 250,000 to 500,000 total position changes. This is also the approximate life expectancy for O-ring shaft seals that are used on rotating shafts in control valves. To reduce maintenance and replacement costs, designers should:

- consider steps to minimize the number of valve actuations;
- specify the operational environment of each valve; and
- establish minimum performance and longevity requirements for each valve.

Valve Characteristics

Both ball and butterfly valves have *equal percentage* operating characteristics. However, when a ball valve is ported for a smaller *flow coefficient*, it begins losing the equal percentage characteristic as the coefficient is decreased below the full port size. Therefore *ported-type ball* valves are not suited for modulating flow control applications unless a special porting configuration has been incorporated that maintains the equal percentage characteristics. If a valve characteristic other than equal percentage is desired, globe valves can be obtained with several different operating valve characteristics. When the valve application calls for a two-position isolation or shutoff valve, the valve characteristic need not be considered.

Valve Shutoff

Modern valve seat materials provide zero leakage for many valves and pressure conditions. However, substantial variances in close-off ratings exist between valves. For both ball and butterfly valves, the fluid pressure does not affect the closing force required, but fluid pressure is a factor with globe valves. With globe valves, it is important to carefully consider the valve operating conditions to ensure that the valve has adequate close-off capability. Many valves will have two close-off ratings, one for two-position duty and another for modulating duty that is sometimes called the "dynamic" close-off rating. The dynamic rating, which is always lower than the two-position rating, is the maximum differential pressure allowed for smooth modulation of the valve, particularly near shut-off. Above this differential pressure, the design turndown ratio will not be achieved. This is the rating that should be used when selecting a valve for modulating applications. In two-way valve systems, a common practice is to require that valves be capable of modulating and/or shutting off against the pump shut-off head plus a safety factor (typically 25% to 50%). This is conservative for systems with variable-speed driven pumps, but still advisable since the pumps may be operated at fixed speed in case of VFD failure.

Valve Actuators

Whenever possible, control valves and actuators for chiller plants should be purchased as a *single unit* that is designed to meet the specified requirements. Variations in breakaway torque as well as closing force variations due to fluid pressure may affect the size of the actuator required. It is also important that the actuator control signal be compatible with the direct digital control (DDC) system output, and that the power requirements of the valve are compatible with the voltage and current phase available. Many electric actuators include an analog point that provides positive position feedback. This additional DDC input point may be very useful for chiller plant applications in which a valve failure may cause serious flow or other problems.

Controllers

This discussion on controllers assumes that chilled water plant controls are direct digital controls (DDC). This will almost always be true for new buildings and is also true for most retrofits since controls are often installed along with new chillers. This section discusses the following aspects of DDC controllers:

- Minimum requirements, including programmability, variables, input/output point capacity, analog-to-digital resolution, automatic networks, and trend logs.
- Interfacing issues encountered with chiller networks, variable-frequency drive networks, and power metering networks.

Minimum DDC Controller Requirements

Programmability

Designers should ensure that any DDC system that is to be used for chiller plant control have a powerful and flexible programming language. This language should permit a nearly unrestrained capacity to incorporate logic, mathematics, timing and other functions. This is important not only for the initial development of the plant's operations but also for future improvements to the plant's performance.

Variables

Chiller plants must be able to operate automatically under various operating conditions, including those caused by equipment failure and manual operator override. It is essential that the DDC system have the capacity for a large number of variables (sometimes called pseudo, virtual, or software points) so that features such as lead-lag sequences, automatic failure remedies, and operator disabling of equipment for maintenance can be implemented simply and effectively.

Flexible I/O Point Capacity

The DDC system must interface to *I/O devices* that use industry standard interfaces. Also, each chiller plant may have a unique mix of inputs, outputs, and *network points*. Therefore DDC controllers that have universal I/O points with flexibility for a software or hardware (via module) definition for each may provide lower cost and greater flexibility for future changes than those with dedicated I/O hardware.

Analog-to-Digital Resolution

It is very important that the *analog-to-digital* (A/D) conversion provide adequate resolution to read all analog inputs accurately to the number of significant digits desired. A 12-bit A/D resolution for analog inputs (4,096 segments for the device span) is recommended for analog inputs. A lower A/D resolution for analog outputs is usually acceptable.

Automatic Network

Each DDC controller used in a chiller plant must have the capacity to automatically and seamlessly share all point and variable information with other controllers in the plant. It must also be able to preserve the required analog value precision for those points whose value must be transmitted across the network. Also, the network characteristics must require that:

- the maximum *Change of Value* or COV (where employed) for all points be the same as the specified precision or accuracy of those points; and
- the maximum scan time (where employed) be less than 30 seconds (this means that all controllers will use point data that is within at least 30 seconds of the current value).

Trend Logs

Controllers should have the capability to record historic data on the status of control points, to analyze this data, and produce trend logs that show the behavior of the control point relative to other variables. Plant operators should have the capability to identify the control points for which trend logs are generated, to set the time interval for taking data, and other specifications of the trend log.

Comparative information on manufacturers control system capabilities is available from the DDC online website: http://www.ddc-online.org/

Interfacing with Chillers and Chiller Networks

Digital Interface

Virtually all new chillers are supplied with a complete package of controls and instrumentation that is factory installed, tested and calibrated, and can be integrated into the DDC system to control and monitor each chiller without requiring any additional sensing equipment or installation. The most economical method of integrating the instrumentation into the DDC system varies by manufacturer. It is advisable to specify a *BACnet gateway* between the chiller(s) and the DDC system. If for some reason BACnet is not used as the system protocol, it may be possible to negotiate a cost deduction from the vendors, under these conditions:

- The chiller and controls manufacturers have an interoperability agreement for which a standard gateway or connection that meets all the requirements of the project may be substituted for the BACnet interconnection at less cost.
- The same manufacturer supplies the chiller and controls so that there is full compatibility without any special gateway or connection.

Minimum Interface Points

The following are the minimum chiller monitor and control points that should be accessible directly from the *chiller panel*:

- Supply (leaving) chilled water temperature
- Return chilled water temperature
- Supply (entering) condenser water temperature
- Leaving condenser water temperature
- Evaporator refrigerant pressure
- Evaporator refrigerant temperature
- Condenser refrigerant pressure
- Condenser refrigerant temperature
- Compressor discharge refrigerant temperature
- Oil temperature
- Oil pressure
- Chiller operating electrical demand (percent of RLKW)
- Condenser water flow status
- Chilled water flow status
- Chiller operating status
- Chiller alarm status (individual alarms as an option)
- Chiller start/stop (enable/disable)
- Chiller demand setpoint (percent of RLKW)
- Chilled water temperature setpoint

Interfacing with Variable-Frequency Drive (VFD) Networks

Digital Interface

Like chillers, most variable-speed drives (most commonly variable-frequency drives, or VFDs) come with a complete control and instrumentation package. Many, but not all, can also be supplied with a link that permits them to communicate directly with the DDC system without having to provide separate hardwired control and monitor points to each unit. The designer should use caution when controlling VFD speed over the network from a control loop and control variable that is located in some other controller on the network. Time delays in transmitting the speed signal over the network are variable and may cause the control loop to be unstable, or at least difficult to tune. It is recommended that either:

- The control point be connecting directly to the VFD and the on-board controller (generally available on modern VFDs) be used to control VFD speed. The control point setpoint may be reliably transmitted over the network to the VFD since it changes only infrequently (if reset) or not at all.
- The VFD signal be hardwired as an analog output from the controller so that the network is not required. The network interface is still desired since other information may still be transmitted over it, such as status, power, alarm conditions, etc.

Minimum Interface Points

The following are the minimum VFD monitor and control points that should be accessible directly from each VFD panel:

- VFD drive status
- VFD fault status
- Output speed (percent of maximum)
- Output power (kW)
- VFD and motor start/stop (enable/disable)
- VFD speed setpoint

Interfacing with Power Metering Networks

Digital Interface

Some *power sensors* have links that permit them to communicate directly with the DDC system without requiring the installation of separate hardwired monitor point connections to the unit. An advantage of using a network to receive data from power sensors is that these sensors can provide a great deal of additional information over the network for a modest cost premium. This is particularly economical if there are many sensors to be connected. Several communication links are available to network power sensors to DDC systems.

Interface Points

The following power monitor points are widely available with network-connected power sensing units:

- Instantaneous kW demand
- kWh reading (resettable)
- Power factor
- Voltage
- Frequency
- Current (amperes)
- kVAR
- VA

Integrating Chiller Plant Efficiency Monitoring with Control

The Benefit of Performance Monitoring

In many climates, chiller plants are responsible for a major portion of a facility's energy use. Performance monitoring can help identify energy efficiency opportunities. Many chiller plants are not fully automated, and nearly all plants require ongoing maintenance to achieve top operating efficiencies. Integrating chiller plant monitoring with the control system helps the plant operating staff to determine the most efficient equipment configuration and settings for various load conditions. It also helps the staff to schedule maintenance activities at proper intervals, so that maintenance is frequent enough to ensure the highest levels of efficiency, but not so frequent that it incurs unnecessary expense.

Monitoring Considerations

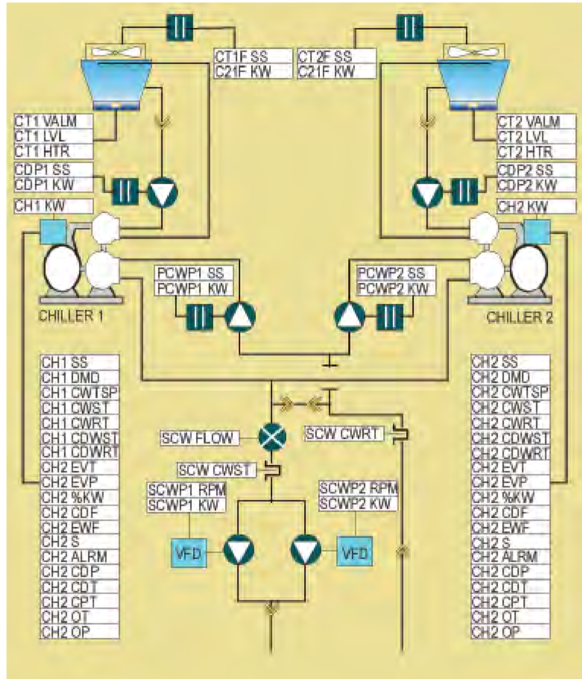
It does not have to be expensive to integrate energy efficiency monitoring with chiller plant control. Most DDC systems that are capable of operating chiller plants effectively are well suited to provide monitoring capabilities. Since chiller plant efficiency is calculated by comparing the chilled water energy output to the energy (electric, gas or other) required to produce the chilled water, efficiency monitoring requires only the following three items:

- ***Chilled Water Output***. Since the control instrumentation already includes chilled water supply and return temperatures, only a flow sensor must be added to normal chilled water plant instrumentation.
- ***Energy Input***. To obtain the total energy input, it is necessary to install kW sensors on the tower fans, condenser pumps, chillers, and chilled water pumps. The installation cost is not greatly increased if the kW sensors are used instead of other status devices for fans, chillers, and pumps. It may also be possible to use only one or two kW sensors to measure the total energy used by the plant. Finally, it is often acceptable to use a predetermined kW draw for constant speed fans and pumps whenever they are operating.
- ***DDC Math and Trend Capabilities***. In addition to the instrumentation requirements, efficiency monitoring requires that the DDC system chosen have good math functions so that the instrumentation readings can be easily scaled, converted, calculated, displayed and stored in trend logs for future reference.

Typical Monitoring and Control Point Configuration

Figure 5-1 illustrates that instrumentation can be simple yet still effectively control and monitor a typical chiller plant. All chiller points except the kW point are obtained through the gateway connection to the DDC. In this configuration, kW sensors are used as the status indicators for fans and pumps. This adds some instrumentation cost, but no additional DDC system cost.

Figure 5-1:
Typical Monitoring and Control Point Configuration



Local Instrumentation

Evaluating the Need

The cost of including a DDC system operators' terminal in the chiller plant is low. The decision of whether to include separate local sensors with readouts is primarily a matter of the operations' staff preference and the cost of the local instrumentation. It is usually less expensive to use the DDC system as the local sensor readout platform by installing an operators' terminal in the plant. However, in very large plants, or those that incorporate several separate rooms or air locks, it may be useful to install separate sensors with local readouts for some areas of the plant.

Local Instruments

Local instruments such as pressure gauges (typically installed on pumps) and thermometers have poor records for accuracy and longevity. It is not unusual to visit machine rooms less than ten years old and find half of the total temperature sensors to be faulty. If local pressure gauges or thermometers are desired, specify devices of the highest quality and install them on the wall in locations that are not subject to vibration or other physical shock.

Pressure and Temperature Test Ports

Chiller plant piping should be specified to contain taps (also called test ports) at all major equipment for testing pressures and temperatures.

Size of Taps

Test taps should be large enough so that the largest instrument that may be attached or inserted at that point does not require any piping modification, but small enough that more typical taps can be made without having to shut down the system. A combination of large taps with reducers and/or shutoff valves can incorporate such flexibility without a lot of hardware.

Location of Test Taps

Test taps should be installed in addition to all other taps that may be required for control or local instrumentation. The purpose of test taps is to maintain easy access to the system by O&M personnel so that tests may be performed to measure flow, temperature, and differential pressure at or across each system component without the need to shut the system down or to disturb or remove any existing instrumentation. Test taps should be located at or near the inlets and outlets of all chiller plant components such as boilers, pumps, chillers, heat exchangers, cooling coils, strainers, chilled water control valves, and other equipment for which periodic testing may be useful.

Control Sequences

Control sequences are covered in Chapter 6.

6. OPTIMIZING DESIGN

Previous chapters discussed the basic principles behind central chilled water plant components and distribution system design. This chapter provides procedures and analysis techniques for optimizing the design to minimize first costs and operating costs (in particular, energy costs) over the plant's life cycle. This chapter primarily applies to new chilled water plants, but many of the techniques can be used for retrofit projects as well.

To rigorously optimize a central plant design would be a Herculean task due to the almost infinite number of design decisions that affect energy costs and first costs. For instance, energy costs are determined by:

- The full-load and part-load efficiency of each piece of plant equipment (e.g., chillers, towers, pumps);
- The number of each piece of equipment and how they are staged;
- The design of the distribution system (e.g., variable flow versus constant flow, primary-only versus primary-secondary);
- The control sequences such as chilled water temperature reset and differential pressure reset; and
- The pipe and valve sizing.

First costs can be even more complex to account for during initial design. There are many reasons for this, including:

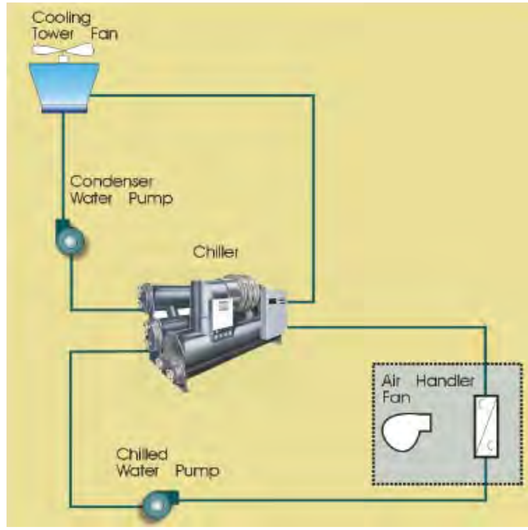
- Costs are not a continuous function of capacity;
- Capacity for some equipment and materials is only available in discrete sizes;
- Costs vary by manufacturer and by market conditions; and
- Costs will vary widely depending on the physical layout of the plant and other design details.

Rather than trying to account for every design variable, this chapter suggests an approach to the design of chiller plants that combines detailed analysis and rule-of-thumb recommendations. This approach provides much better results in terms of plant performance and cost compared to traditional design procedures.

Design Procedure

Figure 6-1 shows a schematic of a simple chiller plant serving an air-handling unit. Operating costs include those from the chiller plant components (chiller, chilled and condenser water pumps, cooling tower) as well as those from air handling system fans. All of these costs can be affected by plant design decisions, such as chilled water temperature. Rigorous optimization of the system's life-cycle cost would require a mathematical model of each possible system component and design option, accurately describing and defining its operating performance and first costs. Unfortunately, this approach is not practical; there are simply too many options and their cost and performance cannot always be described by continuous mathematical functions.

FIGURE 6-1:
CHILLER PLANT SCHEMATIC



Nevertheless, it is possible to partially optimize the chiller plant with a reasonable engineering effort. The key is to break the chiller plant into subsystems and then optimize within those subsystems. The plant will not be completely optimized due to the complex interaction between subsystems, but the result should be close to optimum.

Before beginning the detailed design process, however, it is necessary to first develop plant load profiles as described in Chapter 2. It is also essential that the designer be very knowledgeable about chiller plant equipment (see Chapter 3) and chilled water distribution systems (see Chapter 4).

For most chiller plants, near-optimum plant design can be achieved by the following step-by-step procedure:

1. Select chilled water distribution system flow arrangement.
2. Select chilled water temperatures, flow rate, and primary pipe sizes.
3. Select tower speed control option, efficiency, condenser water temperature range and approach temperatures, and make preliminary cooling tower selection.
4. Select chillers using performance specification and life-cycle analysis.
5. Adjust tower sizing and number of cells if necessary.
6. Finalize piping system design and select pumps.
7. Develop and optimize control sequences.

Again, the ideal solution is to optimize all plant components simultaneously, but that is not usually practical. Even with the simplified approach given here, some interactions are so important that they cannot be entirely evaluated independent of other interactions. This is why the design of the components listed in step 3 is preliminary and must be finalized in later steps.

Each of these steps is discussed in the following sections.

Selecting Chilled Water Distribution System Flow Arrangement

Simplified Distribution System Selection

The most common chilled water distribution system types are discussed in Chapter 4. In most cases, by relying on experience and general rules-of-thumb, the best choice for a given project can be selected without an unnecessarily burdensome analysis. Table 6-1 shows recommendations for distribution system options based on the size and number of loads served and distribution system losses. These recommendations are generalizations that should apply to the majority of typical HVAC applications, but they may not prove to be optimum for every application.

Table 6-1:
Chilled Water
Distribution System

App. No.	Number of Coils/ Loads Served	Size of Coils/ Loads Served	Distribution Losses (excluding chiller)	Control Valves	Flow (Primary/ Secondary)	Recommended Distribution Type
1	One	Any	Any	None	Constant	Primary-only, no control valves (Fig. 4-1)
2	More than 1	Large Campus	Any	2-way	Constant/Variable	Primarydistributed secondary (Fig. 4-11B)
3	More than 1	Large coils (> 100 gpm)	Any	None	Constant/Variable	Primary/coil secondary (no control valves) (Fig. 4-11A)
4	Few (2 to 5) serving similar loads	Small (< 100 gpm)	Low (< 40 feet)	3-way	Constant or Staged	Primary-only, constant flow (Fig. 4-2, 4-3)
5	Few (2 to 5) serving similar loads	Small (< 100 gpm)	High(> 40 feet)	2-way	Variable or	Primary-only variable flow (Fig. 4-7, 4-8) or
6	Many (more than 5) or few serving dissimilar loads	Small (< 100 gpm)	Any		Constant/Variable	Primary-secondary variable flow (Fig. 4-9) See Table 6-2

Note that the suggested definition of “Many” coils (more than 5) and “High” distribution losses (more than 40 feet pressure drop) are rules-of-thumb. When the actual system design is near these limits, both the Few/Many and Low/High distribution system options should be analyzed to see which is optimum.

Application Notes (corresponds to application number at left of Table 6-1):

1. For a plant serving one coil, the system shown in Figure 4-1 is usually the most cost-effective approach. It eliminates the expense and pressure drop of a control valve and realizes significant energy savings from the wide range of chilled water temperature reset that occurs.

2. For plants serving groups of large loads such as buildings in a college campus, terminals in an airport, etc., distributed variable-speed-driven secondary pumps (Fig. 4-11B) is usually the best solution. The use of distributed pumps reduces pump energy by allowing each secondary pump to be selected and operated at the head required to supply water from the plant to the building. With a conventional primary-secondary scheme (Figure 4-9), the buildings closest to the pumps will have more pressure than needed; this pressure is throttled by the control valves, which wastes energy and can cause control problems. Other advantages over conventional primary-secondary systems include:
   - improved balancing,
   - much better energy performance when only a few buildings are on-line, and
   - elimination of the need or desire for reverse return.

3. For plants serving large individual air handling systems, distributed variable-speed-driven coil secondary pumps (Fig. 4-11A) are usually the best solution. This design will be less expensive than a primary-secondary system and cost competitive with a primary-only system. It will be more efficient than both these options for two reasons: First, this design eliminates the pressure drop (and expense) of control valves, generally about a 10-foot reduction in head. Second, as with the campus system, each secondary pump may be selected and operated at the head required to supply water from the plant to the coil. With a conventional primary-secondary scheme (Figure 4-9), the coils closest to the pumps will have more pressure than needed; this pressure is throttled by the control valves, which wastes energy and can cause control problems. Other advantages include better, more responsive control. Disadvantages include the need to have a pump at each coil. For this design to be energy efficient, coils must be large due to the inherent inefficiency of small pumps. If there are both small and large coils, a hybrid system of both distributed coil pumps and conventional secondary pumps to serve small coils is possible. (See Figure 6-9 for an example hybrid plant.)

4. When there are few coils in a system and the distribution head is low, the simplicity and low cost of a constant volume primary-only distribution system (Fig. 4-2, 4-3) is usually optimum despite its higher pump energy costs compared to variable-flow systems. Aggressive use of chilled water reset often allows the plant to have efficient energy usage despite higher pumping costs (as with application 1). With multiple chillers, a quasi-variable-flow system can be attained by staging the chillers on coil demand, provided all the loads tend to vary together (i.e., no coil requires full cooling while others are at part load). This is typical of many HVAC applications where, for instance, the coils all are in VAV systems serving the same occupancy type. If there are multiple chillers in the system and the loads do not vary up and down together, the systems in application 5 should be used.

5. When the system distribution pressure drop is high, the return on investment for variable-flow systems improves and the optimum system will generally be a primary-only pumping system (Figures 4-7 and 4-8) or a primary-secondary pumping system (Figure 4-9). Also, see the application notes discussed under application number 6.

6. When there are many valves in the system, the construction cost savings and start-up cost savings (no balancing) of using two-way valves versus three-way valves will generally offset the added cost and complexity of the bypass valve and controls required for the primary-only pumping arrangements shown in Figures 4-7 and 4-8, or the added pump installation cost of primary-secondary pumping (Figure 4-9). The decision whether to use primary-only or primary-secondary in this application is discussed in more detail below. Variable-speed drives will usually be cost effective in this application and they will improve controllability and possibly reduce simultaneous heating and cooling at heating/cooling air handlers. If variable speed drives are not used, the primary/secondary configuration should not be used since it offers virtually no benefits and adds to first costs. (For more details, see discussion in Chapter 4.)

Primary-only vs. Primary-secondary

Table 6-2 summarizes the advantages and disadvantages of primary-only variable flow distribution systems compared to traditional (non-distributed) primary-secondary systems.

Advantages of Primary-only	Disadvantages of Primary-only
Reduced pump horsepower	Complexity of bypass control
• More efficient pumps	Complexity of staging chillers
• Fewer pump connections	• Possible chiller trips
Lower pump energy usage	• Possible evaporator freezing
Lower first costs	• Temporary high temperatures
Less plant space required	

Table 6-2: Advantages and Disadvantages of Primarily-only vs. Primary-secondary Variable Flow Distribution Systems

Primary-only systems always cost less and take up less space than primary-secondary systems, and with variable-speed drives, primary-only systems also always use less pump energy than traditional (non-distributed) primary-secondary systems. The latter point may be contrary to conventional wisdom and to heavy marketing by some pump manufacturers, but it can be shown to be true from basic principles. Figure 6-2 demonstrates this fact for a typical three-chiller, three-pump plant. The pump energy savings are due to:

- Reduced system head as a result of the elimination of the extra set of pumps and related piping and devices (shut-off valves, strainers, suction diffusers, check valves, etc.).
- More efficient pumps. (The primary pumps in the primary-secondary system will be inherently less efficient due to their high flow and low head. This can be partially mitigated by using larger pumps running at lower speed.)
- Variable flow through the evaporator, which allows flow to drop below design flow down to some minimum flow rate prescribed by the chiller manufacturer. (The minimum flow rate is always less than the design flow rate, and in some cases significantly so.)
- Near "cube-law" performance of variable-speed drives that yield significant energy savings for even small reductions in flow.

Even without the savings from first three bullet points above (that is, even if the overall pump head and efficiencies were the same and minimum flow through the chiller were maintained at the design flow), primary-only systems still use less pump energy than primary-secondary systems.

The lower energy costs and lower first costs of the primary-only system often make it an easy choice versus primary-secondary, but the system does have two significant disadvantages:

- ***Complexity of bypass control***. For all but chiller plants with a large number of chillers, some type of bypass valve will be required (Figure 4-8) in order to ensure that minimum flow rates are maintained through operating chillers. This control is somewhat complex.

  First, it requires some means to measure flow through the chillers, such as a flow meter or differential pressure sensor across chillers which can be correlated to flow. In general it is recommended that a flow meter be provided either at each chiller or at the common chilled water line on the chiller side of the bypass.

  Second, selecting the bypass control valve and tuning the control loop is difficult because of the very high differential pressure across it caused by its location near the pumps. (This can be mitigated by placing the bypass valve out in the system near the most remote coils. However, this increases flow through distribution piping, which increases pump energy at low loads. Also, it increases piping heat gain if flow is maintained in piping that might otherwise be inactive. On the other hand, placing the valve in a remote location can prevent "slugging" the plant with warm water when a remote coil starts up and may result in more stable plant control.) A pressure independent control valve should be considered for this application.

Third, control system programming is difficult when there are multiple chillers or stages each requiring different minimum flow setpoints. (Note that the pressure-activated bypass valves commonly used in the past with constant speed pumping systems will not work with variable speed pumping because the differential pressure across the valve will always be less at part load so the valve will never open.)

- ***Complexity of chiller staging.*** When one or more chillers are operating and another chiller is started by opening its isolation valve or starting its pump, flow through the operating chillers will abruptly drop. The reason for this is simple: flow is determined by the demand of the chilled water coils as controlled by the control valves. Starting another chiller will not create an increase in required flow, so flow will be split among the active machines. If this occurs suddenly, the drop in flow will cause a nuisance trip in the operating chillers, or may cause evaporator freezing if the safety controls are sluggish. To stage the chillers without a trip, active chillers must first be temporarily unloaded (demand-limited), then flow must be slowly increased through the new chiller by opening its isolation valve slowly. Then all chillers can be allowed to ramp up to the required load together. (The need for slowly allowing flow to pass through the new chiller makes it advantageous to pipe the chillers and pumps using a headered arrangement shown as Option B in Figure 4-14. See also the Pumps section later in this chapter.)

  The sheer complexity of the bypass and staging makes it likely that the controls will fail at some point in the life of the system. Thus, primary-secondary systems, despite their higher costs, do offer the benefit of fail-safe operation, very simple staging control, and no need for bypass control.

  Given these considerations, primary-only systems are most appropriate for:

  - Plants with many chillers (more than three) and with fairly high base loads where the need for bypass is minimal or nil and flow fluctuations during staging are small due to the large number of chillers.
  - Plants where design engineers and future on-site operators understand the complexity of the controls.
  - Plants for mission critical facilities like data centers and chip fabrication plants where chiller failure could disrupt the process.

A primary-secondary system is probably a better choice for buildings where fail-safe operation is essential or on-site operating staff is unsophisticated or nonexistent.

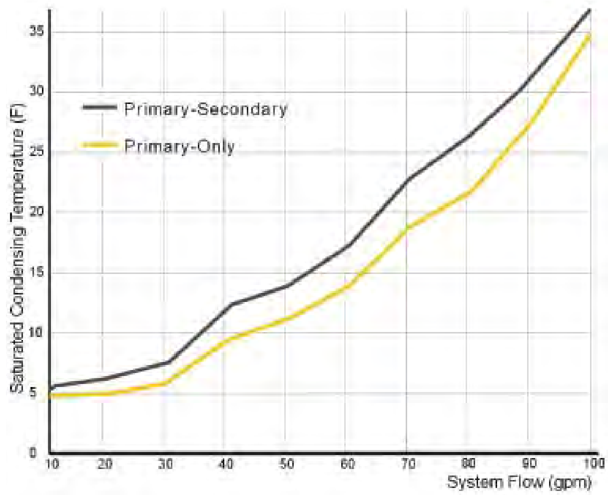


FIGURE 6-2:
PRIMARILY-ONLY VS.
PRIMARY-SECONDARY PUMPING
(FOR A TYPICAL THREE-PUMP CHILLER PLANT WITH VARIABLE SPEED DRIVES)

Optimizing Chilled Water Design Temperatures

Optimizing chilled water supply and return temperatures involves not only minimizing the life-cycle costs of the chiller plant itself but also the costs of the air handling systems that the chiller serves. This is because these temperatures will have an impact on the characteristics of the chilled water coil design, which in turn will affect the pressure drop capacity and the energy use of the supply air fan.

Table 6-3 shows the typical range of chilled water temperature difference (commonly referred to as delta-T) and the general impact on energy usage and first costs. The table shows that there are significant benefits to increasing delta-T from a first-cost standpoint, and there may be a savings in energy cost as well, depending on the relative size of the fan energy increase versus pump energy decrease as delta-T increases. Chiller energy usage is largely unaffected by delta-T for a given chilled water supply temperature.

(*Assuming constant chilled water supply temperature*)

TABLE 6-3:
IMPACT ON FIRST COSTS AND ENERGY COSTS OF CHILLED WATER TEMPERATURE DIFFERENCE

		ΔT	
	Low		High
Typical Range	8°F	to	20°F
First Cost Impact	smaller coil		smaller pipe smaller pump smaller pump motor
Energy Cost impact	lower fan energy		lower pump energy

Fortunately, within the range of commonly used chilled water supply and return temperatures, the impact on the airside of the system is seldom significant. Table 6-4 shows a typical cooling coil's performance over a range of chilled water delta-Ts. While the example in the Table will not be true of all applications, it does suggest that airside pressure will not increase very much as chilled water delta-T rises, while waterside pressure drop falls significantly. For variable air volume systems, the impact is even less significant because any airside pressure increase will fall rapidly as airflow decreases.

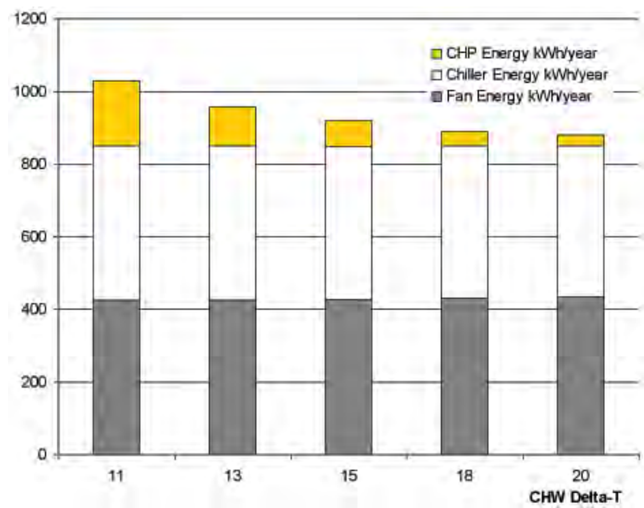
Table 6-4: Typical Coil Performance vs. Chilled Water Temperature Difference

Chilled Water ΔT	11	13	15	18	20
Coil water pressure drop, feet H_2O	27.5	19.6	14.7	10.0	8.1
Coil airside pressure drop, inches H_2O	0.46	0.48	0.49	0.52	0.54

Notes for Table 6 4: Cooling coil pressure air- and waterside drops were determined from a manufacturer's ARI-certified selection program assuming 525 fpm coil face velocity, smooth tubes, maximum 120 fpf fin spacing, 43°F chilled water supply temperature, 78°F/62°F entering air and 53°F leaving air temperature.

Fans in general use much more energy than chilled water pumps, and in most California climates they can use more energy than the entire chiller plant because the fans operate so many more hours than the chiller plant does. So although the airside pressure drops shown in Table 6 4 may be small, one might still expect that they will have a large impact on overall annual energy usage. However, this usually is not the case. Figure 6-3 shows the impact of chilled water delta-T on energy usage for a typical office building. Fan energy rises only slightly as delta-T increases. Changing delta-T with a constant chilled water supply temperature has little or no effect on chiller efficiency in the typical delta-T range. If pipe size is left unchanged as delta-T increases, chilled water pump energy will fall substantially due to reduced flow and reduced piping losses. In real applications, pipe sizes are often reduced to decrease first costs, but pump energy will still fall, although not as dramatically as in Figure 6-3, due to reduced flow rates.

Figure 6-3: Typical Annual Energy Usage vs. Chilled Water Temperature Difference



Note for Figure 6-3 and Figure 6-4: Energy demand for each component was calculated from actual equipment models for a typical chilled water plant consisting of a single chiller and chilled water pump serving an office building in Northern California. Cooling coil pressure airside and waterside drops were determined from a manufacturer's ARI-certified selection program. Pump energy assumes that pipe sizes are not reduced as flow reduces with increasing delta-T and pumps are constant volume. Fan energy is based on a variable air volume system with variable speed drive. Figure 6-3 assumes constant chilled water temperature at 43°F.

Reducing chilled water temperature can eliminate the fan energy penalty. Figure 6-4 shows the same system as Figure 6-3. However, instead of holding chilled water temperature constant, chilled water temperature is lowered to keep airside pressure drop (and therefore fan energy) constant as delta-T increases. Dropping chilled water temperature increases chiller energy but pump energy savings more than make up the difference. As with Figure 6-3, pump energy shown in Figure 6-4 assumes that pipe sizes remain constant, which is not always the case.

The example illustrated by these two figures is simple: it assumes a single chiller, constant chilled water temperature, and constant volume pumping. In applications with multiple chillers using chilled water temperature reset strategies and variable flow pumping, the results will be different in magnitude but the trend will be similar. In particular, if pipe sizes are reduced with increases in delta-T and if variable flow pumping is used, the pump energy impact of increasing delta-T will be much smaller than what is shown in Figure 6-4. Nevertheless, the trend for most applications is that higher chilled water delta-Ts result in lower energy costs, and they will always result in the same or lower first costs.

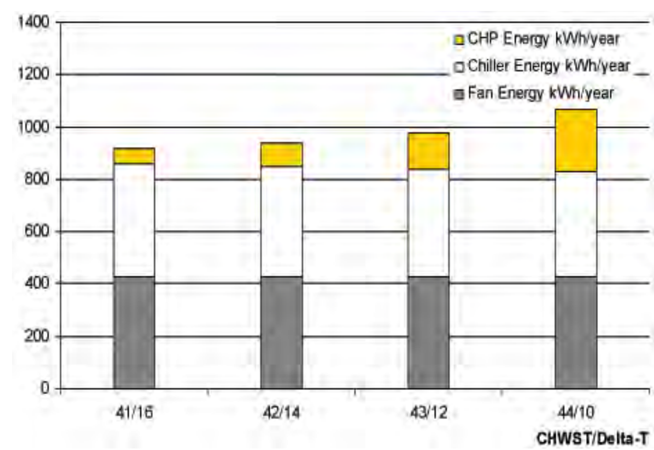


Figure 6-4: Typical Annual Energy Usage with Coils Selected for Constant Airside Pressure Drop

There are disadvantages to using high delta-Ts:

- Using a low delta-T is a conservative approach since it results in larger pump and pipe sizes. If the load increases on a plant, extra capacity can be effectively obtained by dropping chilled water temperature (e.g. from a design of 44°F to as low as 40°F), which will increase delta-T at coils. This increased flexibility to accommodate future changes is probably the most common reason for using low delta-Ts.
- To realize the reduced chiller energy usage due to rising entering water temperatures, the chiller may have to use three passes rather than two. This can add to piping costs and make tube replacement and cleaning more difficult (it is common to pipe the chiller from one side and leave the opposite head clear to provide access to tubes without having to remove piping).
- On some coils, depending on the design air temperature conditions, using a higher delta-T will not just increase the fin density, but it can also cause a jump in the number of rows which can significantly increase first costs and airside pressure drop.
- Delta-T degradation is guaranteed for coils selected with high delta-Ts on air handlers with economizers or that supply 100% outdoor air. This can cause chiller staging problems as described in Chapter 4.

Recommendations for Selecting Design Chilled Water Temperatures

This section provides a recommended approach to selecting design chilled water temperatures (see the Case Study included in this chapter for an example of how to use this approach).

In most cases, this simple procedure results in chilled water supply and return temperatures that are first-cost optimum, will provide reasonable flexibility for future changes, and that provide lower energy costs than more traditional design procedures. Note that by "maxing out" the flows in main pipe sizes, the system will not be optimum in terms of energy usage. Reducing flow rates by raising delta-T will reduce energy costs further, but this also reduces flexibility to accommodate future unanticipated changes. The procedure described below attempts to balance these competing concerns.

1. Determine the flow rate required using a delta-T of 12°F on the low end and 20°F on the high end.
2. See what pipe size (see Pipe Sizing section below) these two flow rates will result in for the main distribution pipes. In most systems, there will be many branch piping circuits so it is not practical to test every pipe in the system. However, piping costs will usually be most heavily affected by the largest piping in the system and by the piping with the most fittings and valves. This is generally at the chiller plant itself, i.e., the piping going to each chiller and chilled water pump and the main headers and risers. In almost all cases, the flow at 20°F will require one or perhaps two pipe sizes lower than the flow at 12°F.
3. Pick the smallest pipe size that can accommodate the flow ranges above, then adjust the chilled water delta-T downward so that the resulting flow rate will "max-out" the most common and largest pipe sizes at the plant. Note that this selection procedure will not maximize energy savings; it is intended to balance the advantages and disadvantages of using high delta-Ts, as discussed above and at the end of this section.

For example, assume that the simple constant flow chiller plant in Figure 6-2 is a 450-ton plant serving a typical office building (operating less than 2,000 hours/year) with piping located in non-noise sensitive areas. The flow rates and associated pipe sizes (see Pipe Sizing section) are shown in Table 6-5. At 12°F delta-T, an 8" pipe is selected, while a 6" pipe is selected for the 20°F delta-T. The 6" pipe is selected because it is smaller. The pipe sizing chart (see Table 6-8) states that 780 gpm is the maximum flow rate for this pipe size for this application. At 780 gpm, the delta-T is calculated to be 13.3°F.

Table 6-5: Example Chilled Water Temperature Difference Selection

	Low	High	As Designed
ΔT, °F	12	20	13.8
Flow rate, GPM	900	540	780
Pipe Size	8"	6"	6"

4. Select coil design delta-T. The previous step fixes the average plant delta-T. However, the average plant delta-T should not be used for coil selections. It is recommended that coils be selected for a delta-T that is 2°F larger than the design plant delta-T to allow for coil heat transfer degradation as the coil ages. The coil performance modeled in manufacturers' selection programs is ideal and assumes clean waterside and airside

surfaces. Real coils will not perform as well, particularly as they age and collect dirt on heat transfer surfaces. For example, if a 16°F delta-T is selected in step 3, coils should be selected for an 18°F delta-T.[1]

5. Select chilled water supply temperature. The previous steps set the design coil and plant delta-T, but did not set the actual water temperatures. These are determined by looking at coil selections for typical coils served by the system. Using a manufacturer's coil selection program, fix the delta-T and experiment with different chilled water temperatures between 40°F and 46°F (or higher if air handlers use warm supply air temperatures such as those serving *under-floor systems*). Through experimentation, find the warmest chilled water temperature that will keep coils from jumping into the next higher number of rows (e.g., from 6 to 8) or from requiring more than a given limit in fin-spacing density (e.g., 10 to 12 fins per inch) above which coil cleaning can be very difficult.

Note that not all coils have to be selected for the same delta-T as long as the overall average is maintained. For instance, one coil may serve a load that requires a lower supply air temperature than the average. If the plant design delta-T is used to size this coil, it may require additional rows (e.g., 8 rows instead of 6). Try lowering the delta-T for this coil until the number of rows drops, then select other coils with higher-than-design delta-Ts to make up for the lower delta-T from this coil.

Selection of Condenser Water Design Temperatures and Cooling Tower

Selecting optimum condenser water temperatures is more complex than selecting chilled water temperature due to the complex interaction between cooling towers and chillers. As with chilled water, there can be significant first-cost savings from using high condenser water delta-Ts, but the impact on energy usage is more complex. This is because the effect on chiller efficiency varies so much depending on chiller type and manufacturer. With chilled water, the supply fan energy impact was small so increasing delta-T usually reduces total system energy costs. With condenser water, the energy impact on the chiller of increasing supply and return condenser water temperature is not small. Therefore, the optimum entering condenser water temperature and delta-T are not readily determined.

The cooling tower selection and the strategies used to control the tower fans also has a significant impact on the chilled water plant's performance. Sizing the tower for a close approach to ambient wet-bulb temperature will improve chiller efficiency, but it increases tower fan energy and first costs. Other factors to consider are the efficiency of the tower itself and the tower fan speed control options.

Ideally, the chiller, condenser water pumps and piping, and cooling tower would be optimized as a package, but that is seldom practical. The chiller manufacturers do not usually sell towers,

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1. Sometimes selecting coils for one delta-T (18°F in this example), then designing the chillers, pumps, and piping system for another (16°F in this example) can cause confusion on design drawings since flow rates may not "add up." This especially can cause confusion if the system is to be balanced. To mitigate this problem, coil schedules on plans should list both the "coil selection flow rate" (based on the 18°F in this example) and the "design flow rate" (based on 16°F) that is used for pump and piping design and for system balancing. There is an alternative way to achieve a similar result but avoid the flow discrepancies: select the coils for the same delta-T as the chillers but at a warmer entering water temperature. For instance, if the chillers are selected for 42°F and a 16°F delta-T, select the coils for 44°F water and a 16°F delta-T.

seldom sell pumps, and almost never sell and install piping systems, so there is no single source for bidding and pricing these components. Also, because there are so many variables to be simultaneously optimized, it is not practical to consider the entire range of possibilities.

To make optimization reasonable and practical, a simplified condenser water system design procedure was developed. The principles underlying the procedure are based on a detailed study conducted for a simple chiller plant. As with the optimization procedure used to design the chilled waterside of the system (see Optimizing Chilled Water Design Temperatures), this procedure is simplified and therefore may not always result in a truly optimum design for all applications.

Here is a summary of the condenser water system design procedure. Each step is discussed in detail in the subsequent sections.

1. Select tower fan speed control option.
2. Select tower efficiency.
3. Select tower range and approach temperatures.
4. Make preliminary tower selection(s) for use in selecting chillers (see Chapter 7).
5. Fine tune tower selections after chillers are selected.

Tower Fan Speed Control

Tower fan control is required to maintain tower leaving water temperature under all load and weather conditions. The common options (listed roughly in ascending order of first cost premium) are:

- One-speed motor (cycling).
- Two-speed motor where low speed is half of full speed (1,800 rpm /900 rpm).
- Two -speed motor where low speed is two-thirds of full speed (1,800 rpm /1,200 rpm).
- Dual-motor drives. The smaller “pony” motor is typically designed for two-thirds of full speed.
- Variable-speed drives.

Figure 6-5 shows the energy performance of these options. The performance of the pony motor option will typically be close to the 100%/67% two-speed motor option. The performance of the one-speed fan is not linear because of the “free” cooling that occurs in the off-cycle due to the stack-effect induced natural draft through the tower.

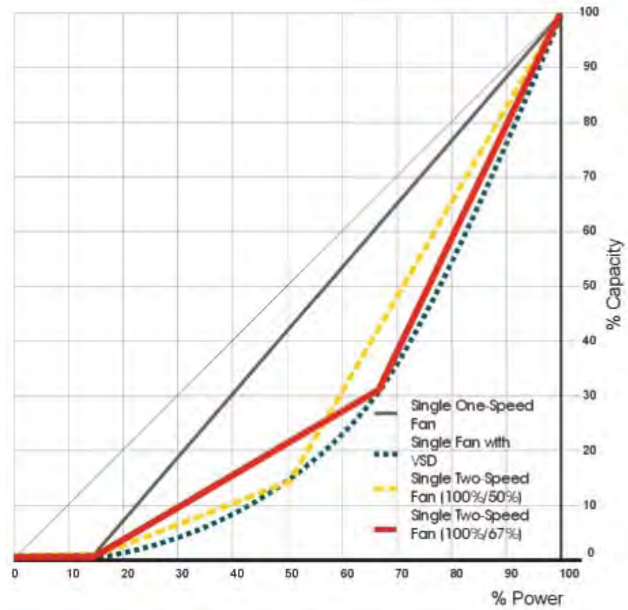


Figure 6-5:
Cooling Tower Fan Model at Part Load: Percent Power as a Function of Percent Load

The optimum fan control option depends primarily of the number of hours the tower operates under various load conditions. This in turn is a function of the application (e.g., office building, data center) and the tower control setpoint strategy. The setpoint strategy is a function of the chiller type (see Optimizing Control Sequences). Despite these complex interactions, the following generalizations can be drawn from studies of typical plants:

- One-speed control is almost never the optimum strategy for the typical chiller plant, regardless of weather, building occupancy type, or tower control strategy. Energy savings from at least one of the two-speed options result in very short payback periods. In addition, the two or variable-speed options reduce noise and wear-and-tear on belts.
- Although variable speed drives save more energy than 2-speed motors, they were not as cost effective when the life-cycle cost study was done for this standard's requirement in the late 1980s. Since that time variable speed drive costs have plummeted and they are more commonly used than two speed motors. In addition to energy savings, variable-speed drives offer other benefits, such as soft start (which reduces belt wear and electrical in-rush) and lower noise when operating at less than full speed. For towers controlled by direct digital control systems, variable speed drives can reduce control system costs if they are fitted with network cards that allow them to be connected to control system local area networks, eliminating the need for installing individual control and status points.
- The 100%/50% (1,800 rpm/900 rpm) two-speed motor is somewhat more energy efficient than the 100%/66% option, particularly when controlled as recommended in the Condenser Water Temperature Reset section. The 1,800/900-rpm motor is available in a one-winding motor, which is less expensive than the two-winding motor required for 1,800/1,200-rpm operation, although this savings is partly offset by the higher cost of the starter. At low speed, towers driven by 1,800/900 motors area also quieter than those driven by an 1,800/1,200 motor.

- Pony motors are usually more expensive than standard two-speed motors but the additional motor reduces exposure to motor failure. Note that this option is not available on all towers.
- For plants with multiple towers or multiple cells, provide two- or variable-speed control on all cells, not just the "lead" cells. The towers are most efficient when all cells are running at low speed rather than some at full speed and some off. For instance, two cells operating at half speed will use about 15% of full power compared to 50% of full power when one cell is on and the other is off.

In summary, the lowest life-cycle cost system may be the two-speed, 1,800/900-rpm one-winding motor. However in most cases, variable-speed drives are cost effective and should be considered due to their additional benefits.

Tower Efficiency

Tower efficiency (as defined in Title 24 and ASHRAE Standard 90.1) is the ratio of the maximum tower flow rate (gpm) to the motor horsepower (hp) at standard Cooling Tower Institute (CTI) rating conditions (95°F to 85°F at 75°F wet bulb). The optimum tower efficiency depends on the number of hours the tower operates under various load and weather conditions.

Two primary factors affect tower efficiency:

- ***Fan type***. Towers are commonly available with either propeller fans or centrifugal blowers. The latter require about twice the fan power and are no less expensive. From an energy and first-cost perspective, propeller fan towers (whether draw-through or blow-through) should always be used. There are a few advantages to centrifugal fan towers: they are generally quieter than propeller fan towers and they can operate against a larger external static pressure drop, such as that caused by louvers when towers are located indoors. However, the design can usually be modified to accommodate propeller fan towers, such as by oversizing the tower to reduce fan speed and noise, or oversizing intake louvers to reduce pressure drop. Because of the severe energy penalty associated with centrifugal blowers, every effort should be made to accommodate a propeller fan tower before considering a centrifugal fan tower.
- ***Fan pressure drop through the fill***. Pressure drop is primarily a function of the fill's size and design, but also can be affected by fan inlet and discharge configuration. Most engineers use the standard tower selections available in manufacturers' catalogs. However, most manufacturers can provide custom selections: the tower and fill can be oversized to reduce pressure drop, thereby allowing the fan to be slowed down, which reduces motor power. Whether this is cost effective depends on the application, weather, and the added cost to oversize the tower and to accommodate the larger tower footprint and weight. Because these cost considerations vary so much by project, it is difficult to generalize about whether oversizing towers is cost effective.

To assist with this decision, Table 6-6 shows the maximum cost premium (dollars per nominal cooling tower ton) that will result in a cost-effective tower for various climates and two extreme applications, a data center (continuous operation, no economizers) and an office (weekday operation with outdoor air economizers). The data were developed from a simple chiller plant model (see the table notes for assumptions) in typical California climates.

For example, the Table shows that an improvement in tower efficiency from a standard tower (with an efficiency of about 38 gpm per motor horsepower) to a high efficiency tower (80 gpm/HP) for a cooling tower serving a San Diego office building must cost no more than $1.00 per 1,000 Btu/h of rejected heat (including the cost of the tower, any additional architectural enclosure costs to hide the tower, any additional structural costs due to the added weight, and any additional rigging and installation costs). If the tower were serving a data center, the premium could be as high as $2.80 /ton.

The premium for increasing tower efficiency varies widely depending on where the initial selection falls in the tower manufacturer's equipment line. This study found costs for the tower alone (price to owner, no installation premium) for an increase from standard to medium efficiency to be as low as $0.20 up to more than $1.00 per 1,000 Btu/h. Costs to increase tower efficiency to a very high efficiency tower (80 to 170 gpm/HP) were found to be about $2.00 to $3.00 per 1,000 Btu/h.

These cost data suggest that, unless there is a large premium due to architectural or structural impacts, all towers should be oversized from standard selections to achieve efficiencies on the order of 50 to 60 gpm/HP. Furthermore, towers in data centers should be oversized to achieve very high efficiencies, above 80 gpm/HP.

In Table 6-6, the maximum premium increases as the climate gets milder. Although counterintuitive, this is the result of two factors. Mild climates tend to have less of a range in wet-bulb temperatures (i.e., the difference between the average annual wet-bulb temperature and the design wet-bulb temperature is less in mild climates than in warm and hot climates). Also, in the analysis the condenser water setpoint was set lower in milder climates. (See footnotes to Table 6-6.)

Here is a summary of tower efficiency recommendations:

- Use propeller fan towers as a general rule;
- For plants serving offices and other standard commercial occupancies, evaluate oversizing tower heat transfer area to improve efficiency to 50 to 60 gpm/HP at CTI conditions; and
- For plants that operate many hours such as those serving data centers, evaluate oversizing tower heat transfer area to improve efficiency to over 80 gpm/HP at CTI conditions.

California Climate	Design Wet-bulb Temp. (0.1%)	Example Cities	Tower Efficiency @CTI Conditions (per Standard 90.1-1999)		Maximum premium $ per rejected 1000 Btu/h Application	
			Description	**gpm/HP**	**Data Center**	**Office**
Hot Valley	≥73°F	Fresno, Sacramento	Medium	58	$1.30	$0.50
			High	80	$2.00	$0.75
Warm Coastal	70°F to 72°F	Los Angeles, San Diego	Medium	58	$1.80	$0.65
			High	80	$2.80	$1.00
Mild Coastal	<70°F	San Francisco Bay Area	Medium	58	$2.20	$0.75
			High	80	$3.30	$1.10

Table 6-6: Maximum Cost Premium ($ per nominal ton) for Cost-Effective Tower Oversizing

Assumptions: Cost of electricity = $0.08/kWh. No demand charge. Standard tower assume to have 38 gpm/HP efficiency. Simple payback period fixed at 7.4 years. Base on tower performance data from Marley NC series towers. Tower leaving water setpoint set equal to wet bulb.

Tower Range and Approach Temperatures

Table 6-7 shows the first-cost and energy impacts of condenser water temperature difference within the ranges commonly used in practice. Higher delta-Ts will reduce first costs (because pipes, pumps, and cooling towers are smaller), but the net energy-cost impact may be higher or lower depending on the specific design of the chillers and towers.

(Assuming constant condenser water supply temperature)

Table 6-7: Impact on First Costs and Energy Costs of Condenser Water Temperature Difference

	ΔT		
	Low		High
Typical Range	8°F	to	18°F
First Cost Impact	smaller condenser		smaller pipe smaller pump smaller pump motor smaller cooling tower smaller cooling tower motor
Energy Cost impact	lower chiller energy		lower pump energy lower cooling tower energy

Figure 6-6 shows chiller, tower and condenser water pump energy usage for the example building introduced in Figure 6-3 and Figure 6-4. The condenser water temperature and delta-T were selected so that the cooling tower size and energy use do not change. As the delta-T decreases, the temperature of the water returning to the cooling tower decreases and the tower becomes less efficient. This requires the condenser water temperature leaving the tower to rise (or the tower size must be increased).

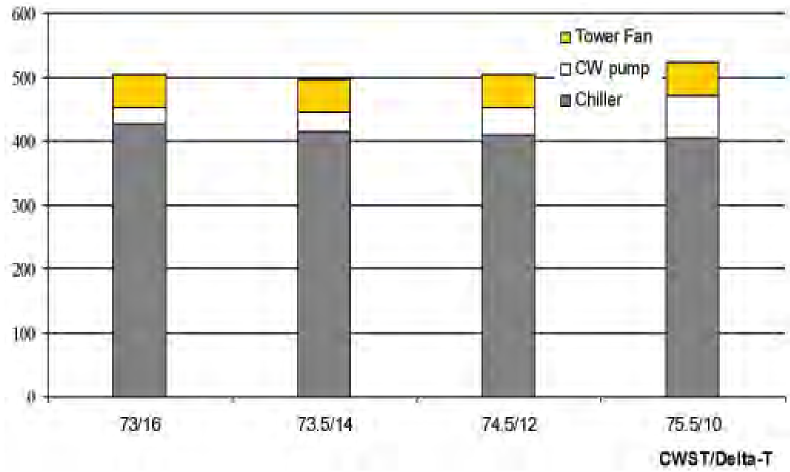


Figure 6-6: Typical Annual Energy Usage with CWST/Delta-T Selected for Constant Tower Energy

For the chiller used in this analysis, a 14°F delta-T resulted in an optimum overall plant energy usage when pipe sizes are maintained constant (the assumption in Figure 6-6), which maximizes pump energy savings. When pipe sizes are reduced, pump energy savings are reduced and the plant's overall energy usage will be optimum in the 12°F to 14°F range for this particular chiller. But not all chillers behave the same way: some, such as the 1-stage hermetic centrifugal chiller modeled here, are relatively sensitive to increases in condensing pressure that occur when delta-T is increased, while others, such as multi-stage centrifugal chillers, are not.

As part of the simplified design presented in this chapter approach, the following procedure is suggested to pick the tower range (condenser water delta-T):

1. Determine the flow rate required using a range of 10°F on the low end and 15°F on the high end for 1-stage centrifugal chillers, and 12°F on the low end and 18°F on the high end for positive displacement chillers or multi-stage centrifugal chillers. If the chiller type has not yet been determined (as is often the case if the performance specification bid technique recommended in Chapter 7 is used), look at flow rates using 12°F and 18°F.
2. See what pipe size (see Pipe Sizing section below) these two flow rates will result in for the main distribution pipes. In most systems, there will be a main header pipe and branch piping to each condenser and each tower cell. Check the pipe size in the header as well as in each branch if that is practical for each flow rate. In most cases, the flow at the high range will require one or perhaps two pipe sizes smaller than that at the low range.
3. Pick the smallest pipe size that can accommodate the flow ranges above, then adjust the condenser water delta-T downward so that the resulting flow rate will "max-out" the largest pipe sizes at the plant.

This procedure attempts to minimize cost by reducing pipe size as much as possible but then takes full advantage of the resulting pipe size to minimize delta-T to reduce chiller energy.

Next, determine the tower approach. For almost all plants, the chiller and tower efficiency will be lowest when the condenser water temperature is lowest. In the analysis of the simple plant described in Table 6-6, it was found that for plants that run long hours (e.g., data centers) a medium-efficiency tower with a 5°F approach or a high-efficiency tower with a 7.5°F approach were generally life-cycle cost optimum unless the costs to house and support the larger towers were very high relative to towers of lower efficiency and higher approach. For plants operating relatively few hours per year, such as those serving typical office and retail occupancies, the optimum tower selection was not definitive and depended on weather, tower efficiency, range, fan control, and (perhaps most significantly) where the tower selection fell in the manufacturer's range. In some cases a standard tower with 15°F approach would be life-cycle cost optimum while in others the high efficiency 5°F approach was optimum if installation costs (other than for the towers themselves) were similar.

Because the tower selections and costs play a large part in selecting the optimum tower, before laying out systems or selecting chillers, towers should be selected using a competitive bid process as follows:

1. Prepare a bid specification for the cooling towers much like the sample specification included in Chapter 7.
2. Select one or more tower vendors based on past relationships and experience.
3. Limit tower selection to those with propeller fans unless the physical location requires the use of centrifugal fans.
4. Require that towers be CTI-certified, or require some other form of performance assurance such as a field test.
5. Specify a minimum number of towers or tower cells, the tower materials (e.g., stainless steel), and other options.
6. Include space and weight constraints, if any.
7. Specify design wet-bulb temperature. This is generally the 0.1% Design Wet Bulb temperature from ASHRAE Publication SPCDX.
8. Specify condenser water flow rate and range determined as described above in this section.
9. Specify tower efficiency at a minimum of 50 gpm/hp at CTI conditions.
10. Specify approach temperatures for alternate pricing of standard catalog offerings:
    - For plants that operate many hours, such as those serving data centers, request multiple selections with approaches from approximately 7.5°F down to a 5°F.
    - For standard commercial plants, evaluate three or four approaches from 15°F down to 5°F.
11. Note that it is not necessary to deliver a certain approach temperature precisely since they are arbitrary; the selections should simply state exactly how cold the water can be delivered with the fan at full speed.
12. For each selection made, request that the vendor then oversize the tower heat transfer area in order to drop motor horsepower one size. For plants operating long hours, also evaluate dropping motor horsepower two sizes.

Request the following data for each selection:

- Model number
- Price including taxes and freight to jobsite
- Physical data including operating weight
- Motor horsepower
- Leaving water temperature at specified flow rate and wet-bulb conditions
- Flow rate at CTI standard conditions (95°F to 85°F at 78°F wet bulb). This will be used to calculate tower nominal efficiency.

Once prices are collected, many options can be eliminated by closer analysis. For instance, some higher performing towers may have a lower price than lower performing towers due to the shape or added cost of higher horsepower motors. Eliminate other options related to size constraints and weight constraints. At this point, there will usually be several options left. The designer can then follow one of two approaches:

1. Evaluate the energy performance of each option using the CoolTools Chilled Water Plant Analysis Program (CWPAP) with a "standard" default chiller plant found in the CoolTools library. Pick the tower with the lowest life cycle cost and base the chiller specification (see Chapter 7) on this selection.
2. Provide multiple tower choices in the chiller specification (see Chapter 7) and allow the chiller vendors to designate which tower they would like to use with each of their chiller options. Include both performance and price in the specification so that chiller vendors will understand the cost premium. This will allow vendors with chillers that operate efficiently at high head to pick a tower with high approach, while another vendor whose chillers operate best at low head may choose a tower with low approach. (The disadvantage of this step is that it further complicates an already complex chiller selection procedure.)

Tower Flow Turndown

Once you have selected the tower, you should have the manufacturer equip it to operate with the lowest flow turndown that they can. Typically you can achieve turndowns at or below 30% design flow through application of a mixture of high and low flow nozzles and for towers with hot water basins, weirs to separate the high and low flow nozzles. The minimum rate is usually that required to keep the fill ("wet deck") of the tower fully wet. If part of the fill is allowed to run dry, air will bypass the wet sections and tower efficiency will fall. With centrifugal fan towers, this can also cause the fan motor to overload and trip as pressure drop falls and the fan runs out its curve. It also can cause increased scale build-up on the wet deck surfaces. For these reasons, it is best to design the system so that flow will remain within the range recommended by the manufacturer. As described in the control section, you always want to run the most cells that you can as it will decrease the tower fan energy. For a given flow, range and approach on a tower with a variable speed drive running twice the cells will use ¼ of the fan energy. Designing towers for low flow is a requirement in Title 24 and ASHRAE Standard 90.1.

Fine-Tuning Tower Selections

Once the final chiller selections are made (see Selecting Type, Number and Size of Chillers below), it may be necessary to fine-tune the selection of cooling towers. For instance, the size and number of tower cells may have to be adjusted to correspond to the size and number of chillers. For multi-cell towers, isolation valves may also be required at each cell to prevent low tower flow rates when only one or a few chillers are operating.

Selecting Type, Number, and Size of Chillers

Design and selection procedures for chillers are discussed in Chapter 7. The procedures are included in a separate chapter to make them easier for engineers to use for retrofit and other projects that do not include the design of the chilled and condenser water systems addressed in this chapter.

Optimizing Piping Design

Pipe Sizing

Traditionally, most designers size piping using rules of thumb, such as limiting friction rate (e.g., 4 feet per 100 feet of pipe), water velocity (e.g., 10 feet per second), or a combination of the two. These methods are expedient and reproducible, but they seldom result in an optimum design from either a first-cost or life-cycle cost perspective.

Selecting the optimum pipe size for a given design flow rate is a function of:

- Location of pipe in the system (whether or not in the "critical circuit," i.e., the circuit that determines pump head);
- First costs of installed piping;
- Pump energy costs, which in turn will depend on pump and motor efficiency, distribution system type (constant or variable flow), annual flow profile through the system as well as the pipe in question, type of pump control (variable speed or *riding pump curve*), etc.;
- Erosion considerations (high velocities can contribute to hastening of pipe wall deterioration);
- Noise considerations, such as velocity limits to minimize noise caused by turbulence and the proximity of the pipe to noise-sensitive areas;
- Physical constraints; and
- Budget constraints.

First costs and energy costs could be combined into a piping system life-cycle cost. Ideally, each pipe in a system would be selected to minimize life-cycle costs, within the erosion, noise, physical and budget constraints noted above. This is obviously not practical with today's design tools. The sizing procedure described in this section was developed to approximately account for the above considerations while still being fast and easy to use.

Table 6 8 is a pipe-sizing chart that can be used for sizing chilled and condenser water piping. This chart was developed by the pipe size optimization applet that is included with this document. The criteria used to generate the table is detailed in the worksheet "Introduction" in the attached tool. To develop the criteria for Table 6-8 a simple chilled water plant model was input into the sheet and the flow rate was iterated on to develop the limits of each pipe size. Table 6-8 was adopted as Addendum af to ASHRAE/IESNA Standard 90.1-2007. The requirement includes the following guidance in applying the table:

All chilled water and condenser water piping shall be designed such that the design flow rate in each pipe segment shall not exceed the values listed in Table 6-8 for the appropriate total annual hours of operation. Pipe size selections for systems that operate under variable flow conditions (e.g. modulating 2-way control valves at coils) and that contain variable speed pump motors are allowed to be made from the "Variable Flow/Variable Speed" columns. All others shall be made from the "Other" columns.

EXCEPTION:

1. Design flow rates exceeding the values in Table 6-8 are allowed in specific sections of pipe if the pipe in question is not in the *critical circuit* at design conditions and is not predicted to be in the *critical circuit* during more than 30% of operating hours.
2. Piping systems that have equivalent or lower total pressure drop than the same system constructed with standard weight steel pipe with piping and fittings sized per Table 6-8.

Where "*critical circuit*" is defined as follows: The hydronic circuit that determines the minimum differential pressure that the pump must produce to satisfy the zone loads (e.g. the circuit with the most open valve). The critical circuit is the one with the highest pressure drop required to satisfy its load. At part load conditions the critical circuit can change based on zone loads.

The fist exception allows for undersizing the pipes that are not in the *critical circuit*. In general it is good practice to treat most if not all of the pipes as the *critical circuit* as you can accidentally cause a shorter run to become the *critical circuit* by undersizing the pipe.

The second exception addresses the fact that Table 6-8 was developed based on the performance of steel and copper pipe. If you are using concrete or plastic pipe the friction rates and optimal LCC performance will differ. The attached spreadsheet application can be used to analyze other piping systems if the appropriate cost and friction data is provided.

When using this chart, you still need to consider acoustical performance of the pipes in noise-sensitive environments. Again the attached spreadsheet application allows you to put acoustical limits on sections of the pipe (this causes the pipe sizes to increase to keep the velocity below recommended acoustical limits).

The case study at the end of this chapter provides an example of how to use this pipe sizing chart.

Table 6-8: Pipe Sizing Chart (from Addendum af to ASHRAE Standard 90.1-2007)

Operating hours/yr	2,000 hours/yr		>2,000 and ≤ 4,400 hours/year		>4,400 and ≤ 8,760 hours/year	
Nominal Pipe Size (in.)	Other	Variable Flow/ Variable Speed	Other	Variable Flow/ Variable Speed	Other	Variable Flow/ Variable Speed
2-1/2	120	180	85	130	68	110
3	180	270	140	210	110	170
4	350	530	260	400	210	320
5	410	620	310	470	250	370
6	740	1,100	570	860	440	680
8	840	1,300	650	970	510	770
10	1,800	2,700	1,300	2,000	1,000	1,600
12	2,500	3,800	1,900	2,900	1,500	2,300
Maximum Velocity for Pipes Over 12" Size	8.5 fps	13.0 fps	6.5 fps	9.5 fps	5.0 fps	7.5 fps

Selecting Valves and Fittings for Energy Savings

To minimize the energy use of piping systems, piping system pressure drop must be minimized. The procedures described in Pipe Sizing above will result in near optimum pipe size for straight pipe and fittings. However, accessories such as valves, strainers, etc. also are major contributors to system pressure and must be optimized by proper selection. The following recommendations should be considered:

- *Valves*. Use ball valves or butterfly valves for all isolation and balancing valves. These valves offer both very low pressure drop and low cost. Standard two-piece ball valves come in two types: full port (the opening in the ball is the same as the pipe size) and standard port (the hole is smaller than the pipe size). Full port ball valves have a lower pressure drop but cost more than standard port ball valves. It may be cost effective to use the full port valve for valves located in the "critical circuit" (the circuit that has the highest pressure drop and that determines the pump head). Otherwise, standard port ball valves are the most cost effective. Gate valves also have a low pressure drop but they are more expensive, less reliable for tight shut-off, and cannot be used for balancing. Do not use globe valves, plug valves, or angle valves as isolation valves.
- *Balancing valves at pumps*. Do not install balancing devices, such as calibrated balancing valves or so-called "triple-duty" valves, at pumps. These devices have much higher pressure drops than standard shut-off valves (butterfly or ball valves) and check valves. This is particularly true for pumps with variable-speed drives for which balancing should never be done at the pump. All balancing (flow throttling) should be done using the balancing device at each coil or heat exchanger, if it is done at all (see Balancing Considerations in Chapter 4).
- *Balancing devices at coils*. For the "critical circuit," either eliminate the balancing valve (see Balancing Considerations in Chapter 4) or use a low pressure drop balancing valve. The lowest pressure drop and lowest cost option is to use ball or butterfly valves for balancing, with flow indicated indirectly by coil pressure drop. This is typically measured using a hand-held pressure gauge at test ports installed on either side of the coil. Automatic flow control valves (particularly those that require inlet strainers) and

calibrated balancing valves will require significantly higher costs and pressure drop, although they allow flow to be more accurately measured and balanced. Calibrated balancing valves that use ball valves have a lower pressure drop than calibrated balancing valves that use globe valves.

- *Strainers at coils*. Consider not installing strainers upstream of coils, as is often done to protect control valves. Strainers located at pumps usually eliminate sufficient debris from the system that problems seldom occur at coils. Fine debris can pass through coils and control valves without causing damage. In fact, strainers at coils may be the cause of flow problems since they often are not easily accessible for maintenance so they get clogged and restrict flow.

Optimizing Control Sequences

This section provides assistance for design and operating engineers in developing chiller plant control sequences. Because all plants are different, sequences may not be applicable or may not be optimum for all applications.

Staging Chillers

The optimum strategy for staging chillers will depend on whether or not the chillers have variable-speed drives. Figure 6-7 and Figure 6-8 show the performance of fixed-speed versus variable-speed chillers. The efficiency of the fixed-speed chiller is better at part load than at full load (in part because condenser water temperature is assumed to fall as load falls, so-called "condenser relief") but for most chillers, efficiency will start to rise above design efficiency at about 40% to 50% load. Because of the operation of ancillary equipment, such as condenser and chilled water pumps, the overall plant efficiency will start to degrade at an even higher part-load point. But variable-speed chillers, like variable-speed pumps and fans, do not suffer from this degradation in efficiency until the load is very low, about 20% to 25% of full load.

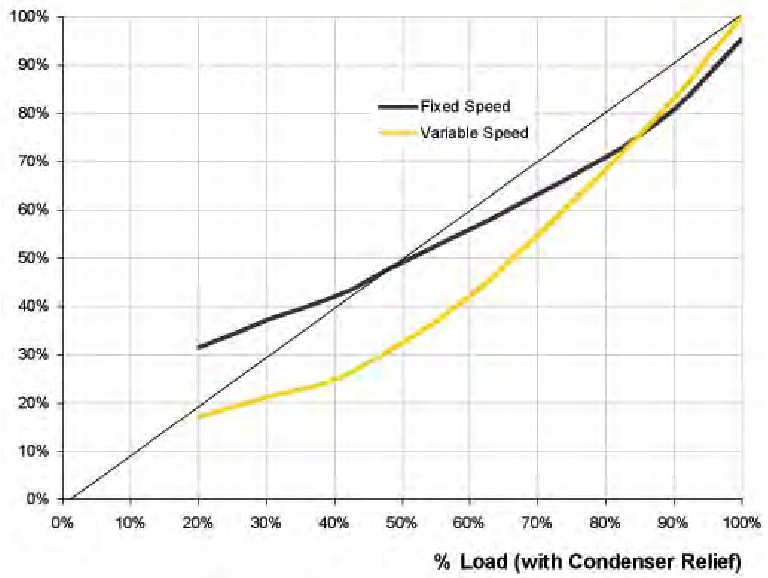
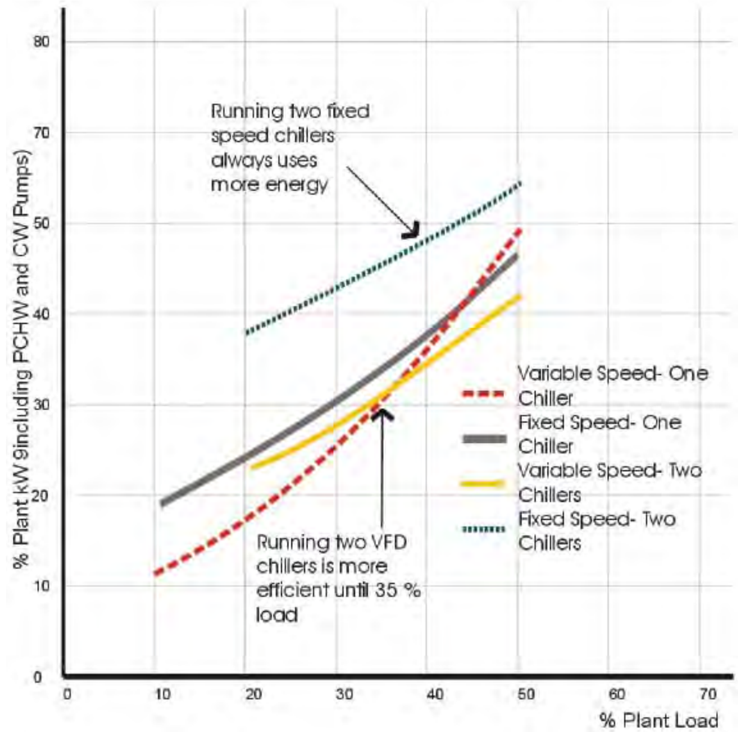


Figure 6-7: Typical Chiller Part-Load Performance with and without Variable-Speed Drives

(*Includes "condenser water relief" as defined in ARI 550/590*)

Figure 6-8:
Two-Chiller Plant
Part-Load Performance
With and Without
Variable-Speed Drives



The net effect of these chiller performance characteristics, when combined with the energy usage of ancillary pumps and tower fans, is that:

- For a plant composed of single-speed chillers, control systems should be designed to operate no more chillers than required to meet the load. This is less important if there are very many chillers and it is possible to keep them all at 75% load or higher. In general, running more chillers than required to meet the load will cause the chillers to operate inefficiently and/or cause additional pumps and tower fans to operate such that overall plant efficiency will degrade.

  In a single-speed chiller plant, staging chillers on is straightforward: a new chiller is started when the operating chillers are no longer able to meet the load, as indicated by one or more of several "load indicators." Common load indicators include plant leaving water temperature and chilled water valve position. For plants that have the ability to reset chilled water temperature, it is important that the chilled water temperature of operating chillers be reset as low as possible to ensure chillers are fully loaded before starting the next chiller. This reduces the efficiency of the operating chiller, but for most single-speed chiller plants, the total plant energy use will be less than if another chiller were started.

  Staging chillers off is much more difficult because "load indicators" usually only indicate when chillers should be staged on; they cannot readily indicate when the plant could be operating with one less chiller. Typically, chillers are staged off based on plant tonnage calculated from flow and supply/return water temperatures. For constant volume systems, such as the primary side of a primary-secondary system, flow is typically presumed from design data or measured once during balancing and assumed constant thereafter, thereby making a flow meter unnecessary. For primary-secondary systems, a flow meter in the bypass can be used for staging; a chiller is staged off when the excess flow in the primary circuit exceeds the flow from the chiller to be staged off. Chillers must be staged off conservatively; staging a chiller off prematurely will cause it to stage back on too quickly, causing excessive wear on the chiller and starters.

- A plant composed of variable-speed chillers should attempt to keep as many chillers running as possible, provided they are all operating at above approximately 20% to 35% load. For example, for the typical variable-speed chiller plant, it is more efficient to run three chillers at 30% load than to run one chiller at 90% load. The exact minimum load point will depend on the relative power required by ancillary devices (particularly condenser water pumps), cooling tower control strategies, the number of chillers, and exact chiller performance characteristics. It can be determined by experimenting with the CoolTools model of the plant. Operating more chillers than necessary to meet the load may seem counterintuitive but the logic is the same as when operating other multi- or variable-speed HVAC equipment. For instance, as noted above, running two tower fans at half speed will use about one-third the power of running one tower fan at full speed and shutting the other fan off.

  One of the benefits of variable-speed chillers is that they make staging chillers off and on less complex. While staging single-speed chiller plants can require complex stage-on and stage-off routines designed to keep as few chillers operating as possible, variable-speed chillers are efficient over a wide range of load conditions so it is not critical to stage chillers on and off with precision. For optimum control, chillers must be staged by load, typically calculated from flow and supply/return water temperature measurements. The load indicators used for single-speed plants (described above) do not work for variable-speed chiller plants because they only indicate when the plant has "maxed out" operating chillers and needs to start another chiller. For variable-speed chiller plants, it is more efficient to start chillers long before any chiller has reached full load.

  In some plants, chillers cannot be staged on based on load alone but must also be staged to maintain minimum or maximum flow rates. The strategy will depend on whether the chilled water distribution system is a primary-only or a primary-secondary system:

- For primary-only systems, if delta-T degrades significantly at part load, chillers may have to be staged on to prevent flow through active chillers from exceeding the maximum flow rate recommended by the manufacturer. In cases of significant delta-T degradation, flow may be bypassed through an inactive chiller with active chillers producing colder chilled water, so that the blended temperature meets the plant leaving water temperature setpoint. Note that reaching this maximum flow limit should never occur in a well-designed plant, that is, a plant designed to minimize delta-T degradation and designed for high chilled water delta-Ts at design conditions.

- For primary-secondary systems, chillers must be started in order to ensure that the primary chilled water flow rate exceeds the secondary loop flow rate. If it does not, then the supply water temperature to the coils will be a mixture of the supply water from the chillers and the return water from the coils. This warmer water will cause chilled water valves to open, further increasing secondary flow, which in turn further warms supply water. Eventually, the chilled water valves will be wide open and coils can be "starved," in other words, unable to meet loads. (Placing a check valve in the common leg can also resolve this problem; see Add Check Valve in Common Pipe in Chapter 4).

Pumps

Primary pumps on primary-secondary systems generally must be staged with chillers: if a chiller is started, then a pump is started; if a chiller is stopped, a pump is stopped. Condenser water pumps in conventional designs are controlled in the same manner. It is usually a good idea to have the chiller control panel make these start/stop commands as opposed to the plant control system. In other words, the plant control system tells the chiller to start, then the chiller tells the chilled water and condenser water pumps to start (and also opens automatic isolation valves on headered pumping systems; see Figure 4-14B and 4-27). This ensures that the pumps are started and stopped with the proper timing and time delays, potentially eliminating nuisance chiller trips. It also can save a little condenser water pump energy at low load: when a chiller internally cycles the compressor off at low load, it can command the condenser water pump to stop. The plant control system could not reliably provide this sequencing.

Staging secondary pumps in primary-secondary systems is generally done based on valve position (when this is known), flow, or differential pressure measured in the system near coils. The latter is most common, particularly when pumps have variable-speed drives. With both fixed-speed and variable-speed pumps, the optimum control is to operate only as many pumps as necessary to satisfy flow and pressure requirements. For fixed-speed pumps, this also minimizes differential pressure across control valves since running excess pumps causes the pumps to ride up their curves. Staging logic for systems with three or more fixed-speed pumps also must ensure that pumps do not ride too far out their curves, causing cavitation. It is common, for instance, that on a three-pump system, at least two pumps must operate due to this limitation.

Variable-speed pumps eliminate this concern, in addition to their energy saving benefits. Unlike variable-speed chillers though, variable-speed pumps are usually best staged like fixed-speed pumps to minimize the number of pumps operating. This is because the pumps all pump through the same circuit (other than the pipes into and out of the each pump between headers) so there are not "cube-law" energy benefits for each pump individually. Because of the minimum differential pressure being maintained at coils (which causes the system curve to bend off of the ideal curve at low flow, reducing pump efficiency) and because motor efficiency falls rapidly at low loads, running excess pumps will increase energy use. The typical control sequence with multiple variable-speed pumps is to stage a pump on when the operating pumps are at full speed for a period of time and differential pressure falls below setpoint, and to stage pumps off when the speed of operating pumps falls below a certain setpoint. This setpoint can be derived from pump curves or by experimenting with the actual system in the field.

Staging primary pumps in a primary-only variable-flow system is identical to staging secondary pumps as described above. The pumps must respond to the flow and pressure requirements of the system, not to the load. For headered variable-speed pumps (Figure 4-8B), it is not necessary to start a pump when a chiller starts. For instance, two pumps may be able to meet the flow requirements for three chillers over a wide flow range. This is one of the advantages of this design and it is therefore recommended for primary-only variable-flow systems over dedicated pumps piped directly to each chiller (see Primary-Only vs. Primary-Secondary in this chapter and the discussion on Figure 4-14 in Chapter 4).

For variable-speed pumps, speed is generally controlled to maintain differential pressure (DP) measured in the system near the most *"hydraulically remote" coils* (those requiring the most pump head). The DP setpoint should be determined by the system balancer using one of the following procedures depending on whether or not the system is balanced (see Balancing Considerations in Chapter 4):

- For systems that are fully balanced, the setpoint is simply the differential pressure reading when all coils are balanced and operating at design flow rates. The most remote coil should have its balance valve fully open if the system is properly balanced.
- For systems that are not balanced, the setpoint is determined by closing all control valves except the one serving the most remote coil. Manually adjust pump speed until flow through the coil reaches design rates. The differential pressure reading at this condition becomes the setpoint. (If there are several coils that might be considered the most "remote", repeat the procedure for each and use the highest resulting setpoint.)

The DP setpoint should be reset to minimize pump energy usage under low load conditions. This is most reliably done by monitoring valve position and resetting the setpoint as required to maintain the most open valve near full wide open, although this control loop can be difficult to tune. Reset is highly recommended when the DP sensor is close to the pump. This location necessitates that the setpoint be relatively high so that water can be delivered under design flow conditions to all coils, but the setpoint is much higher than it need be when flow rates are less than design.

Chilled Water Temperature Reset

Chillers are more efficient at higher leaving water temperatures. Resetting the chilled water temperature setpoint upward when loads are low is always an effective energy-saving strategy for constant-flow systems. Reset may or may not be an energy-saving strategy in variable-flow systems. High chilled water temperature will reduce coil performance, so coils in two-way valve systems will demand more chilled water for the same load, degrading delta-T and increasing flow and pump energy requirements. Whether the net energy savings (chiller energy decrease less pump energy increase) is positive and sufficient to offset the cost of implementing the reset strategy will depend on chiller performance characteristics and the nature of coil loads. During mild weather when delta-T is bound to degrade because of low flow and low load effects, resetting chilled water supply temperatures will have little or no effect on delta-T and thus will surely provide net energy savings. Smaller plants with low pumping distribution losses will also usually benefit from chilled water reset. For large plants with high pumping distribution losses, raising chilled water temperature will probably increase pumping energy more than it reduces chiller energy, resulting in a net increase in plant energy usage. Large plants may even benefit from lowering chilled water setpoint below design levels in mild weather due to the increase in delta-T this causes. As noted in the Staging Chillers section above, chilled water reset should also be used to lower chilled water temperature to load up constant-speed chillers before staging on new chillers.

Reset strategies include:

- Resetting inversely proportional to outdoor air temperature;
- Resetting from return water temperature, either indirectly by maintaining a constant return water temperature or resetting the setpoint proportional to return water temperature; and
- Resetting from chilled water valve position.

The last strategy is theoretically the best since, unlike the other two strategies, it will prevent reset from inadvertently starving coils. However, it may be impractical in some plants (e.g., one serving a large campus) to feed all the valve signals back to the central plant control system. Even where the valve positions are known, this is a complex strategy to implement and tune, particularly when valve position is also being used to reset pump differential pressure setpoint. Therefore, a conservative reset strategy based on outdoor air temperature can be the most practical and stable approach. (When reset of both DP setpoint and chilled water supply temperature are done by valve position, the two must be coordinated to prevent unstable control. One way to do this is to use the valve position of the most remote coil(s) for DP reset and use all the others for chilled water temperature reset. Both loops must be slow acting to prevent hunting.)

Note that, contrary to conventional wisdom, the impact of reset on the dehumidification capability of air handlers is quite small and should not be a concern in any California climate. The humidity of air leaving a coil is primarily a function of the coil leaving dry-bulb temperature, which is a function of the cooling load and independent of water temperature. Lowering water temperature only slightly improves the dehumidification capability of cooling coils for the same sensible load.

Condenser Water Temperature Reset

The optimum condenser water setpoint strategy is a function of the efficiency of the tower and the impact reset has on the chiller's efficiency, which will vary by chiller type. For chillers that gain a large efficiency improvement by operating at low condenser water temperature (e.g., open-drive single-stage centrifugal chillers), the tower optimum setpoint will be low at all times. For chillers that are efficient at high head (e.g., multi-stage chillers), the best strategy will be to have higher setpoints and take advantage of tower fan energy savings. The following are suggestions for determining the best condenser water temperature reset strategy for a given plant:

- Some chiller manufacturers offer proprietary control schemes for dynamically optimizing reset. This is probably the best option if it can be implemented without a large investment in control equipment (e.g., if the manufacturer will allow the scheme to be programmed into the plant's control system rather than having to purchase a second control system, or if the chiller plant control system was provided by the chiller manufacturer).
- If the plant is modeled on the CoolTools Chilled Water Plant Analysis Program (CWPAP), different strategies can by evaluated by trial and error. (However, the current CWPAP is limited in its ability to model sophisticated reset schemes. Future improvements to the program will include additional control models.)

- For two-speed tower fan motors, studies have shown that the following simple strategy yields near optimum performance for many chiller/tower combinations. Stage low speed on each tower cell to maintain the minimum temperature the chiller manufacturer recommends (typically around 60°F but often expressed as a fixed difference above the leaving chilled water temperature). Stage high speed on each cell to maintain a setpoint equal to the design wet-bulb temperature. This strategy prolongs the time that the towers are operating at low speed, which is when they are most efficient.
- For variable-speed tower fan motors, the optimum setpoint will tend to be higher than with two-speed towers since the tower is most efficient when operating at low speeds. The above sequence for two-speed motors can be mimicked by controlling fan speed to maintain the manufacturer's recommended minimum setpoint, but limiting maximum speed from 50% - when the water temperature is at or below setpoint - proportionally up to 100% when water temperature reaches the design wet-bulb temperature.
- Resetting the condenser water setpoint to a fixed value above the outdoor air wet-bulb temperature has not proven to provide near optimum results. It also requires the use of a maintenance-intensive outdoor air humidity sensor.

Optimizing Control Sequences

There are a number of papers on optimization of controls for chilled water plants. In the June 2007 ASHRAE Journal article, "Optimizing Chilled Water Plant Control," Mark Hydeman and Guo Zhou present a process for control optimization that provides near optimal energy performance. The difficulty in applying this in practice is that there is no commercially available tool that automates these techniques. The article is available from the following link:

http://www.taylor-engineering.com/downloads/articles/ASHRAE%20Journal%20-%20Optimizing%20Chilled%20Water%20Plant%20Controls.pdf.

In brief, the technique involves developing a calibrated simulation model of a central plant that is run against an annual chilled water load profile with coincident weather data while parametrically studying all of the potential modes of operation at each hour to determine which mode is the most efficient. For a water-cooled plant, the parameters include:

- The number of chillers operated
- The speed of the condenser water pumps, and
- The speed of the cooling tower fans

The number of cooling tower cells is automatically calculated based on the maximum and minimum flow rates for each tower cell: at each time increment, the maximum possible number of cells is provided (as this always uses the least energy).

In the case study presented in the article, a direct control scheme based on the total plant load was found to be near optimal for the control of both the condenser water pump speed and the cooling tower fan speed. This is similar to work previously reported by Mick Schwedler and Tom Hartman in their many articles on the subject.

Thermal Storage

Thermal storage systems can offer significant energy cost savings where energy costs are based on time-of-day or real-time pricing. The design of these systems is beyond the scope of this Design Guide. For information, refer to ASHRAE's Design Guide for Cool Thermal Storage (Dorgan and Elleson, 1994).

Case Study

This example shows how the chiller plant optimization technique described in this chapter can be applied to a large high-rise office building. The sample building served is 15 stories high, enclosing 540,000 ft2 in San Francisco, California. The building primarily contains offices, with some assembly and retail space on the ground floor, a 5,000-ft2 data center, and a large cafeteria. The data center and various small server rooms operate continuously and place roughly a 50-ton base load on the plant. Total plant load was calculated to be 1,100 tons.

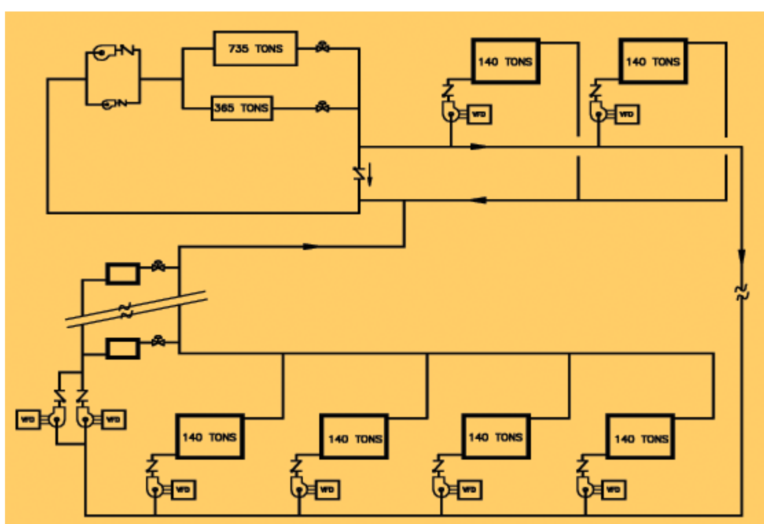
The HVAC system for most areas consists of six large air handlers serving a variable air volume distribution system. Auxiliary loads, such as the data center and server rooms, are served by chilled water computer room units or fan-coils.

Here is a list of steps followed to optimize the chiller plant:

1. Select chilled water distribution system flow arrangement (see Selecting Chilled Water Distribution System Flow Arrangement and Chapter 4).

   The system has six large air handlers plus various smaller fan-coils and computer room units. Based on Table 6-1, the application is partly application 3 (many large coils) and partly application 6 (many small coils). Therefore, a hybrid of the recommended systems for these two applications is selected, as shown in Figure 6-9. (The figure shows the final chiller selections which were determined later in the analysis.) The system is a primary-secondary system with the large coils served by distributed primary pumps located at each coil. The miscellaneous fan-coils and computer room units are served by a pair of conventional secondary pumps.

FIGURE 6-9:
EXAMPLE CHILLED WATER SYSTEM SCHEMATIC

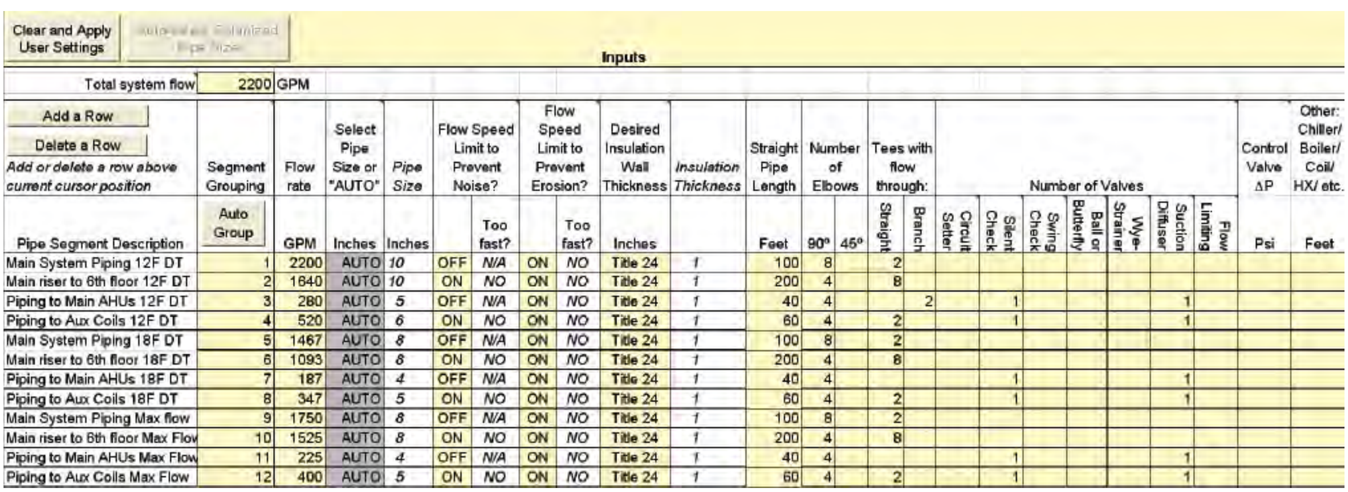


Note that a check valve has been provided in the common leg of the primary-secondary system to ensure that primary flow is always less than secondary flow. This places the primary and secondary pumps in series during transients such as cool-down (which are all but non-existent in the San Francisco climate).

2. Select chilled water temperatures, flow rate, and primary pipe sizes (see Optimizing Chilled Water Design Temperatures).

The overall plant capacity is 1,100 tons. The primary loads are the six 140-ton AHUs, two located on the 16th floor mechanical penthouse and four on the 6th floor. Another 260 tons of auxiliary load is served by the two traditional secondary pumps. This load includes a 75-ton data center (actual expected load of 50 tons), fan-coils serving server rooms, retail spaces, and miscellaneous equipment rooms, plus 125 tons for future unknown tenant loads. The primary piping will be at the chillers (total 1,100 tons), the main chilled water riser down to the 6th floor (820 tons), and the piping to the main fan systems on the 6th floor. Using 12°F and 18°F delta-Ts, the flows and pipe sizes (sized using the pipe sizing spreadsheet 2,000 hour/year operation) are:

Section	**Load, tons**	**Application**	**12°F ΔT**		**18°F ΔT**	
			GPM	**Pipe size**	**GPM**	**Pipe size**
Main system piping	1,100	Non-noise sensitive, variable flow	2,200	10”	1,467	8”
Main riser to 6th floor	820	Noise sensitive, variable flow	1,640	10”	1,093	8”
Piping to main AHU coils	140	Non-noise sensitive, variable flow	280	5”	187	3”
Piping to main auxiliary coils	260	Noise sensitive, variable flow	520	6”	347	5”


Clear and Apply User Settings

Inputs

Total system flow 2200 GPM

Add a Row

Delete a Row

Add or delete a row above current cursor position

Pipe Segment Description	Segment Grouping (Auto Group)	Flow rate GPM	Select Pipe Size or "AUTO" Inches	*Pipe Size* Inches	Flow Speed Limit to Prevent Noise?	Too fast?	Flow Speed Limit to Prevent Erosion?	Too fast?	Desired Insulation Wall Thickness Inches	*Insulation Thickness*	Straight Pipe Length Feet	Number of Elbows 90°	45°	Tees with flow through: Straight	Branch	Number of Valves: Circuit Setter	Silent Check	Swing Check	Ball or Butterfly	Wye-Strainer	Suction Diffuser	Flow Limiting	Control Valve ΔP Psi	Other: Chiller/ Boiler/ Coil/ HX/ etc. Feet
Main System Piping 12F DT	1	2200	AUTO	*10*	OFF	*N/A*	ON	*NO*	Title 24	*1*	100	8		2										
Main riser to 6th floor 12F DT	2	1640	AUTO	*10*	ON	*NO*	ON	*NO*	Title 24	*1*	200	4		8										
Piping to Main AHUs 12F DT	3	280	AUTO	*5*	OFF	*N/A*	ON	*NO*	Title 24	*1*	40	4			2		1				1			
Piping to Aux Coils 12F DT	4	520	AUTO	*6*	ON	*NO*	ON	*NO*	Title 24	*1*	60	4		2			1				1			
Main System Piping 18F DT	5	1467	AUTO	*8*	OFF	*N/A*	ON	*NO*	Title 24	*1*	100	8		2										
Main riser to 6th floor 18F DT	6	1093	AUTO	*8*	ON	*NO*	ON	*NO*	Title 24	*1*	200	4		8										
Piping to Main AHUs 18F DT	7	187	AUTO	*4*	OFF	*N/A*	ON	*NO*	Title 24	*1*	40	4					1				1			
Piping to Aux Coils 18F DT	8	347	AUTO	*5*	ON	*NO*	ON	*NO*	Title 24	*1*	60	4		2			1				1			
Main System Piping Max flow	9	1750	AUTO	*8*	OFF	*N/A*	ON	*NO*	Title 24	*1*	100	8		2										
Main riser to 6th floor Max Flov	10	1525	AUTO	*8*	ON	*NO*	ON	*NO*	Title 24	*1*	200	4		8										
Piping to Main AHUs Max Flow	11	225	AUTO	*4*	OFF	*N/A*	ON	*NO*	Title 24	*1*	40	4					1				1			
Piping to Aux Coils Max Flow	12	400	AUTO	*5*	ON	*NO*	ON	*NO*	Title 24	*1*	60	4		2			1				1			

Figure 6-10:
Example Chilled Water System Analysis in Pipe Size Optimization Spreadsheet

The smallest pipe sizes under the 18°F delta-T were selected and the flows and delta-Ts chosen to "max-out" the pipe size with the following results:

Section	Load, tons	Application	Pipe size	Maximum GPM	Resulting ΔT
Main system piping	1,100	Non-noise sensitive, variable flow	8"	1,750	15.1
Main riser to 6th floor	820	Noise sensitive, variable flow	8"	1,525	12.9
Piping to main AHU coils	140	Noise sensitive, variable flow	4"	225	14.9
Piping to main auxiliary coils	260	Noise sensitive, variable flow	5"	400	15.6

To make the 5" pipe work at the auxiliary coils, the delta-T must be at least 15.6°F. This can be a difficult delta-T to provide at small fan-coils. So two delta-Ts are selected, 16°F for the main AHU coils and 12.5°F for the auxiliary loads. This design reduces the pipe sizing in all branches except for the piping to the auxiliary coils. Hence, the design is based on the following flows and delta-Ts:

Section	Load, tons	Pipe size	Design GPM	ΔT
Main system piping	1,100	8"	1,750	15.1
Main riser to 6th floor	820	8"	1,340	14.7
Piping to main AHU coils	140	3"	210	16.0
Piping to main auxiliary coils	260	6"	500	12.5

Coil selections for the large air handlers were made assuming an 18°F delta-T (to allow for future heat transfer degradation). The maximum desired coil space at 6 rows required a 42.5°F chilled water temperature. A chiller supply water temperature of 42°F was selected to account for piping and pump heat gains.

3. Select tower speed control option, efficiency, condenser water temperature range and approach temperatures, and make preliminary cooling tower selection (see Selection of Condenser Water Design Temperatures and Cooling Tower).

   Given the office occupancy, two-speed motors were selected for the tower fans (see Tower Fan Speed Control). (A detailed analysis of variable-speed drives versus two-speed motors was undertaken for this project. Even with no bypass in the variable-speed drive, the drives were not cost effective over a 15-year life due in part to the condenser water reset strategy which kept fan speeds high to keep condenser water temperatures low, a strategy that was beneficial in this case due to the use of single stage chillers with variable speed drives.)

   Condenser water flows and delta-Ts were analyzed as follows. The condenser water piping was only at the plant and included the mains plus two anticipated branches to the two cooling tower cells and two chillers. (In this case, it turned out that final chiller selections did not have equal flow, but at this point in the analysis, equal flow was assumed.) Delta-Ts were selected at 10°F and 15°F since centrifugal chillers were anticipated. A chiller efficiency of 0.60 kW/ton was assumed.

Figure 6-11: Example Condenser Water System Analysis in Pipe Size Optimization Spreadsheet

Clear and Apply User Settings | Auto-select Optimized Pipe Size

Inputs

Total system flow 3090 GPM

Add a Row | Delete a Row

Add or delete a row above current cursor position

Pipe Segment Description	Segment Grouping	Flow rate	Select Pipe Size or "AUTO"	*Pipe Size*	Flow Speed Limit to Prevent Noise?		Flow Speed Limit to Prevent Erosion?		Desired Insulation Wall Thickness	*Insulation Thickness*	Straight Pipe Length	Number of Elbows		Tees with flow through:		Number of Valves							Control Valve ΔP	Other: Chiller/ Boiler/ Coil/ HX/ etc.
	Auto Group	GPM	Inches	Inches		Too fast?		Too fast?	Inches		Feet	90°	45°	Straight	Branch	Circuit Setter	Silent Check	Swing Check	Ball or Butterfly	Wye-Strainer	Suction Diffuser	Flow Limiting	Psi	Feet
Main System Piping 10F DT	1	3090	AUTO	*14*	OFF	*N/A*	ON	*NO*	Title 24	*0*	100	4		2										
Chiller/Tower Piping 10F DT	2	1545	AUTO	*10*	OFF	*N/A*	ON	*NO*	Title 24	*0*	40	4			2	1	1		2					10
Main System Piping 15F DT	3	2060	AUTO	*12*	OFF	*N/A*	ON	*NO*	Title 24	*0*	100	4		2										
Chiller/Tower Piping 15F DT	4	1030	AUTO	*10*	OFF	*N/A*	ON	*NO*	Title 24	*0*	40	4			2	1	1		2					10
Main System Piping Max Flow	5	2750	AUTO	*12*	OFF	*N/A*	ON	*NO*	Title 24	*0*	100	4		2										
																								Totals

Output Summary:	
Total Head (Feet)	**36.9**
Total Head w S.F. (Feet)	**41.9**
Total Cost	**$273,751**
Lifecycle Energy Cost	**$122,263**
First Cost	**$151,487**

Section	**Load, tons**	**Application**	**10°F ΔT**		**15°F ΔT**	
			GPM	**Pipe size**	**GPM**	**Pipe size**
Main system piping	1,100	Non-noise sensitive, variable flow	3,090	14”	2,060	12”
Piping to each chiller/tower	550	Non-noise sensitive, constant flow	1,545	10”	1,030	10”

Obviously within a reasonable range of DTs only the main system piping is at play. The chiller and tower piping will be 10” no matter what DT we choose. The smaller pipes are selected and the following flows and delta-Ts result:

Section	**Load, tons**	**Application**	**Pipe size**	**Maximum GPM**	**Resulting ΔT**
Main system piping	1,100	Non-noise sensitive, variable flow	12”	2,750	11.2

Because the delta-T ended up being so conservative, the design was based on a 12°F delta-T instead of 11.2°F as would be suggested by this analysis. (Recall that “maxing out” the pipe sizes is to provide conservatism. Since a 12°F delta-T is already conservative, the design was based on that so that pump savings and tower savings would result.)

Section	**Load, tons**	**Pipe size**	**Design GPM**	**ΔT**
Main system piping	1,100	12”	2,574	12
Piping to each chiller/tower	550	10”	1,287	12

Tower efficiency and approach were limited on this project due to severe space constraints. Several propeller fan selections were made from three vendors at the local design wet-bulb temperature of 64°F. Using standard tower selections, the largest two-cell size tower that would fit into the space provided a 9°F approach (73°F condenser water supply temperature). High efficiency towers were not evaluated given the space constraints.

4. Select chillers using performance specification and life-cycle analysis (see Selecting Type, Number, and Size of Chillers and Chapter 7).

   See the sample specification in Chapter 7. The final chiller selection was for two chillers sized for one-third and two-thirds of the load, respectively, each with variable-speed drives. The unequal sizing was favorable in this case due to the low base load of 50 tons from the data center, which was more efficiently handled by a smaller chiller.

5. Adjust tower sizing and number of cells if necessary (see Selection of Condenser Water Design Temperatures and Cooling Tower).

   Towers were initially selected for equal cell sizes. Once unequal chillers were selected this was reevaluated. Because of space constraints, unequal tower sizes would not have fit. In any case, maximum and minimum flow rates in each tower cell could be maintained even with the unequal condenser water pump sizes provided low flow distribution pans were selected for the towers. It was decided that the initial two-cell tower selection would be retained.

6. Finalize piping system design and select pumps (see Optimizing Pipe Design and Chapter 3).

7. Develop and optimize control sequences (see Sequence of Controls and Chapter 5).

The following control sequences were developed . Note the unusual chiller loading/unloading sequence due to the chiller variable-speed drives.

Sequence of Controls

Chiller plant

1. If a chiller fails or has been manually switched off, as indicated by its alarm contact or if its leaving water temperature remains 5°F above setpoint (see reset strategy below) for 15 minutes or if its kW is zero for 15 minutes while its on/off point is on, the chiller is placed in a high level alarm (Level 2). A failed chiller is not locked out, but the chiller stage where the failed chiller runs alone is locked out. (See staging below.)

2. If a chiller has been manually turned on, as indicated by a chilled water delta-T across the chiller greater than 3°F and chilled water supply temperature within 5°F of setpoint and kW>10% and its on/off point is off, the chiller stage where the other chiller operates alone shall be locked out and a low level chiller alarm (Level 4) shall be set. (See staging below.)

3. The chiller plant is enabled if any secondary pump is on for 2 minutes, and disabled if all secondary pumps are off.

4. Chillers are staged based on calculated load. Load is calculated by secondary delta-T and flow. Once both chillers are operating, load is calculated by assuming flow is balanced between chillers proportional to design flow.
5. Due to the chiller variable speed drives, it is more efficient to operate chillers at low load (above about 20%) than at high load. Thus the normal staging rules that tend to max-out chillers before staging the next one on do not apply.
6. Staging shall be:
   a. Stage 1: CH-1 on alone.
      A. Locked out if:
         i. CH-1 has failed, or
         ii. CH-2 is manually on.
      B. Minimum operating load: 0%
      C. Stage down point: none
      D. Stage up point: 30% of total plant load
   b. Stage 2: CH-2 on alone
      A. Locked out if:
         i. CH-1 is manually on, or
         ii. CH-2 has failed.
      B. Minimum operating load: 15% of total plant load
      C. Stage down point: 20% of total plant load
      D. Stage up point: 50% of total plant load
   c. Stage 3: CH-1 and CH-2 on
      A. Locked out if:
         i. Either CHP is in alarm.
      B. Minimum operating load: 30% of total plant load
      C. Stage down point: 40% of total plant load
      D. Stage up point: NA
7. Stage up (1 to 2 or 2 to 3) if the current stage is on and has been on for 15 minutes, and either:
   a. Any secondary pump is at full speed for 15 minutes and chiller load is above the minimum operating range of next stage, or
   b. The plant load becomes larger than the stage up point for the stage.
8. Stage down (2 to 1 or 3 to 2) if the current stage is on and has been on for 45 minutes, and the plant load becomes lower than the stage down point for the stage.

9. CHW pumps:

   a. Note that both pumps can serve either chiller given piping arrangement and minimum flow limitations.

   b. CHP-1 shall be logically interlocked to CH-1 and CHP-2 shall be logically interlocked to CH-2, except as noted below.

   c. If a pump fails or has been manually switched off, as indicated by its status point not matching its on/off point for 30 seconds after the pumps was turned on, the pump shall be placed into alarm and locked out (until reset by operator), and the other pump shall start automatically in its place. In other words, if a CHP fails, the other CHP will start whenever either pump operation is called for.

   d. If CHP-2 has been manually turned on, as indicated by its status point being on while its on/off point is off 30 seconds or more after the pump was commanded off, it shall run instead of CHP-1 for Stage 1. (If CHP-1 has been manually turned on, CHP-2 will still be operated normally during Stage 2; both pumps will run.)

   e. During Stage 1, CHP-2 shall be started then CHP-1 turned off 15 seconds later if the secondary flow exceeds the design flow rate of CHP-1 by 10% and the primary CHW return temperature is no lower than 0.5°F below the secondary CHW return temperature (indicating little or no bypass in the common leg). The two shall switch back in the same manner if, for 10 minutes, both the secondary flow is less than CHP-1 design flow and the primary CHW return temperature is lower than 0.5°F below the secondary CHW return temperature.

10. Chilled water supply temperature setpoint shall be reset within the range 40°F to 46°F using a reverse-acting control loop whose output is the CHWST setpoint, whose control point is the highest main fan system AHU-1 to 6 CHW control valve signal, and whose setpoint is 95% open. In other words, the CHWST is reset to provide water only as cold as required to satisfy the air handler with the largest cooling load. Both chillers shall be sent the same reset signal. (This sequence will result in 46°F when the main fan systems are off or not calling for cooling, which is most of the time. Non-economizer fan-coil systems must be designed to handle loads at this reset supply water temperature when outdoor air temperatures are below 55°F or so.)

11. Secondary pumps serving fan-coils (CHP-S-7, S-8, S-9, S-10):

   a. The pump shall start when any CHW control valve served by the pump is greater than 50% open for 5 minutes. The pump shall stop when all valves are 0% open for 3 minutes.

   b. Chilled water pump speed shall be controlled when the pump is commanded on to maintain chilled water differential at a fixed setpoint determined by the water system balancer. Minimum speed is 10%.

12. Condenser water system:

   a. Condenser water pumps are interlocked to the chillers via the chiller control panel and are not DDC controlled. Similarly chilled water isolation valves are interlocked to the chiller control panel chilled water pump contacts and are not DDC controlled.

b. Cooling tower fans are enabled when either CW pump is proven on. Cooling towers fans are lead-lag alternated based on running hours, with the lead change made when both fans are off or both are on. Lead cell isolation valve is opened and lead tower fan is enabled when the small CW pump CWP-1 is proven on. Both cell isolation valves are opened and tower fans enabled when the large CW pump CWP-2 is proven on, whether alone or with CWP-1. With CT-1 the lead fan, fans are staged to maintain CW supply water temperature at the low-speed setpoint as follows: both off, CT-1 low plus CT-2 off, CT-1 low plus CT-2 low. Again with CT-1 as the lead, fans are staged to maintain CW supply water temperature at the high-speed setpoint as follows: both low, CT-1 high plus CT-2 low, CT-1 high plus CT-2 high. The sequence is similar with CT-2 as the lead fan. For the first 15 minutes of chiller plant operation (after initial CH start), the low-speed setpoint shall be 70°F. Afterwards, the setpoint is reset as follows: Fan low-speed CWST setpoint is equal to the lower of the two CHW return temperatures from each active chiller plus 15°F, but no lower than 60°F and no higher than 70°F. The high speed CWST setpoint shall be 73°F. (The logic is to keep chiller head as low as allowed by the chiller manufacturer to maintain head pressure since tower fan energy is lower than the energy saved by the compressor when the fans are at low speed.)

c. When either CWP is on, conductivity in the CW system shall be monitored and bleed valve operated to maintain setpoint. Inhibitor shall be injected as a function of the bleed rate (time the bleed valve is open). Biocides shall be injected based on a time schedule, alternating each month. Setpoints for conductivity and time schedules for inhibitor and biocide shall be determined by the water treatment chemical supplier/specialist (see Water Treatment specification section).

13. Performance Monitoring

a) Chiller Performance Check. Chiller energy performance shall be calculated as:

$$kW/ton = \frac{kW}{Q_E}$$

Equation 6-1

where kW is the measured power to the chiller and Q is the calculated evaporator load in tons.

For one chiller on:

$$Q_E = GPM_S(T_{CHWS} - T_{SCHWR})/24$$

Equation 6-2

where GPMS is the secondary flow rate in gpm, T is temperature in °F, subscript CHWS refers to the chilled water supply leaving the chiller, and subscript SCHWR refers to secondary chilled water return. For both chillers on:

Equation 6-3

$$
\begin{aligned}
Q_{ET} &= GPM_S(T_{SCHWR} - T_{ACHWS})/24 \\
T_{ACHWS} &= \frac{GPM_{D-1}T_{CHWS-1} + GPM_{D-2}T_{CHWS-2}}{GPM_{D-1} + GPM_{D-2}} \\
X_1 &= GPM_{D-1}(T_{PCHWR} - T_{CHWS-1}) \\
X_2 &= GPM_{D-2}(T_{PCHWR} - T_{CHWS-2}) \\
Q_{E1} &= \frac{X_1 Q_{ET}}{X_1 + X_2} \\
Q_{E2} &= \frac{X_2 Q_{ET}}{X_1 + X_2}
\end{aligned}
$$

Where subscripts D means design (per equipment schedules), 1 means CH-1, 2 means CH-2, ACHWS is average chilled water supply, PCHWR means primary chilled water return, and SCHWR means secondary chilled water return.

Also calculate predicted performance from regression equations developed from manufacturer's data:

Equation 6-4

$$
\begin{aligned}
CAP_{FT} &= a + bT_{CHWS} + cT_{CHWS}^2 + dT_{CWS} + eT_{CWS}^2 + fT_{CHWS}T_{CWS} \\
EIR_{FT} &= a + bT_{CHWS} + cT_{CHWS}^2 + dT_{CWS} + eT_{CWS}^2 + fT_{CHWS}T_{CWS} \\
PLR &= \frac{Q_E}{CAP_D CAP_{FT}} \\
EIR_{PLR} &= a + bPLR + cPLR^2 \\
kW_{pred} &= kW_D * CAP_{FT} * EIR_{FT} * EIR_{FPLR} \\
kW/ton_{pred} &= \frac{kW_{pred}}{Q_E}
\end{aligned}
$$

where CAP is the capacity of the chiller, EIR is the electric input ratio (like kW/ton but non-dimensional), PLR is the part load ratio, kW is the power input to the chiller, subscript FT means "as a function of chilled and condenser water supply temperatures," and subscript PLR means "as a function of part load ratio."

Regression coefficients for CH-1 and 2 are as follows:

CH-1 -- Carrier 19XR 3031 w/ VFD:

Design CAP_D	Design kW_D
365	201

Curve	a	b	c	d	e	f
CAP_{FT}	1.5756009E+00	-7.2779520E-03	-3.3153877E-04	-1.2075224E-02	-1.3519250E-04	6.2585781E-04
EIR_{FT}	1.4774920E+00	-3.5351047E-02	5.0752172E-04	2.3695355E-03	1.7902557E-04	-3.3549182E-04
EIR_{PLR}	3.5713682E-01	-5.2674945E-01	1.1963575E+00			

CH-2 -- 19XR 6565 w/ VFD:

Design CAP_D	Design kW_D
735	367

Curve	a	b	c	d	e	f
CAP_{FT}	1.0000000E+00	0.0000000E+00	0.0000000E+00	0.0000000E+00	0.0000000E+00	0.0000000E+00
EIR_{FT}	-1.4166071E+01	6.6438356E-02	-3.9138943E-04	3.8095890E-01	-2.4334638E-03	-5.7925636E-04
EIR_{PLR}	1.0735092E+00	-2.1862650E+00	2.1154485E+00			

Calculation shall be done whenever chiller power exceeds 10% power and the chiller has operated for at least 10 minutes. Accumulate over time and calculate average performance, both measured and predicted. Display both instantaneous and average kW/ton on chiller graphic.

7. PROCUREMENT

This chapter discusses general strategies for procuring chilled water plant design and construction services. It also recommends specific procedures for evaluating chiller options and selecting an energy-efficient and cost-effective chiller. Case studies of the chiller selection process are provided for a new building and a retrofit project. Also provided are a sample chiller bid specification (Appendix A) and a sample chiller bid form (Appendix B).

Design and Construction Approaches

There are a number of strategies for designing and building chilled water plants. The preferred approach varies depending on the qualifications of local engineers and contractors. For a detailed discussion of design and construction approaches, see the CoolTools document, *Project Implementation Plan: Achieving Successful Chilled Water Plants*. What follows is a brief discussion of the most common approaches, including:

- Conventional Design/Bid/Build
- Negotiated Design/Build
- Negotiated Design/Assist
- Design/Build-Bid and Design/Assist-Bid

This section also provides some general recommendations about design and construction procurement strategies.

Conventional Design/Bid/Build

With this approach—perhaps more commonly called the Plan & Spec' approach—a consulting engineer designs the project and prepares a set of plans and specifications that are put out to bid to qualified contractors. Generally the contractor with the lowest bid is awarded the contract. This was by far the most common chiller procurement approach in the United States from post-World War II until the early 1980s, when the design/build strategies became more common.

Plan & Spec' is best used on projects where negotiated design/build or negotiated design/assist are not possible (for example, state and federal institutional projects where laws require contracts to be awarded based on the lowest bid). Plan & Spec' also may be the best approach if the project is complex or if it is difficult to provide a definitive scope of work early in the project, a necessary step for the Design/Build-Bid approach. If the consulting engineer's design is a good one, the Plan & Spec' approach can result in low first costs through the competitive bid process. There are disadvantages to Plan & Spec', however:

- Many consulting engineers do not design the most cost-effective systems, either because they lack intimate knowledge of system costs and construction issues or because of liability concerns. The low-bid process then produces the best price for the system the consultant designed, but that design may not be the most cost effective for the application.

- The low-bid process can create an adversarial relationship between the contractor, engineer and owner. The contractor often takes a "deaf and dumb" approach, providing no more than what is shown on plans and offering little help in resolving problems. Some contractors are "change-order artists": they take advantage of design errors or oversights to increase their margin, thereby reducing or eliminating the initial low-cost advantage of the competitive bid process.
- The conventional design/bid/build approach creates multiple sources of responsibility. If system performance problems occur, it is often difficult to establish whether the consultant or contractor is responsible.

Negotiated Design/Build

With this approach, a consulting engineer is not needed; the design/build contractor provides all engineering services, preferably using their in-house staff. Three elements are essential for a successful negotiated design/build project:

- The design/build contractor must have good engineering talent and a solid reputation for doing design/build projects. Many contractors think they understand how to engineer systems but do not.
- It is essential that the owner work with a design/build contractor with whom they have a long-term working relationship. This relationship helps to ensure that prices are reasonably competitive. The contractor will view this project as one in a string of many and not try to maximize profit on this particular project. The fact that the project is negotiated with one contractor rather than put out to bid helps to ensure a positive "team" attitude and avoids the adversarial relationship that can result from the low-bid process. However, without a bid process, there may be less incentive for the contractor to minimize profit margins.
- It is crucial that a detailed specification be developed early in the project to establish the project's scope. If a system cost is established early in the project before a detailed specification is prepared, the contractor may have an incentive to reduce the quality or scope of their work to conform to the promised cost.

Negotiated design/build usually results in designs that are practical and cost-effective, although not necessarily imaginative. Hoping to avoid the start-up problems and risks associated with more complex, innovative designs, contractors often opt for rather conservative designs. One of the primary advantages of design/build is single-source responsibility for the performance of the system, which avoids the "finger-pointing" common to Plan & Spec' projects.

Negotiated Design/Assist

With this approach, a design/build contractor and an engineering consultant team up during the design phase. One advantage over design/build is the synergy between the consultant and the contractor's engineers, often resulting in a design that is better than either would have developed alone. As with negotiated design/build, negotiating construction costs with the contractor helps to ensure the proper team attitude, but it is essential that the owner and contractor have a long-term relationship to ensure reasonable pricing.

Design/assist has higher engineering costs than design/build because there is some overlap in engineering responsibilities. This split in responsibilities also does not always provide the single-source responsibility benefits offered by the design/build approach. This can be mitigated, however, by requiring the contractor to have primary liability for the design. Many owners feel more comfortable with this approach than with design/build because the consultant can represent the owners' interests without the apparent conflicts of interest facing the design/build engineer.

Design/Build-Bid and Design/Assist-Bid

In these variations of design/build and design/assist, the project is bid rather than negotiated. In either case, a consultant is retained to write a performance specification that becomes the basis of the bid. Since there are no Plans & Spec's, it may be a challenge to compare contractors' proposals—the proposals are not always "apples to apples." The consultant must decide which proposal has the best value. This may or may not be the proposal with the lowest price tag. Bidding the project has the advantage of reducing costs and encouraging bidders to develop "a better mouse trap," in other words, to come up with intelligent designs that reduce costs. On the other hand, bidding can result in the adversarial relationship common to Plan & Spec' projects. Also, the incentive to be the lowest bidder can drive contractors to cut corners. This approach can also lead to problems if the performance specifications do not thoroughly describe the scope and coordination issues. The key to a successful design/build-bid or design/assist-bid is a specification that is comprehensive enough to prevent corner-cutting, yet flexible enough to encourage intelligent and original designs.

General Recommendations

The negotiated design/assist or negotiated design/build approaches are recommended because of the positive team attitude they engender. However, many owners are not willing to negotiate and insist on bidding projects to get a lower price. With the bidding process, the price is usually lower up front, but by the time the project is complete the price may have significantly increased due to change orders and inefficiencies caused by adversarial relationships. If the project must be bid, the design/assist-bid approach is preferred because it leads to a better engineer/contractor relationship than the design/bid/build approach, and it usually results in a better design due to the contractor's input.

With all of these approaches, the chillers can be selected using the life-cycle cost techniques described in this chapter. However, the selection process is simpler when a contractor is involved; the contractor often has more clout with the vendors than consultants do, and can therefore obtain better pricing. Many owners and general contractors balk at consultants obtaining pricing because they are not confident that the consultant will have the buying power of a mechanical contractor. Having the contractor on board early to conduct the chiller bid is another advantage of the design/assist or design/build approaches compared to conventional design/bid/build.

While having a contractor obtain pricing is preferred, consultants on design/bid/build projects should not be discouraged from using this approach. The new building case study in this chapter presents an actual—and successful—design/bid/build project in which the chillers were priced by the consulting engineer. The sample chiller bid specification contains language to help consultants obtain competitive pricing when they are conducting the chiller bids.

Chiller Procurement Procedures

Table 7-1 compares typical versus recommended approaches for procuring chillers. The typical approach, shown in the table's left column, is expedient but it seldom leads to an optimum selection. The recommended approach is outlined in the table's right column and is discussed in more detail in the subsequent sections of this chapter.

Table 7-1: Chiller Procurement Approaches

Typical Approach	Recommended Approach
1. Calculate or estimate the required plant total tonnage.	1. Calculate or estimate the required plant total tonnage.
2. Decide how many chillers are needed. This is usually a somewhat arbitrary decision, although it often is a function of space constraints or the need for a certain level of redundancy.	2. Pick a short list of chiller vendors based on past experience, local representation, etc.
3. Calculate chiller size by dividing the total tonnage by the number of chillers. Each chiller is equally sized.	3. Request chiller bids based on a performance specification.
4. Pick a chiller manufacturer, usually based on past successful experience or perhaps based on which vendor has the most attentive (or generous) sales representatives.	4. Adjust bids for other first-cost impacts.
5. Have the chiller vendor make a few recommendations, perhaps with first-cost budgets attached to each.	5. Estimate energy usage of options with a detailed computer model of the building/plant.
6. Decide which chiller to use based on instincts and perhaps past experience, with little or no analysis.	6. Estimate maintenance cost differences between options.
	7. Calculate life-cycle costs.
	8. Select the chiller option with the lowest life-cycle cost.

The recommended approach has many advantages:

- It generally results in a better chiller selection.
- The owner benefits from lower life-cycle costs.
- Energy costs are usually lower than what would have resulted from the more conventional procurement procedure.
- Chiller vendors can make proposals that take advantage of their systems' strengths or "sweet spots" both for cost and efficiency. The conventional approach, where size and efficiency are more arbitrarily selected, usually favors one vendor (inadvertently or intentionally) who happens to have a "sweet spot" for the selected piece of equipment.
- In design/bid/build projects, the chiller selections are finalized in the design stage; therefore, no substitutions and associated redesign will occur during the bid phase.

There are also disadvantages to the recommended approach:

- It takes more time, both for the engineer and the equipment vendors.
- Errors in the energy calculation may skew the selection. For instance, utility rates, internal loads, and occupancy patterns assumed in the analysis may be incorrect or may change over time.

- The computer model used to calculate energy usage is imperfect in the way it models building and system loads, the chiller plant and pumping systems, and plant control strategies. Optimum plant control strategies are often complex and usually vary from one chiller option to another. These strategies cannot always be modeled accurately by existing energy simulation tools.
- The benefits of long-term product reliability and vendor support are seldom included in the life-cycle cost calculations because their cost benefits are difficult to estimate, although they may be significant.

Despite these disadvantages, the recommended chiller procurement approach represents an improvement over conventional approaches and has been successfully used on several projects. The case studies at the end of this chapter are based on actual projects that followed this recommended approach.

Procurement Step #1: Calculate Plant Tonnage

See Chapter 2 for an in-depth overview of available methods of calculating or estimating the required plant total tonnage.

The chiller procurement procedures described below assume that chiller plant design parameters, such as entering and leaving condenser and chilled water temperatures, have been determined. For retrofit applications, these design parameters are typically set by the constraints of the existing system design. or new chiller plants, Chapter 6 includes procedures for selecting these design parameters.

Procurement Step #2: Develop Vendor Short List

After determining plant loads, pick a short list of chiller vendors based on past experience, local representation, and other factors. Not all of the major vendors need to be on the bid list; however, the more vendors on the bid list, the better the competition will be and therefore the lower the chiller energy costs and first costs are likely to be. But low first costs and energy costs are not the only selection criteria; low maintenance costs, system reliability, vendor support, and other issues are also considerations.

However, just as it is often best to negotiate with one contractor rather than put the project out to bid, in some cases it may be best to negotiate with a single chiller vendor. Optimizing chiller selection with a single vendor is still possible because within a vendors' product line there are many options (for example, larger condensers and evaporators, variable speed drives, etc.).

Procurement Step #3: Obtain Chiller Bids

Once a bid list is established, develop a bid *performance specification*. In the conventional procurement approach, a chiller is typically specified by capacity, efficiency, construction details, etc. These details, however, are not usually appropriate for a *performance specification*. The vendor should be encouraged to propose chillers that take advantage of their product's strengths or "sweet spots." Although redundancy requirements must be taken into account, it is important not to otherwise constrain chiller size.

As a rule, do not include the following in the performance specification, but rather leave them up to the bidders:

- Number of chillers
- Chiller size
- Chiller efficiency
- Variable speed or fixed speed

Encourage multiple options from each vendor. They may, for instance, propose one design that uses several equally sized chillers and another that includes unequally sized chillers or a *"pony" chiller*. Another option may include variable speed drives on one or more of the chillers. Encourage the vendors to be imaginative.

Eliminate unnecessary boilerplate requirements. Conventional specifications often contain a long list of construction details that are sometimes proprietary and usually not very important from a quality and reliability perspective. If a chiller vendor has been approved for the bid list, that should mean that the vendor's standard construction details are acceptable. Including endless boilerplate requirements in the specifications usually is a waste of time, and can be counter-productive if they mask truly important specification details. Most of the boilerplate requirements have been eliminated from the attached sample chiller bid specification.

Include the following details in the specification:

Total plant tonnage

Specify only total capacity, not the capacity of each chiller. See Chapter 2 for an overview of methods of determining plant total tonnage.

The minimum number of chillers or required level of redundancy

For redundancy and reliability, the specification may require, for instance, that there be at least two chillers in the plant, or possibly at least two compressors to allow for dual compressor options. Or for critical plants, such as those serving an industrial process or data center, the specification may require an "n+1" redundancy, meaning that the chiller plant must be able to handle the load even with the failure of the largest chiller.

Design entering and leaving water temperatures of the plant as a whole

In Chapter 6, the section Optimizing Chilled Water Design Temperatures describes how to determine these parameters for new buildings. For retrofit applications, these factors are usually fixed by the constraints of the existing design.

Energy sources available, such as electricity or gas

This specification is not limited to electric chillers, although mixing gas and electric chillers will require more cost adjustments to account for the larger heat rejection required by absorption chillers, and for differences in gas and electric utility service sizes and distribution systems. (Similarly, thermal storage options may also be evaluated, but this requires substantial additional work to estimate costs for storage devices and the spaces required to house them.)

Typical cooling load profile

In order to offer intelligent proposals, the chiller vendors will need to know the cooling load profile. For instance, if there are many hours at low load, the vendor may wish to propose a small "pony" chiller or a variable speed drive. Figure 7 1 shows a typical load profile for a system with a relatively small data center that provided a constant base load. This profile was developed from a computer model of the new building case study provided in this chapter. (For a more detailed discussion of cooling loads, see Chapter 2.)

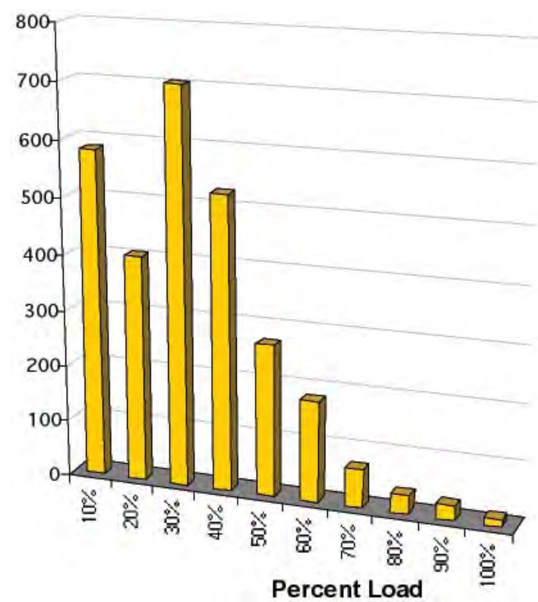


Figure 7-1:
Example Office Building
Load Profile

There are a number of design constraints that may affect chiller options and should be described in the bid performance specification. These include:

- Limited mechanical room space. If this is a factor, include a plan of the mechanical room in the specification.
- Available voltage and, for retrofit projects, available capacity of the electrical service.
- Type of allowed refrigerants. For instance, the owner may prefer non-ozone depleting refrigerants such as R-134a and disallow R-123 as an option. (See Chapter 3 for additional comments on refrigerant issues.)
- Noise. If noise is a design constraint because the chiller is located adjacent to noise-sensitive spaces, have an acoustical engineer back-calculate the maximum chiller sound power levels, and include these in the performance specification. The proposed chillers would either have to meet these sound power limits inherently or the vendor would have to include a chiller sound enclosure in their proposal.

Options

Make sure that specifications clearly state which standard options must be included in the pricing, such as freight to the jobsite, water box insulation, sales tax, etc.

Witnessed factory tests

It is highly recommended that factory tests be required, despite their cost. (See the sample bid specification for suggested language about testing requirements.) There are three reasons for conducting witnessed factory tests:

- Tests are more easily and less expensively done in the factory than in the field; doing the factory tests obviates the need for field tests for commissioning or performance verification purposes.
- Factory tests make it possible to reject the chiller if it does not meet performance specifications. Once a chiller is installed in the field, rejecting the chiller is no longer a practical option.
- Testing is more accurate in the factory than in the field, resulting in less controversy about whether and to what extent chillers fail to meet performance requirements. If a witnessed factory test is not called for, the specification should require that printouts of the ARI-certified performance program be provided along with the completed performance data forms from the chiller simulation modeling.

Liquidated damages

If performance tests are required, it is recommended that monetary penalties apply if a chiller fails to meet performance requirements. Without “teeth” in the specification, vendors will be more likely to exaggerate chiller performance in their proposals. An example liquidated damages clause is included in the sample bid specification.

Zero ARI tolerance

ARI Standard 550/590-98 allows a tolerance in the measured versus actual chiller capacity and efficiency. The magnitude of the tolerance varies as a function of load, increasing as load decreases as shown in Figure 7-2 .

The tolerance was intended to account for variations in manufacturing, but over time manufacturers have found that their manufacturing tolerances are smaller the ARI Standard tolerances. Market forces have then caused manufacturers to take credit for the tolerance in the chiller performance reported by their rating programs. In other words, the power reported by manufacturers’ rating programs for a given chiller has been decreased from the actual power required by an amount equal to the allowed ARI tolerance. Obviously, to get an accurate picture of predicted chiller performance in the life-cycle cost analysis, actual chiller performance data must be obtained from chiller vendors. It is recommended, therefore, that the performance claims by manufacturers be required to have “zero ARI tolerance.” See the sample bid specification for suggested language.

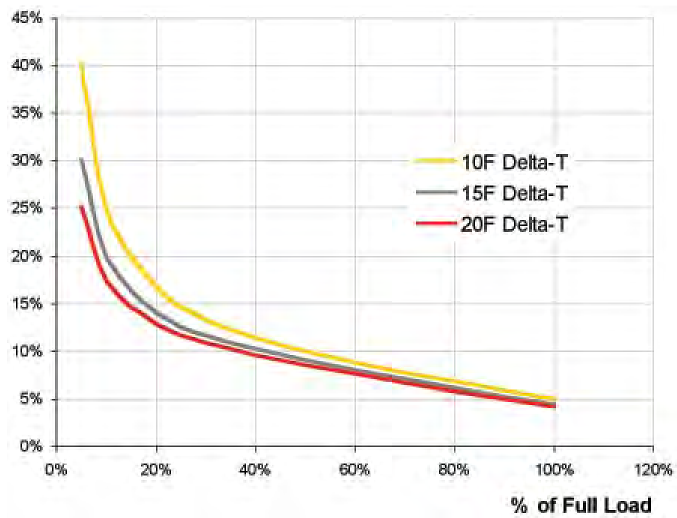


Figure 7-2:
ARI 550/590 Tolerance

Performance forms

To create chiller simulation models using the CoolTools Chilled Water Plant Analysis Program (CWPAP), chiller performance data over a wide range of operating conditions is required. A sample performance form is included in this chapter. The range of operating conditions will vary from project to project, so the form may have to be modified to ensure that it includes all operating conditions expected for the particular project.

Procurement Step #4: Adjust for Other First-Cost Impacts

After obtaining chiller bids, make adjustments to account for other installation cost impacts. Chiller prices obtained through the bidding process are not necessarily directly comparable. Here are some reasons why prices may not be "apples to apples":

- One option may have more chillers than another (for instance, three versus two). The added cost of rigging, piping, wiring, and controls for the extra chiller must be accounted for.
- Three-pass evaporators can increase piping costs compared to two-pass evaporators.
- An open-drive compressor may require more chiller room ventilation (or cooling) than hermetic motor options.
- One option may require field-mounted variable speed drives as opposed to factory-installed drives.
- One option may have less efficient chillers which can affect electrical service costs.
- One proposal may include a control system that can be directly connected to the energy management system network, while another may require a gateway, and still another may not be able to connect to the system at all, requiring that essential control points be hardwired to the EMS.

Estimating the costs of these secondary impacts is best handled by a contractor, which is another advantage of the design/build and design/assist approaches. For the first pass of the life-cycle cost analysis, these other first-cost impacts can be roughly estimated. Then, if the chillers for which rough estimates were made end up with the lowest life-cycle costs, a more accurate estimate of the secondary impacts should be made.

Procurement Step #5: Estimate Utility Costs

One of the most important elements of the chilled water plant procurement process is accurately estimating the utility costs of each chiller option. Utility costs include electricity, gas (if applicable), and water (if both water- and air-cooled chillers are being considered). The level of accuracy and detail necessary for energy calculations depends on the project's size and engineering budget. Even for very small projects, tools such as CoolTools Chilled Water Plant Analysis Program have made it practical and affordable to use chiller plant computer simulation models to accurately estimate chiller plant performance.

The primary engineering cost of the computer model is not creating the chiller plant model itself, but creating the model of the building and systems generating the plant load. For smaller projects or those with lower engineering budgets, the cost of creating the plant profile can be substantially reduced by using prototypical profiles developed from generic models of buildings having similar occupancy and HVAC systems. For large, new projects, creating a detailed building and HVAC system model is generally affordable and will improve the accuracy of the results. For existing projects, measured performance data may be used.

Since an accurate energy simulation model is essential to the chiller selection procedure, it is important that the engineer creating the model be experienced with both chilled water plant design and the simulation tool. (See Chapter 2 for a more detailed discussion of load and energy calculation programs.)

Once a computer model of the building and HVAC systems is complete and performance bids from chiller vendors have been obtained, create models of each of the proposed chillers using the performance data included on the forms completed by the vendor as part of their proposal. (See sample chiller performance form.) If using the CoolTools chiller model creation tool, enter the data into this program, which then calculates DOE-2 regression coefficients.

The following factors should be considered in the simulation:

Utility rates

Utility rates vary over time and are difficult to predict. There are a number of energy forecasts that project electricity and natural gas costs up to 30 years in the future (e.g., DOE/EIA, AGA., GRI) that attempt to account for the impact of deregulation as well as world oil and gas reserves. However, these reports are typically generalized for the nation or the world as a whole and may not reflect local utility rate trends. Given this uncertainty and for simplicity, it is typically assumed that current rates, or something similar, will be in effect during the chiller plant's life cycle.

Virtually all utilities charge both for energy consumption as well as for demand. Since chillers are one of the largest energy users in typical buildings, it is essential that demand charges be properly taken into account. This is particularly true when demand charges are ratcheted, meaning the owner pays some percentage of the maximum peak demand over the year, regardless of actual monthly demand.

Tower fan control (condenser water setpoint reset)

The optimum condenser temperature control varies by the type of chiller, the chiller load, the size and efficiency of the cooling tower, tower fan control options (for example, two-speed or variable speed), and the outdoor air wet-bulb temperature. Some plants have their lowest energy usage if the cooling tower fans are controlled to attempt to maintain the minimum condenser water temperature at which the chillers are capable of operating. If the tower fans have two- or variable-speed control, the plant may use less energy at higher setpoints depending on the plant load and outdoor air wet-bulb temperature. In most cases, the optimum control scheme is not easily determined. (See Chapter 6, Optimizing Control Sequences, for an additional discussion of condenser temperature control.)

In addition, CoolTools Chilled Water Plant Analysis Program (CWPAP) and other chiller plant simulation tools are limited in their ability to model different tower control options. It is expected the CoolTools CWPAP program will be enhanced in the future to simulate optimum tower control.

In the meantime, to properly estimate the energy performance of each chiller option, and to gain insight into how to best control the plant once it is built, it is recommended that various tower control strategies be modeled. For instance, different settings for the leaving condenser water setpoint can be modeled, including:

- the minimum allowed by the chiller manufacturer (this strategy will usually be optimum for open-drive, single-stage compression chillers, but not for multi-stage compression chillers);
- the design wet-bulb temperature (this simple rule-of-thumb surprisingly results in very good performance in many climates); and
- a constant approach to outdoor air wet-bulb temperature (this strategy requires that a humidity or wet-bulb sensor be installed in the control system which increases maintenance costs and reduces reliability).

For towers with two-speed fans, the optimum control strategy will most likely be to control low speed and high speed to different setpoints. For instance, the low-speed setpoint could be controlled to the minimum allowed by the chiller manufacturer while the high-speed setpoint could be controlled to the design wet-bulb temperature. Unfortunately, current simulation tools are not able to model this strategy.

Staging

Although the chiller vendor proposes the size and performance of chillers, the design engineer must determine how the chillers are to be staged, both in the computer model and in "real life."

Fixed-speed chillers are almost always optimally controlled by running the fewest chillers necessary to meet the load. This is easily modeled by most plant simulation tools and is usually the default control strategy.

Variable-speed chillers, however, use less energy when they are operating at low load, except at very low loads. Therefore, a plant with multiple variable-speed chillers will be more efficient when more chillers are operating at part load than if fewer are operating near full load, despite the additional energy used by chilled and condenser water pumps. (See Optimizing Control Sequences in Chapter 6 for an additional discussion of optimizing chilled water plant design.) For estimating energy usage of chiller plants with multiple variable-speed chillers, two or more stage-on points should be modeled. For instance, chiller staging could be controlled so that multiple chillers never operate below 25% load for one run and below 35% load for another.

Chilled water reset

Most chiller plants use some form of chilled water setpoint reset as a control strategy to reduce chiller energy usage. The strategy's energy effectiveness depends on:

- the types of chillers (for instance, screw chillers and absorption chillers benefit more from reset than centrifugal chillers);
- the type of pumping system (constant volume pumping system will benefit more than variable flow systems since reset tends to increase chilled water flow rates);
- the range of reset that is possible based on the loads served (for instance, relatively constant loads such as data centers may not allow much reset, and reset may also be limited in humid climates to maintain dehumidification capability); and
- how the reset is controlled (see Chilled Water Temperature Reset in Chapter 6 for alternative schemes and their limitations).

The computer model's ability and accuracy to simulate reset is a significant issue. The program must be able to model the effect reset has on cooling-coil effectiveness, and should be able to allow the user to limit reset within a certain temperature range. Few computer tools are capable of doing this well even when the coils are being modeled by the program. If all the chillers being evaluated are of a similar type (for example, all are centrifugal chillers), it may be best not to model reset to avoid skewing the results due to coil modeling errors. When coils are not being modeled, such as when prototypical load profiles or actual load profiles are used, it is not possible to model reset at all unless these profiles also include corresponding chilled water temperature profiles.

Variable-flow pumping systems

Most plant computer models are limited in their ability to accurately model variable-flow pumping systems, particularly primary-only systems. It is often necessary to "fake" the program into modeling the actual system by changing default performance curves or adjusting pump heads. This can require considerable judgment on the part of the engineer doing the modeling. Fortunately, in most cases, the pumping scheme is the same for all the chiller options being considered, so errors in the model tend to cancel out when options are compared.

The important differences between chiller options that must be accounted for are variations in condenser and evaporator pressure drops and, for variable-flow primary systems, the minimum and maximum evaporator flow rates. Pressure drop differences can usually be accurately accounted for by adjusting pump heads in the computer model, and in any case have a small impact on overall energy. Differences in minimum and maximum flow rates with variable-flow primary systems cannot be modeled well using current simulation programs. However, the impact of varying minimum flow rates is usually small unless there are large differences between options and many hours when the plant operates at low loads. The impact of varying maximum flow rates is seldom an issue because the maximum rates seldom occur unless the system experiences very bad delta-T degradation. (It is expected that future versions of CoolTools CWPAP will be capable of modeling all of the distribution systems recommended in Chapter 6 and most of the schemes shown in Chapter 4, so that these nuances can be accurately taken into account.)

Another complexity of variable-flow primary systems is the impact of flow variations on chiller efficiency and capacity. This cannot easily be modeled by most simulation tools, and cannot be modeled with the current CoolTools CWPAP and DOE-2 software. However, in most cases the impact is small due to offsetting effects of increasing mean temperature difference (which improves efficiency) and reducing inside film heat transfer coefficient (which reduces efficiency) as flow decreases.

Procurement Step #6: Estimate Maintenance Costs

Maintenance costs are more difficult to estimate accurately than energy costs. There is little data available indicating the relative maintenance costs of various chiller types (for example, screw, centrifugal, absorption, engine-driven, etc.) and among the various manufacturers of each chiller type. Manufacturers make claims about their products' advantages, but they seldom have hard, independently collected data to support those claims. To further complicate the issue, annual maintenance costs are not constant. The costs are low for the first few years, jump during years when a complete overhaul is required, and increase gradually as equipment wears. The length of time that different pieces of equipment last before they must be replaced also varies, although usually this is not an issue in chiller selection unless very long life cycles are analyzed.

Because maintenance costs are difficult to estimate, they are often ignored in the chiller selection life-cycle costing and considered only as a "soft issue" used when making the final chiller selection. This is probably a reasonable approach when the number and types of chillers are the same. However, when the number of chillers in each option varies, both air- and water-cooled options are being considered, or chillers of different types (e.g. electric and gas-engine driven) are being considered, maintenance costs need to be included explicitly in the life-cycle cost calculation for best results.

Factors that affect maintenance costs include:

Gear versus direct drive

Gear-drive machines have slightly higher maintenance costs than direct-drive machines.

Open-drive versus hermetic motors

Open-drive machines have slightly higher maintenance costs than hermetic machines, although they are less costly to repair should there be a motor burnout (in chillers with hermetic motors, motor burnout typically contaminates the refrigerant).

Variable-frequency drives

Variable-frequency drives introduce another component in the system subject to failure. Little data is available on their reliability since the latest generation of VFDs is fairly new.

Varying manufacturer quality

There are many claims and anecdotes but little data on who makes the best chiller. For the purpose of maintenance cost analysis, it is generally assumed that if the chiller manufacturer is on the bid list, they make a reliable product.

Compressor type

Screw chillers will usually require less maintenance than centrifugal chillers, although this varies from manufacturer to manufacturer.

Number of chillers (for options where the number of chillers differ)

The more chillers in the system, the higher the maintenance costs will be even for the same total plant capacity. The costs for maintaining a chiller are not strongly dependent on the chiller's size. If costs are not available, a reasonable estimate of annual maintenance cost per water-cooled chiller is about $2,000 to $4,000 for the first few years, with an additional $1,000 per year in repairs after the first five years or so. There is a slight increase in the per-machine maintenance cost at about 1,000 tons and there may be another incremental jump at 2,000 tons and up.

Air- versus water-cooled

Whether the system is air- or water-cooled has by far the most significant impact on maintenance costs among the issues listed here. Water-cooled systems will always cost more to maintain due to the constant water treatment requirements and the need for regular tube cleaning. Water-cooled chillers will generally last longer, however, particularly in harsh environments such as near oceans where salt in the air can significantly shorten the life of air-cooled condensers.

To estimate the differences in maintenance costs between air- and water-cooled systems, request input from local HVAC service companies. The ASHRAE *Handbooks* series also provide some maintenance cost information. The retrofit building case study in this chapter compares air- and water-cooled chillers in a retrofit application. Absent other information, a reasonable estimate of annual maintenance cost savings is $1,000 to $2,000 per year per chiller for the first 5 to 10 years (plus the cost of cooling tower maintenance and chemicals).

Gas-engine driven chillers

Gas-engine driven chillers are known to have much higher maintenance costs than either electric or absorption chillers. These chillers have frequent engine-related maintenance requirements (e.g., for spark plugs, oil/oil filters, air cleaners, belts, etc.) that are not required for other chiller types. Manufacturers of these chillers can provide estimates of the frequency and cost of this work.

Procurement Step #7: Calculate Life-cycle Costs

The life-cycle cost of a chiller plant is the present value of the total cost of owning and operating the plant over a specified period of time. A detailed description of the parameters used in calculating life-cycle cost is beyond the scope of this Design Guide. For more details, refer to the CoolTools *Project Implementation Plan: Achieving Successful Chilled Water Plants* and other engineering manuals. Below is a brief summary of the relevant variables and formulas for calculating life-cycle costs. These formulas should be entered into a spreadsheet so that the sensitivity of various assumptions can be evaluated. (See the **new building case study** for an example that uses these equations.)

The life-cycle cost can be calculated using the following equation:

$$LCC = FC + \sum_{j=1}^{N} \frac{UC_j + MC_j}{(1+d)^j}$$

Equation 7-1

Where,

LCC = present value of the owning and operating costs of the chiller plant.

FC = first costs of the plant. In this case, this is the cost of the chillers as proposed by the vendor adjusted for associated installation factors.

UCj = plant utility costs for year j. If there is more than one utility type (e.g. electricity, gas, water), this component would be duplicated for each.

MCj = relative maintenance costs for year j.

d = discount rate, also called the "cost of capital" or the "minimum rate of return." This rate is used to discount future cash flows, converting them to present costs. The higher the rate, the less future energy savings will help offset the first cost penalty of a more expensive chiller plant. The discount rate for a typical business owner might be 8% to 15%, reflecting the cost of borrowing money plus a few points to reflect the investment risk. A more progressive owner or government entity may be less conservative and use rates near 5%.

N = Number of years of analysis, or life cycle. While called the life cycle, N is seldom equal to the actual number of years the chiller plant will operate. Most studies only look at the first 10 to 15 years since there are so many uncertainties in the utility costs and overall energy usage the further into the future one looks. Also, most businesses will expect a return on investment in well less than 10 years.

The life cycle cost equation above does not include tax impacts, depreciation, investment tax credits, financing costs or salvage value. The reader is referred to more detailed texts on life cycle costing if these factors are considered significant enough to take into account.

If energy and maintenance costs are constant, this equation can be simplified to:

Where,

$$LCC = FC + PWF_e * UC + PWF_m * MC$$

Equation 7-2

Equation 7-2 (cont'd)

$$PWF_e = \frac{(1+e')^N - 1}{e'(1+e')^N}$$

$$e' = \frac{(d-e)}{(1+e)}$$

$$PWF_m = \frac{(1+m')^N - 1}{m'(1+m')^N}$$

$$m' = \frac{(d-m)}{(1+m)}$$

UC = Utility costs at current rates.

MC = Maintenance costs at current prices.

e = escalation (inflation) rate for electricity (or whatever fuel type is being used) above current rates. Estimating escalation rates is very difficult in this transitional market between regulated and unregulated utilities. See discussion under utility rates above.

m = maintenance cost escalation (inflation) rate. This can typically be assumed to be equal to the consumer price index (CPI), although, like energy escalation rate, this is difficult to predict with any certainty. A 1% escalation is usually reasonable.

Procurement Step #8: Final Chiller Selection

Evaluating chiller options using the procedures recommended in this Design Guide requires making many assumptions and simplifications. To pick the "best" chiller plant option, it is important to test the sensitivity of various assumptions. For instance:

- If there is uncertainty about the loads the plant will need to handle, develop energy costs under various load profiles. For example, a plant may be expected to ultimately handle a large load, but it is possible that the actual load may be smaller. Be sure that the plant can efficiently operate at low loads and that the plant that proved to be optimum under high-load conditions also was near optimum under low-load conditions.
- Utility rates may change dramatically when utility markets are fully deregulated. Experiment with rates that include very high on-peak energy charges and demand charges. For mixed fuel plants, experiment with escalating rates for one utility and deescalating rates for the other.
- Life-cycle cost assumptions, such as discount rate and years of analysis, affect how much future energy savings are weighted. Experiment with various assumptions to see how they affect the results.

More than likely, after calculating results over a range of assumptions, the life-cycle costs of several chiller options will be close to the "optimum" option (the one with the lowest life-cycle cost). It is also possible that the optimum choice will vary depending on the assumptions made. In this case, the final selection must also consider "soft" factors (those to which a dollar amount cannot be easily attached) to break the tie. The final selection should be made by the entire design and ownership team, not just by the engineer and contractor. This will ensure that all "soft" issues have been considered and everyone had a fair chance to express any vendor preferences.

These "soft" factors include:

Reliability and reputation of the manufacturer and local representation

Most facility owners and operators will have a favorite chiller vendor based on past experience. This can be a very important tie breaker when making the final selection. If the owner's favorite vendor offers an option that has close to the optimum life-cycle cost, it may be a politically sensible idea to choose that option.

Refrigerant type

Even if a range of refrigerant options is allowed in the bid specifications, the owner or engineer may have a preference for a given refrigerant type based on the refrigerant's impact on the ozone layer and global warming, or impending production phase-out dates.

Redundancy

Typically, an analysis of chiller options assumes that the chiller plant operates normally. But what happens if a chiller fails? Different chiller options may accommodate outages better than others. Options offering the most chillers will generally offer the least exposure to chiller failure. But other more complex failure considerations should be considered. For example, if a plant with unequally sized chillers loses the large chilled water pump, will the flow ranges allow the large chiller to stay on-line using the small chilled water pump?

Case Studies

New Building

In the case study introduced in Chapter 6, chiller optimization techniques for a high-rise office building were presented. This chapter builds on that case study to demonstrate the chiller selection process. You may find it useful to review the case study in Chapter 6 before continuing with this section.

The sample building is 15 stories high, enclosing 540,000 ft^2 in San Francisco, California. The building primarily contains offices, with some assembly and retail space on the ground floor, a 5,000-ft^2 data center, and a large cafeteria. The data center and various small server rooms operate continuously and place roughly a 50-ton base load on the plant. Total plant load was calculated to be 1,100 tons.

After the primary design temperatures and flow rates were determined (see case study in chapter 6), the performance specification and bid forms were developed (see sample specification and forms). Four chiller vendors were invited to bid. The data from the bid forms were entered into the CoolTools Chilled Water Plant Analysis Program, which then developed the regression coefficients of each chiller in each option. These coefficients were then inserted into the DOE-2 model of the building. Energy savings were calculated using various control scenarios such as condenser water setpoint reset and staging points. The lowest energy costs for each option were selected and life-cycle costs were calculated as shown in Table 7-2.

Table 7-2: Life-Cycle Cost (LCC) Summary

Option	Description	Cost w/ Contr. Markup ($)	Other first cost add/ deduct ($)	Added Exhaust Fan ($)	Total Cost ($)	1st Cost Rank	Total Building Energy Costs ($)	Energy Cost Rank	Life-cycle Cost ($)	LCC Premium vs. Base ($)	LCC Rank
1	400 ton, 0.50 kW/ton; 700 ton, 0.55 kW/ton	268,235	6,000	0	274,235	6	712,293	9	6,371,092	142,016	10
2	400 ton w/VFD, 0.50 kW/ton; 700 ton, 0.55 kW/ton	301,235	6,000	0	307,235	9	694,427	2	6,251,168	22,092	4
3	365 ton, 0.56 kW/ton; 735 ton, 0.50 kW/ton	199,980	3,000	0	202,980	1	724,350	12	6,403,038	173,962	12
4	365 ton w/VFD, 0.56 kW/ton; 735 ton, 0.50 kW/ton	240,130	0	0	240,130	3	702,168	5	6,250,322	21,246	3
5	365 ton w/VFD, 0.56 kW/ton; 735 ton w/VFD, 0.50 kW/ton	289,190	0	0	289,190	8	694,854	4	6,236,778	7,702	2
6	200 ton, 0.50 kW/ton; 900 ton dual 0.54 kW/ton	212,031	0	0	212,031	2	712,100	8	6,307,235	78,159	5
7	550 ton dual, 0.56 kW/ton; 550 ton dual, 0.56 kW/ton	256,485	0	0	256,485	5	714,269	11	6,370,255	141,179	9
8	400 ton dual, 0.53 kW/ton; 700 ton dual 0.53 kW/ton	254,533	0	0	254,533	4	711,137	7	6,341,495	112,419	8
9	200 ton, 0.53 kW/ton; 350 ton dual 0.57 kW/ton; 550 ton dual 0.59 kW/ton	252,485	25,000	0	277,485	7	712,547	10	6,376,516	147,440	11
10	550 ton w/VFD , 0.49 kW/ton; 550 ton, 0.48 kW/ton	307,106	6,000	3,000	316,106	11	702,969	6	6,333,154	104,078	7
11	300 ton w/VFD , 0.50 kW/ton; 800 ton, 0.48 kW/ton	305,151	6,000	3,000	314,151	10	691,038	1	6,229,076	0	1
12	365 ton w/VFD , 0.52 kW/ton; 366 ton, 0.51 kW/ton 366 ton, 0.51 kW/ton	338,320	33,000	3,000	374,320	12	694,806	3	6,321,497	92,421	6

Note that first-cost adjustments were made for additional piping and pumping system costs caused by the use of three-pass evaporators in some cases and multiple chillers in others. Additional exhaust fan capacity was added for open-drive machines due to the higher heat load they generate in the chiller room. (In more severe climates, this high heat load may require that mechanical cooling be added to the room.)

The total building energy cost column was taken from the DOE-2 model. Life-cycle costs were calculated based on a 15-year life, eight percent discount rate, and zero percent energy escalation rate.

Option 11 has the lowest life-cycle cost, but Option 5 is very close and even Options 2 and 4 have life-cycle costs close enough to Option 11's that they need to be considered. Ultimately, Option 5 was selected for the project based on the following "soft" considerations:

- It uses R-134a whereas Option 11 uses R-123. R-123 was considered acceptable, but R-134a was preferred by the Owner because it has zero ozone depletion potential and is not scheduled to be phased of production.
- Both chillers in Option 5 have variable-frequency drives, whereas only the small chiller in Option 11 has a VFD. Having two VFD chillers improves redundancy since either chiller should be able to handle the low nighttime data center loads.

- Both chillers in Option 5 have design chilled and condenser water flow rates and minimum flow rates that overlap so that the small chilled water pump is large enough to keep the big chiller on-line. That is not the case with Option 11. In case of any pump failure, the large chiller can remain on-line with Option 5 whereas if one of the large pumps fails, only the small chiller can remain on-line with Option 11.
- Option 5 uses hermetic motors whereas Option 11 uses an open drive. Both have advantages and disadvantages, but the owner's engineer preferred hermetic motors based on past seal problems experienced with open-drive machines. The hermetic motors also made the chiller room have a net neutral load so no conditioning was required.
- Option 5 is less expensive by about $15,000, making it easier to reach first-cost budgets.

After the bid, chiller vendors were given a modified version of Table 7-2 that included only the columns for first cost rank, energy cost rank, life-cycle cost savings vs. base, and life-cycle cost rank. This allowed the vendors to see how their proposals compared to their competitors without seeing actual pricing.

8. START-UP AND COMMISSIONING

Commissioning Basics

Introduction

An excellent chilled water plant design does not guarantee that the system will achieve optimal performance. Many things can happen—especially during construction and equipment installation—that may have a negative effect on the plant's operation. A well-designed and properly implemented commissioning plan can help ensure that the system will meet the design intent.

This chapter provides an overview of the commissioning process, including:

- What commissioning is
- Why commissioning is important
- Benefits and costs of commissioning
- Levels of commissioning
- Who should act as the commissioning authority

This overview is followed by a discussion of the various phases of commissioning, from the development of the commissioning program to post-occupancy commissioning activities. Examples of a commissioning plan, a commissioning specification and test procedures are also included (see Appendices C, D, and E).

For further information on the commissioning process, refer to the CoolTools™ document, *Project Implementation Plan: Achieving Successful Chilled Water Plants; and ASHRAE Guideline 1-1996, The HVAC Commissioning Process*.

What is Commissioning?

ASHRAE defines commissioning as "a systematic process of ensuring that systems are designed, installed, functionally tested, and capable of being operated and maintained to perform in conformity with the design intent."

Commissioning is intended to achieve the following objectives:

- Ensure that equipment and systems are properly installed and receive adequate operational checkout by the installing contractors.
- Verify and document proper operation and performance of equipment and systems.
- Ensure that the design intent for the project is met.
- Ensure that the project is thoroughly documented.
- Ensure that the facility operating staff is adequately trained.

Procurement processes for complex building systems such as chilled water plants typically have gaps that may lead to less than optimal system performance. For example, although contractors are contractually obligated to install equipment according to the construction documents, they are not required to meet the project's design intent. An installation that follows the construction documents to the letter does not necessarily ensure that the design intent will be met or that the systems will function properly. Commissioning is meant to address these gaps. It can be considered a quality control process that adds value by helping to meet the project's design intent and achieve top performance of a chilled water plant.

Commissioning involves more than start-up, construction observation, or testing, adjusting and balancing (TAB). These are important elements of the overall commissioning process, but these procedures alone do not constitute commissioning.

Start-up. Start-up is a one-time, primarily "static" activity focused on making sure that equipment is installed correctly and that all necessary safety features and controls are working. Start-up is mostly limited to the equipment level, with little or no work at the system level. Commissioning, on the other hand, focuses on the system level and on "dynamic" processes that occur during start-up, shutdown, full and part load, and alarm conditions.

Construction Observation. Construction observation verifies that the quantity, size and capacity of installed equipment comply with specifications, and that the quality of workmanship is satisfactory. Even more so than with start-up, this work is limited to "static" activity, with few or no operational checks. As part of the commissioning process, construction observation can help uncover problems with installation and help reach solutions that are less costly to correct at this stage than after occupancy.

Testing, Adjusting, and Balancing. Testing, adjusting and balancing (TAB) work is generally limited to setting flows, pressures and temperatures to design values at minimum and maximum conditions. Much of the time, however, chilled water plant equipment operates at part-load conditions, which are generally not addressed by TAB work. Except for testing static pressure, TAB usually involves neither optimizing operating and performance conditions nor checking sequences of equipment and systems operation. TAB work must be complete before performing the "dynamic" verification tests that are a major aspect of commissioning. The commissioning authority will review TAB results to ensure that the work has been completed properly.

Commissioning Activities. Commissioning activities, which occur in addition to the normal project procurement activities, include:

- design review,
- additional submittal review,
- construction observations,
- pre-verification and verification test procedures (including checking part-load operation and sequences of operation),
- training coordination, and
- detailed project documentation.

The Commissioning Authority. The commissioning authority coordinates all commissioning activities. One very important function of the commissioning authority is to facilitate communication among the project team members.

Why Is Commissioning Important?

A chilled water plant represents a significant investment for the building owner, especially when lifetime operating and maintenance costs are considered. An excellent design has the potential to reduce these lifetime costs. When integrated with the design and construction processes, commissioning can help achieve these expected savings.

Commissioning is important for many reasons:

- *Buildings/systems not meeting owners' expectations*. Too often, building owners settle for systems that work but have problems or have not been set up to run optimally. There may be higher than expected energy costs, frequent occupant complaints, and an operating staff frustrated with equipment or systems that don't work properly.
- *Systems—particularly control systems—are complex*. Today, more than ever, the equipment and systems that make up a chilled water plant can be very complex. In particular, the integration of modern DDC control systems with all of the chilled water plant equipment tends to be a source of trouble.
- *Tight project budgets*. Project budgets are almost always tight in construction projects today. Design fees typically are not large enough to cover a thorough analysis of system options or to provide an appropriate level of quality control during construction. Contractors are generally chosen for their low bid. Value engineering almost always forces cuts to be made somewhere in the project. It is often the mechanical systems that are affected.
- *Projects are often on a fast track*. In addition to having tight budgets, owners are often in a hurry to occupy the building or to start using a new chilled water plant. Contractors often face significant financial penalties for not meeting deadlines. Tight scheduling pushes the contractors to work faster and may affect installation quality. Some contractors do not always have time to check the quality of their work.
- *Detailed checkout/testing not part of normal construction process*. Many owners resist commissioning because they assume that this type of service is already part of normal construction practices. However, due to many factors, including any one or all of the previously mentioned items, thorough equipment and system testing is not part of typical construction projects. It is interesting to note that standard American Institute of Architects (AIA) contract language, which is commonly used for contractual arrangements between parties, specifically does not include "…exhaustive or continuous on-site inspections to check the quality or quantity of work." (*Section 2.6.2.1, AIA Document B141, Standard Form Services, 1997 Edition, The American Institute of Architects, Washington, D.C.*)

Benefits of Commissioning

There are many benefits to commissioning, including commissioning in the project procurement process. Although the building owner realizes the majority of these benefits, other parties in the project can benefit too. In addition to the specific benefits listed below, commissioning usually gives all parties more confidence that the building systems will function as intended when the owner accepts the project.

Benefits to the owner include:

- Fewer system deficiencies at project turnover
- Fewer contractor callbacks
- Fewer occupant complaints
- Increased employee comfort which could boost productivity
- Systems that meet the design intent and operate optimally
- Well-trained operating staff
- Improved energy efficiency
- Thorough project documentation

Benefits to the design professional include:

- Help with refining design concepts
- Reduced post-construction trouble calls
- Fewer premature design error accusations
- Increased knowledge that benefits future designs

Benefits to the contractor include:

- Reduced callbacks
- Earlier resolution of problems
- Increased emphasis on quality control
- Help meeting scheduled completion
- Smoother installation

Costs

At first glance, it might appear as if the commissioning process increases a project's cost. However, when all benefits are considered, commissioning actually has the potential to save money. With tangible benefits alone (for example, systems working properly from the beginning, project schedules met, improved energy efficiency, etc.), there is potential to reduce project costs. When less tangible benefits are included (for example, fewer occupant complaints, increased productivity, well-trained operating staff), the potential for decreased project costs (or put differently, for better value) becomes even more likely. The up-front cost of commissioning services should be considered money well spent.

There is an economy of scale associated with the cost of commissioning, as there is with design services in general. The basic tasks for the commissioning process are the same in a 300-ton chilled water plant as in a 3,000-ton plant. Obviously, the cost of commissioning as a percentage of the total project cost will be higher in a smaller plant than they will be in a larger plant.

As discussed in the next section, different levels of commissioning are appropriate for different projects. The relative costs of these levels are shown in Tables 1 and 2 below (with level 1 representing the least work and level 3 representing the most work). These costs are the typical fee for the Commissioning Authority. Other contractors, in particular the controls contractor, may also have additional costs, but these are usually fairly insignificant. Depending on the owner's needs, the level of commissioning chosen could increase with plant size. Especially in larger, more complex plants, the question should be asked: Can a project afford not to include commissioning?

Table 8-1: Commissioning Cost as Percentage of Total Project Cost

Level	300-Ton Plant	3,000-Ton Plant
1	1% – 2 %	½% - 1%
2	3% - 5%	1% - 2%
3	4% - 10%	2% - 5%

Table 8-2: Commissioning Cost in Dollars

Level	300-Ton Plant	3,000-Ton Plant
1	\$3,000 - \$6,000	\$15,000 - \$30,000
2	\$9,000 - \$15,000	\$30,000 - \$60,000
3	\$12,000 - \$30,000	\$60,000 - \$150,000

Notes relative to both tables: 1) Assumes total installed cost of \$1,000/ton;
2) Figures are for chiller plant only, not "building commissioning,"
i.e., does not include load-side, electrical or other work.

Levels of Commissioning

The level of commissioning effort should be dictated by the building size, the complexity of the systems and associated controls, and by how much the owner is willing to spend. The levels of commissioning are related to the intensity of the work, but also are related to when the commissioning process begins, as shown in Figure 8-1.

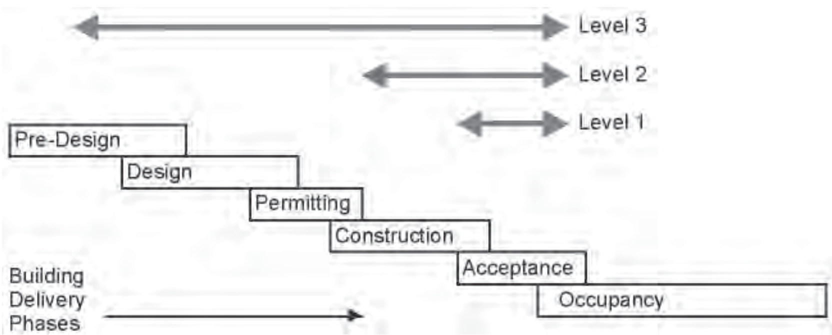


Figure 8-1: Levels of Commissioning

Level 1

The work included in a level 1 project might appropriately be called "enhanced start-up services." This work would not begin until at, or near, the end of construction. Level 1 commissioning, a less rigorous process than level 2 or 3 commissioning, might use standard forms and checklists to document the process.

Typical tasks in level 1 commissioning include:

- Walk through site to verify equipment installation
- Conduct simple test procedures on selected (not all) equipment
- Issue of a report on findings

The commissioning authority is not as actively involved with the project team as in level 2 or level 3 commissioning. Level 1 commissioning has the lowest cost, but also leaves out much of the important detailed testing.

Level 2

Here, the commissioning process starts during the construction phase. There are two scenarios for when the work actually begins—either at the start of construction (the preferred approach) or at the end of construction just before the owner accepts the project. In both scenarios, the amount of detail included in testing procedures is similar. Important aspects that would be missed by starting commissioning at the end of the construction phase include active involvement with the project team and the potential to find and correct problems during equipment installation. Level 2 commissioning will cost more than level 1, but the benefits will generally outweigh the additional costs.

The *verification testing*[1] in level 2 commissioning is much more rigorous and comprehensive than in level 1. Testing involves all installed equipment and systems, as well as interactions between systems. The commissioning authority will develop customized test procedures specific to the actual installed equipment.

Typical tasks in level 2 commissioning include:

- Review equipment submittals
- Hold periodic *commissioning team*[2] meetings
- Visit site periodically
- Develop commissioning plan
- Develop and oversee execution of *pre-verification*[3] and verification test procedures
- Coordinate operation and maintenance (O&M) training
- Review O&M manuals and incorporate into systems manual
- Prepare and issue commissioning report

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1. Series of test procedures to verify that equipment/systems are operating in accordance with the design intent and contract documents.
2. The commissioning team is essentially the same as the project team. It includes the commissioning authority, owner, design professional, general contractor/construction manager, mechanical contractor, electrical contractor, controls contractor, TAB contractor, manufacturer's representatives, and others as necessary.
3. Primarily "static" checks to verify that equipment has been installed properly and is ready for further testing. May be done in conjunction with normal start-up procedures.

Level 3

Level 3, the most comprehensive level of commissioning, begins during the program phase or at the start of the design phase. The work here includes all of level 2, and adds tasks such as design review, *functional performance tests*[4], and review of *O&M* manuals. In this scenario, the commissioning authority is involved during the entire project. Level 3 commissioning has the highest costs but can also have the most significant benefits.

Typical tasks in level 3 commissioning include:

- Help select design team (if requested by owner)
- Help select desired system configuration (if requested by owner)
- Review design
- Develop commissioning specification
- Develop commissioning plan
- Review equipment submittal
- Hold periodic commissioning team meetings
- Visit site periodically
- Develop and oversee execution of pre-verification, verification, and functional performance test procedures
- Review O&M manuals and incorporate into the systems manual
- Coordinate O&M training
- Prepare and issue commissioning report

4. Test procedures to determine as-installed capacity and performance.

Ison's Rule of 10

When considering commissioning costs, it pays to keep in mind Ison's Rule of 10. This simple rule states that in any designed process, the earlier a problem is uncovered, the less costly the solution. The following table illustrates Ison's Rule using an example of the relative cost associated with correcting a problem found at various points in a typical construction project.

Illustration of Ison's Rule of 10

Problem Uncovered	*Relative Cost*
During preliminary design	*$3*
During final design	*$30*
During construction	*$300*
Post-occupancy	*$3000*

To illustrate how Ison's Rule might apply to a chilled water plant, consider the relative costs of finding and correcting an undersized plant during the various design stages:

- During preliminary design, re-doing load calculations might cost $500.
- During final design, re-doing load calculations and design drawings might cost $5,000.
- During construction, re-doing load calculations and drawings, plus change orders for demolition and purchasing new equipment might cost $50,000.
- After occupancy, complete re-design, demolition and replacement might cost $500,000 (or the owner may just live with an undersized plant and the associated problems).

This simple example shows that by including commissioning as early in a project as possible, the chances of catching mistakes are increased while the cost of remedying those mistakes is decreased.

Who Should Function as Commissioning Authority

The commissioning authority coordinates commissioning activities and provides project oversight. An important function of the commissioning authority is to facilitate communication among project team members, which can help ensure a smooth installation. The commissioning authority needs to have a wide range of skills to be effective in the commissioning process.

Some of the qualifications needed by the commissioning authority include:

- In-depth knowledge of building systems performance and interactions
- Familiarity with equipment installation, start-up, troubleshooting, and operation and maintenance procedures
- Familiarity with testing, adjusting and balancing procedures
- Good oral and written communication skills
- Familiarity with the building design and construction process

Several parties have the potential to function as the commissioning authority, including:

- Owner
- Independent third party
- Design professional
- General contractor
- Mechanical contractor
- TAB contractor

Each of these parties would have a different answer to the question: Who should be the commissioning authority? The following discussion points out some of the pros and cons of each option. The final choice may depend on the owner's needs and the complexity of the construction project. An independent third party or design professional may be more appropriate for larger projects with more complex systems, while contractors may be appropriate for more simple projects, e.g., those with packaged chilled water plants.

Owner

If the owner has a fairly sophisticated facilities staff, it is possible for them to perform the commissioning process. However, many owners do not have a staff capable of performing the commissioning activities or believe that it is more cost effective to use "outside" services. Owners more likely to perform commissioning themselves are those that are constantly in the process of constructing new buildings. Such owners include colleges and universities, banks, and other growth organizations with multiple branch offices.

Independent Third Party

Many owners choose to have an independent third party (someone not part of the normal project team) serve as the commissioning authority on their projects. An independent party may be better able to maintain an objective point of view on the project. They provide another set of eyes, helping to improve project quality and ensure that the owner's goals are met satisfactorily. An independent commissioning authority will typically have a diverse range of experience with many types of projects. A drawback to the independent party route includes having an additional contract to manage, which can increase administrative costs.

Design Professional

Some people feel that the design professional is the right party to serve as the commissioning authority. An advantage to using design professionals is that they are familiar with the project's design intent. Another advantage is that they are already under contract for their normal services. However, being expert at design does not necessarily translate into being proficient at commissioning. Many, if not most, design professionals focus on design work and do not have hands-on experience with start-up, operation and troubleshooting, which are essential skills for commissioning. There is also an issue of conflict of interest when the design professional serves as the commissioning authority on his or her own project (e.g., a design professional commissioning his own project would not be inclined to acknowledge a design error discovered during commissioning).

General Contractor

General contractors may be in a position to serve as the commissioning authority. They generally have good experience with the overall construction process and project scheduling. They should be fairly familiar with the project's layout. However, they typically don't have in-depth knowledge of mechanical system operation, testing and troubleshooting, and they may not be well versed in the design intent. The conflict of interest issue also applies to general contractors (e.g., the contractor's need to stay on schedule may be in conflict with the need for proper commissioning).

Mechanical Contractor

Mechanical contractors might seem like a logical choice to serve as the commissioning authority. They are familiar with the installation. They should be good at scheduling. They may also have the capability to perform some level of tests. However, they are generally not familiar with the design intent, and they may lack other skills necessary for the commissioning authority. Here too the conflict of interest issue applies when the mechanical contractor serves as the commissioning authority.

TAB Contractor

TAB firms could be considered for the role of commissioning authority, particularly since TAB work is a significant part of the overall commissioning process. Quality TAB firms are skilled at testing and are knowledgeable about a variety of mechanical systems. However, they are generally not familiar with the design intent; other aspects of the commissioning process also often exceed the skills of TAB firms. The conflict of interest issue applies here too. The question must be asked: Can the TAB firm objectively assess its own work?

Table 8-3: Comparison of Various Parties' Abilities to Act as Commissioning (Cx) Authority

Activity	Owner	Independent Party	Design Professional	General Contractor	Mechanical Contractor	TAB Contractor
Design Review	2	3	1	0	1	0
Cx spec. development	2	3	2	0	0	0
Develop Cx plan	2	3	2	1	1	1
Develop design intent	2	2	3	0	0	0
Submittal review	2	3	1	1	0	1
Develop Verification tests	2	3	2	1	1	1
Develop Funct. Perf. tests	2	3	2	1	1	1
Coordination & oversight	2	3	2	2	1	1
Construction observations	2	3	2	1	0	1
Direct testing procedures	2	3	2	1	1	1
Assemble systems manual	2	3	2	1	1	0
Prepare Cx report	2	3	2	1	1	1

0 – not recommended 1 – should be considered 2 – recommended 3 – strongly recommended

This section discusses the tasks involved in each phase of the commissioning process for chilled water plants. It also provides guidance on how to develop some of the important materials needed, such as the preliminary commissioning plan and the commissioning specification. The recommendations presented are geared toward a level 3 commissioning process. A level 1 or level 2 commissioning process would call for fewer tasks than are described below. The tasks and materials that are required for level 3 depend on whether the commissioning authority begins work at the start of the program phase or the design phase.

Program Phase (Level 3)

If the owner plans to perform the commissioning work in-house or elects to have a commissioning authority begin their involvement in the program phase, the following tasks may be undertaken:

- Develop the owner's program
- Help select the project team
- Develop the preliminary commissioning plan

Develop the Owner's Program

The commissioning authority (CA) may assist the owner in developing the basic project requirements. This might include a description of goals, objectives and system performance criteria, as well as cost and other limitations. The commissioning authority would probably not help specify the space requirements, but rather help set performance requirements for envelope, mechanical and electrical systems. The scope of the commissioning process should be defined and the project's organizational structure should be described.

Help Select the Project Team

The commissioning authority can assist in reviewing the qualifications of potential project team members and selecting firms to interview. After the interviews, the commissioning authority may help select the appropriate firm or firms for the project. The commissioning authority is not likely to assist clients who regularly construct buildings, but the CA's assistance might be very welcome and helpful for owners who are going through the process on a "one time" basis.

Develop the Preliminary Commissioning Plan

The preliminary commissioning plan, which is developed during the program phase, provides an overview of the commissioning process. It outlines the anticipated scope of work, time required for completion, commissioning team members and their responsibilities, anticipated test requirements, and operator training requirements. As the project progresses, this preliminary plan is expanded and updated, to become the final commissioning plan that defines the commissioning requirements at every stage of the project. The sooner the commissioning plan is developed the better, since it will help team members understand their roles and responsibilities.

Design Phase (Level 3)

The commissioning authority's involvement in the design phase helps to ensure that the owner receives an adequate system design. Work during this phase includes:

- Helping to select the desired system configuration
- Reviewing the design
- Developing the commissioning specification

Help Select Desired System Configuration

Depending on the project and owner requirements, it may be appropriate for the commissioning authority to set criteria for the desired system configuration.

Design Review

It is valuable to have an experienced engineer on the commissioning team to provide an independent design review. A peer review of the design offers many benefits to the owner and the designer. Most importantly, it provides oversight; a second pair of eyes can help prevent expensive design errors. Design review requires thorough documentation of all design assumptions and calculations. This helps uncover inconsistencies, errors and omissions. The design review helps improve quality control by promoting conformance with the design intent.

Design review may be either conducted or simply overseen by the commissioning authority. Design review may be a one-time activity occurring when the design is complete, or it may occur at multiple points in the design process (schematic design, design development, construction drawings). The design reviewer should be independent and work directly for the owner. Potential reviewers include one or more of the following: the owner's in-house engineering staff, the commissioning authority, a mechanical contractor, or an independent mechanical engineering consultant.

The reviewer's goal is to support the design team in developing a system design that meets the design intent.

The person performing design review would check the following:

- Is the statement of design intent complete and consistent with the requirements of the owner's program?
- Does the load profile calculation include all critical factors?
- Is the control strategy described completely, and does it meet the design intent?
- Is the system layout shown completely and consistently in the drawings? Is there a schematic diagram? Will the system as shown meet the design intent?
- Is appropriate equipment called out in the specification, as well as in the installation requirements?
- Do the drawings and specification adequately describe the instrumentation required for control and performance monitoring?

- Are *acceptance-testing*[5] requirements defined properly?
- Does the commissioning specification adequately describe the process, as well as the roles and responsibilities of the parties?
- Has the economic analysis been updated to match the current design?

The following information should be provided to the reviewer:

- Load calculations for the facility, with sufficient detail to aid in understanding the factors affecting the load
- Equipment selection criteria and equipment specifications
- Detailed drawings, including schematic diagrams, plan drawings, control diagrams, piping and instrumentation diagrams, equipment schedules, and construction details
- Description of the control strategy, control specification, ladder logic diagram, and energy management system (EMS) interface specifications
- Commissioning plan and specification

To be effective, the design review process requires open lines of communication and cooperation between the reviewer and design team. The process should begin early in the design phase. Typically a design review will occur near the end of the design development (DD) phase, but before the construction documents (CD) are prepared. Input from the reviewer to the design team should occur throughout the design process. This allows any needed modifications to be incorporated with minimal disruption to the schedule.

Commissioning Specification

The commissioning specification defines in detail the activities that will occur during the construction and acceptance phases of a project to fulfill the commissioning plan. The commissioning specification is a part of the contract documents, and is an important element of the project implementation. The specification defines contractual relationships for carrying out commissioning. It also describes the roles and responsibilities of each party in the commissioning process. It must also specify requirements for construction observation, start-up, and acceptance testing, and define how the performance of the system will be evaluated.

The commissioning specification must describe the commissioning requirements in enough detail so that contractors can competently submit bids to perform the work. The specification, with a section in *Division 1*[6] and in *Division 15*[7], should include the following information:

- Lines of communication and reporting requirements
- Roles and responsibilities of all parties

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5. Testing that occurs during the acceptance phase including pre-verification, verification, and functional test procedures.
6. The divisions refer to the standard way of organizing construction specifications, as established by the Construction Specifications Institute (CSI). Division 1 is the general section that lays out procedures and methods for all building systems.
7. The divisions refer to the standard way of organizing construction specifications, as established by the Construction Specifications Institute (CSI). Division 15 is the section that deals with mechanical systems.

- Requirements for team meetings
- Identification of the equipment and systems to be commissioned
- Criteria for acceptance of the system, including how failure is defined
- Recourse if required performance is not achieved
- Project documentation requirements, including control submittals, equipment submittals, test results, as-built documentation, and system manual
- Description of test methods to be used

Construction Phase

Construction-phase activities for the commissioning authority are focused on preparation for the acceptance-phase testing procedures. The goal during the construction phase is to complete the installation efficiently, with minimal problems to hamper acceptance-phase activities. Commissioning tasks during construction include:

- Submittal review
- Construction observations
- Commission meetings
- Test procedures development

Submittal Review (Level 2, 3)

Construction specifications often require that contractors or subcontractors submit detailed drawings or product samples for approval by the design professional. "Submittal Review" is the process of evaluating this information for compliance with the requirements in the specifications. If a complete and thorough design has been prepared, with material and equipment requirements clearly defined, submittal review should be limited to verifying that the proposed equipment meets the stated requirements. However, pressure to cut costs, or unfamiliarity with the type of system (for example, cool storage), can cause contractors to propose equipment that does not meet the design intent.

Control system submittals are of particular concern. In terms of overall system operations, the control system is probably the most critical part of a chilled water plant. Control system submittals must be carefully reviewed to ensure that the system will meet the design intent. Proposed control sequences must be consistent with the control sequence description provided in the project specification.

The submittal review verifies all the details of equipment performance and configuration that are necessary for proper system functioning. *The review process focuses on the products and shop drawings proposed for the project, including*:

- Verifying that submittals are complete
- Confirming that specified performance at full and part load is documented
- Confirming that pressure drops of chillers, heat exchangers, and accessories are within specified ranges

- Ensuring that instrumentation is selected for appropriate measurement ranges and accuracy
- Verifying that pump head and flows are as specified

Construction Observations (Level 2 and 3)

Periodic site observations during construction are an important way to keep tabs on the installation's progress. These site observations may uncover problems that can be corrected easily. For example, it is a lot easier and less costly to change piping mistakes while the pipes are empty than when they are full of water.

The number of site visits required generally depends on the size and complexity of the project. As a rule of thumb, *site visits should be conducted during*:

- piping installation,
- major equipment placement,
- pressure and leakage testing, and
- periodically during general construction.

Some projects may warrant site visits as often as every week during construction.

Construction observations typically include:

- Verification that equipment is installed in specified locations, with appropriate workmanship
- Observation of procedures such as pressure and leakage testing, and flushing and cleaning

Assessment of the energy efficiency impacts of the installation.

> ***Examples of Problems Addressed During Field Observations***
>
> In one project, a magnetic flow meter was found during a site visit to have been piped backwards with regards to upstream and downstream piping. Fortunately, the system had not been filled, and it was easy to cut out the piping and correct the mistake. The contractor had read the plans backwards.
>
> In another project, a control valve and a manual isolation valve (both butterfly valves) on a bypass line were placed too close together. When the control valve was operated, it got hung up (stuck) on the isolation valve. In this case, the system was already full, and it was much more difficult to implement a solution. Also, in this case, the piping had been completed before the commissioning authority began work on the project. Construction observations probably would have picked this up.

Commissioning Meetings (Level 2, 3)

Periodic commissioning team meetings during construction are useful for scheduling activities, discussing and resolving problems, and tracking solutions. A commissioning kickoff meeting is usually held at the beginning of the commissioning process to explain to the project team what will occur during the process and to ensure that everyone understands their roles and responsibilities. The commissioning authority should periodically attend regular project team meetings during construction that include commissioning on the agenda and may call special commissioning team meetings if needed. During acceptance testing, it is advisable to hold team meetings to discuss issues that arise as a result of testing. Minutes should be taken at each meeting and distributed to team members in a timely manner. The meetings should be attended by the owner, general contractor/construction manager, and by the lead person from each major trade.

Develop Test Procedures (Level 2, 3)

This is a very important part of the commissioning process. Careful planning and preparation are critical to ensuring that the test procedures will yield reliable and useful test results. The time spent on developing thorough test procedures will help lead to effective application of the tests.

Things to consider when developing test procedures include:

- What are the overall goals of test?
- What results are expected in each test? What is the best way to accomplish these results? Does the test focus on the equipment level or system level?
- What is supposed to happen, and when?
- Are all of the control sequences documented? For example, for a chiller to start, there must be flow of both chilled water and condenser water to the chiller. The sequence might be that first the condenser water pump starts and proves flow, then the chilled water pump starts and proves flow, and then the chiller is enabled to start. This would be documented in the test procedures.

Exactly what data and/or measurements are needed to accomplish the test? Also, how will this information be obtained? DDC control systems can log control point data over specified time periods. Can the control system supply all necessary data for a given test? If not, are stand-alone data loggers needed to provide additional data? Will manual measurements be needed?

- Depending on the circumstances, the amount of data and/or measurements can be quite large. Typical data needed for various tests in a chilled water plant project include:
    - Supply and return, chilled water and condenser water temperatures
    - Chilled and condenser water flows
    - Pressure differences across pumps, chiller barrels and coils
    - Outside air temperature and humidity
    - Control valve positions at certain times
    - Motor speeds, especially those with variable frequency drives
    - Motor voltage and amperage (and kW if possible) for chillers, pumps, towers, peripherals at various loads

- How can various loading conditions be achieved?

  Part-load conditions are usually easy to obtain; chilled water plants typically operate at part load over much of a year. Achieving full-load conditions sometimes presents a challenge, however. If the testing can be done on a peak summer day, there is chance full load conditions will exist. However, testing is rarely timed to coincide with favorable weather conditions. Some form of false loading is generally required to simulate full-load conditions. How can this false loading be accomplished? Solutions may include heating the building overnight or allowing the chilled water loop temperature to get high enough to fully load the chiller for at least a short period.

- Are test procedures repeatable?

  This is important in case a test fails and must be repeated later, after corrections to the system are made. This is also important for ongoing commissioning activities; it will enable the test to be duplicated periodically over the chilled water plant's life.

Acceptance Phase

The acceptance phase is where, so to speak, the rubber hits the road with the commissioning process. As mentioned previously, the focus of all preceding commissioning activities is to enable the acceptance tests to occur with minimal problems. Commissioning responsibilities during the acceptance phase include:

- Start-up and pre-verification tests
- Verification of control device function
- TAB issues
- Verification tests
- Functional performance tests
- Coordination of operator training
- Systems manual development

Start-up and Pre-Verification Tests (Level 1, 2, 3)

Once installation is complete, all equipment, including chillers, pumps and cooling towers, must be started up following the manufacturer's recommendations. This is typically the responsibility of the mechanical contractor; often a manufacturer's representative assists with the process.

Too often, the start-up process is not properly documented, either because no documentation materials are available, or because the contractor neglects to fill them out. The commissioning authority should ensure that start-up documentation materials are available, that they get filled out, turned over to the Commissioning Authority, and included in project documentation.

If deemed necessary, in addition to start-up documentation, pre-verification tests may be developed by the commissioning authority to enhance the start-up procedures and gather additional information. These pre-verification tests, primarily "static" in nature, are intended to show that equipment has been installed and set up correctly, and is ready for the "dynamic" verification test procedures to occur later.

The pre-verification test procedures typically include ensuring that:

- Piping connections are correct
- Pressure and leakage testing and flushing and cleaning have been completed
- Alignment, direction of rotation, and lubrication of rotating equipment are correct

- Electrical equipment is properly fused and grounded
- Voltage and amperage measurements are taken

Verification of Control Device Function (Level 2, 3)

The control system is one of the most important components in a modern chilled water plant. In many ways it is the most complex component. As such, it is critical to ensure that the control system is set up and operating properly. Control systems contain numerous individual control devices, all connected to a "brain" that tells each device what to do and when to do it. These devices need to be closely checked to make sure they are working correctly. This can have a significant impact on making the chilled water plant work optimally.

The control system specification should require that each control device in a chilled water plant be individually checked to ensure that it is working properly. Unfortunately, even when calibration is required in a specification, the control contractor all too often will say that there is no need to calibrate control devices after installation because they are calibrated at the factory. This logic leads to many control systems, and hence the equipment they control, that never function as well as they could.

Critical sensors in chilled water plants whose function must be verified include:

- Chilled water supply and return temperatures
- Condenser water supply and return temperatures
- Differential pressure sensors
- Flow sensors used for control purposes
- Current, voltage, and power sensors used for control purposes
- Humidity sensors

Commissioning Stories

In a project with a plate-and-frame heat exchanger for free cooling, pre-verification checkout indicated a strange problem. Temperature readings from the control system seemed to indicate a "perpetual energy machine." It appeared that temperature sensors might be reversed. However, the controls contractor insisted that they had all been checked and that they were installed correctly. The local thermometers seemed to match the sensor readings because, at the time, there was a small delta-T across the heat exchanger. Further investigation proved that in fact, the supply and return sensors were reversed. The controls contractor was, of course, thoroughly embarrassed by this.

In another project, a differential pressure sensor that was reading across the supply/return loop in the building appeared to be reading within the desired range, although the reading never seemed to change, even with changes in pump speed. Readings from a test gauge indicated that the sensor had failed. The controls contractor had not caught this. Replacing this sensor (which was covered under warranty) saved the owner a significant amount of money in wasted pumping energy.

TAB Issues (Level 2, 3)

Verifying that TAB work has been completed correctly is an important aspect of the commissioning process. TAB report results for critical items in chilled water plants should be checked by the commissioning authority to ensure they are valid. This verification should be done using the same instrument or instruments that the TAB contractor used to take readings. Or, if the commissioning authority has instruments of higher quality, these could be used.

Examples of TAB results to check include:

- Chilled and condenser water flow rates
- In multiple cooling tower plants, proper water distribution across all towers
- Cooling coil data
- Pump data, particularly shut-off head measurements

Pump shut-off head measurements are generally part of TAB work. The results from the TAB report for pumps should be checked as part of the commissioning process to verify that the pump impellers are the correct size. This is done by measuring pressure at the pump inlet and outlet with the pump discharge valve open, and by measuring the outlet pressure with the discharge valve closed. These readings should be taken at pump taps on the pump casing (typically on flange connections), not in the piping upstream or downstream of the pump. These values are then plotted on the manufacturer's pump curve for the specific pump. The results will fall on the line corresponding to the installed impeller size.

Verification Tests (Level 1, 2, 3)

Verification tests are "dynamic" tests conducted to confirm that the equipment and systems perform correctly during all normal and alarm operating conditions.

Critical verification tests for a chilled water plant include:

- Equipment start/stop sequences in normal and abnormal modes
- Chiller staging in multiple chiller plants
- Pump staging in multiple chiller plants
- Cooling tower fan staging
- Reset sequences
- Economizer operation

In general, verification testing proceeds from component level to system level to intersystem level. The actual testing of the chilled water plant equipment and systems takes place using the verification test procedures developed earlier. All TAB work and all control system setup and calibration must be complete before this testing occurs. The commissioning authority coordinates this testing and fills out the test procedure documentation. The appropriate contractor or contractors operate the equipment in the various modes as directed by the commissioning authority.

Commissioning Stories

In a project for a critical data center, testing to ensure that plant equipment quickly came back on-line in a power failure uncovered a unique problem. The chillers were equipped with remote starters. When a power failure was initiated, backup power was supposed to bring the plant equipment back on-line within two minutes. However, the remote starters were not resetting properly, and the chillers were not coming back on-line in the required time frame. The manufacturer's local representative was called to help figure out the problem. Applying numerous different settings to the starters did not solve the problem.

After spending a couple of frustrating days on the problem with no success, a lead engineer from the factory was brought in. Eventually the puzzle was solved: a current was being generated in the chiller motor windings while it was spinning toward a stop. This current was feeding back to the starter, preventing proper reset (the chillers would start fine if there were a wait of two to three minutes for the motor to stop spinning before restoring power). Luckily, the problem could be solved with an electrical jumper and a remedy was relatively simple once the problem was figured out.

In that same project, testing found a number of control sequences that needed to be modified to achieve proper operation. These sequences impacted chillers, pumps and towers.

In another project, a building was false loaded by heating it overnight to try to temporarily achieve full cooling-load conditions. This process uncovered a serious problem with the chilled water distribution system in the building. The building was part of a campus system, and had a fairly large pump (50 hp) with *VFD* to supply the building with chilled water from a central plant. Technically, the pump in the building was a tertiary pump. The primary pump served the chiller at the plant, and the secondary pump moved water from the plant to the building. With a large cooling load, the chilled water supply temperature to the building would not drop below about 52°F (water from the plant was 42°F). This condition occurred when the control valves at the *AHU's* in the building were 90% or more open, and the pump/VFD was operating at 95% flow or higher.

Investigation showed that short-circuiting was occurring in the decoupler line. A valve was placed in the decoupler line (the valve should not have been there), and when it was throttled toward closed, the water temperature to the building would quickly drop to about 42° - 44°F. If testing had not occurred, this problem would not have been discovered until a hot summer day, when the system most likely would not have been able to meet the cooling setpoint in the building.

Functional Performance Tests (Level 3)

If required in the project, these procedures are undertaken to determine the as-installed capacity and performance (overall kW/ton or other criteria) of the chilled water plant. To save time, this work may be done in conjunction with the verification testing. The advantage would be to avoid setting up all equipment again. While it is recommended that chillers be factory tested for performance, field testing can uncover problems arising from installation.

Example

In one project, two identical chillers had matching results from the factory testing. Field testing showed that one machine's efficiency had degraded since the factory testing. Investigation revealed that tubes in that machine had become fouled, causing the difference in performance. Once the tubes were cleaned, both machines operated at the specified performance level.

Operator Training (Level 2, 3)

The commissioning authority coordinates training for the *O&M* personnel to ensure that they adequately understand the operating and maintenance details for the chilled water plant. The commissioning authority should interview the O&M staff to determine where to best focus the training to meet their needs. Training should include classroom sessions to cover general material and hands-on sessions to learn equipment-specific functions. It is recommended that all training sessions be videotaped for future training purposes. Videotaping may be included in the scope of work for the commissioning authority if the owners d want to do it themselves. Experience has shown that when training sessions are videotaped, contractors and others doing the training tend to be better prepared.

Systems Manual (Level 2, 3)

The systems manual is an expansion of the traditional operation and maintenance manual, filling in gaps that exist in typical submitted manuals. It should provide the information needed to understand, operate, and maintain the chilled water plant, and to inform other parties about the system. One important component of the systems manual is a thorough written description of the installed system including intended capabilities and limitations. The commissioning authority should assemble the systems manual, with input from the design professional and the appropriate contractors. For more information on the systems manual, refer to ASHRAE Guideline 1-1996, *The HVAC Commissioning Process*.

Information in the systems manual includes:

- Statement of design intent
- A narrative of how the system is intended to work including a discussion of its capabilities and limitations
- Schematics of the control system and other system components
- Acceptance test procedures and results
- As-built sequences of operation
- Recommended record keeping procedures including sample forms, trend logs, etc.

Occupancy Phase

Commissioning should be considered an ongoing endeavor. After construction is finished, testing procedures are complete, the chilled water plant is operating in a top-notch manner, and the owner has accepted the project, there is still a need to monitor performance and modify control strategies when use patterns or load profiles change. There are many types of changes that are likely to occur over time that can negatively affect the chilled water plant's performance. Ongoing commissioning and long-term performance monitoring can help ensure optimal performance throughout the life of the chilled water plant.

Ongoing Commissioning (Level 3)

Ongoing commissioning is the continued adjustment, optimization and modification of the chilled water plant to maintain as-commissioned operation and performance. It includes updating documentation to reflect setpoint adjustments, maintaining and calibrating the system, and making other significant modifications that might occur over the life of the plant. The plant operating staff is obviously an integral part of this work and should understand the importance of it.

Methodical system testing should be conducted periodically to see how the plant is performing. The same verification test procedures that were used during the acceptance phase should be used for this periodic testing. It is recommended that this testing be conducted every three to five years over the life of the plant.

Long-term Performance Monitoring (Level 3)

Ongoing performance monitoring during system operation helps diagnose problems and keep the system running optimally. Modern control systems allow information from controlled points to be recorded at set intervals and logged to memory. Data for long-term performance monitoring and testing can be collected using sensors installed as part of the control system.

DEFINITIONS

Acceptance tests. Testing that occurs during the acceptance phase including pre-verification, verification, and functional test procedures.

Accuracy. Conformity of the indicated value to the true value.

Actuator. The motor and spring assembly that provides the motive force to automatically operate a control valve.

Adiabatic saturation. Water is evaporated from liquid to vapor to decrease the temperature of the water in a thermal process where no heat is transferred.

Analog-to-digital. The process of converting an analog signal (e.g., a variable voltage or current) to a digital signal (e.g., a data stream of bits).

Axial fan. In-line propeller type fan.

BACnet Gateway. A gateway that converts communication to the BACnet protocol standard developed by the American Society of Heating, Refrigeration, and Air Conditioning Engineers (ASHRAE).

Baffle. Plates inserted in the path of the fluid flow to allow for complete mixing.

Balance Valves. Manual valve in piping system that can be adjusted to regulate flow distribution.

Basket Strainer. Device used to separate solids from the liquid containing them, similar to a filter. Filter basket is perpendicular to fluid flow.

Bits. Binary elements of a digital system.

Bullhead Tee. A pipe tee connected in such a way that the flow enters the side and exits out both sides.

Burst Pressure. Pressure at which a vessel is subject to failure

Bypass Valve. Automatic valve that controls flow through a by-pass pipe typically between the supply and return of a chilled water system.

Canned Vertical Turbine Pumps. Turbine pumps that are enclosed by a pipe sleeve to allow use outside of a tank or sump.

Cavitation. Formation by mechanical forces of vapor within liquids: specifically, the formation of vapor cavities in the interior or on the solid boundaries of liquids in motion, where the pressure is reduced to a critical value without a change in ambient temperature.

Centrifugal Fans. Squirrel cage blower.

Change of Value Threshold. The smallest change for which a change in signal will occur (see Resolution).

Check Valve. Valve allowing fluid flow in one direction only.

Commissioning Team. This is essentially the project team. It includes the commissioning authority, owner, design professional, general contractor/construction manager, mechanical contractor, electrical contractor, controls contractor, TAB contractor, manufacturer's representatives, and others as necessary.

Common Pipe. The pipe that is shared between two pumping circuits.

Control System Points. Discrete interface elements to a device or system being controlled.

Control System. A system that directs or regulates devices or systems in accordance with process that includes comparison with desired outcomes.

Control. Directing or regulating in accordance with process that includes comparison with desired outcomes.

Cooling Ponds. Small body of water used to reject heat by evaporation.

Cross-Feed Manifold. A device for connecting a differential pressure sensor such that the device can be installed and removed from service without it being subject to differential pressures above the differential at the points being monitored.

Crossover Bridge. The branch pipe that connects the supply to the return from which secondary or tertiary pump circuits are attached.

CurrentTransducer. A current transformer with conversion as required to convert the ac output to a suitable analog monitor signal.

Current Transformer. A small transformer that employs a power wire as its primary and provides an AC signal that is a function of the current traveling through the primary wire.

DDC Control. Direct digital control. Mode of control wherein digital outputs are used to control a process or element directly.

Decoupled. Hydraulically independent.

Dedicated Pump. Pump whose use is intrinsically connected with another piece of equipment such as a chiller or a cooling tower.

Delta-P Control Valve. Trade name for a pressure independent two-way control valve that internally compensates for pressure fluctuations in the piping while maintaining constant flow for a given flow setting.

Dew Point Temperature. Temperature at which water vapor has reached the saturation point (100% relative humidity).

Diaphragm Gas Meter. A device for measuring gas flow that employs a diaphragm for the flow measurement.

Direct Drives. The impeller is coupled directly to the motor shaft and rotates at the same speed as the motor.

Discharge Air Velocities. Air speed at outlet of cooling tower.

Double-Seated Valves. Valve where flow is controlled by two circular disks forced against or withdrawn from annular rings or seats that surrounds the openings through which flow occurs. Upstream pressure acts on one side of one disk and the opposite side of the other disc resulting in smaller system forces on the actuator.

Dynamic Compression Devices. Compression of gas is due to the rotation of the impeller that imparts velocity (kinetic energy) to the gas molecules that in turn is converted to static pressure (potential energy) in the diffuser.

Economizer Dampers. Dampers used in the airside economizer process, which refers to the use of outside air to decrease or eliminate cooling load.

Electric Actuation. Actuation whose force is provided by an electric motor or other electric means.

Eliminators. Sheet metal plates formed in a pattern so that water droplets are removed from moisture-laden air that passes through.

End-To-End Accuracy. The accuracy of control or measurement for a system wherein the signal passes through at least one intermediate step and the accuracy is considered at the end points.

Equal Percentage Valves. A valve whose flow characteristics increase exponentially as the valve position increases. This valve characteristic is employed to offset the characteristics of most heating and cooling coils whose capacity characteristics are complementary.

Equalizer Flume Weir Gate. Passage between two cooling tower basins to maintain a common liquid level.

Equalizer Piping. Piping connecting basins of multiple cooling towers to maintain a common liquid level.

Expansion Tank. Partially filled tank, operating at atmospheric pressure at the top of a water system for the accommodation of volume expansion and due to the contraction of water.

Fill. Portion of cooling tower which constitutes the primary heat transfer surface, sometimes called packing.

Fin Density. The number of fins on a coil per unit of finned tube length.

Fixed Orifice. Machined fixed opening that acts as a throttling device between high-pressure and low-pressure side of refrigeration process.

Float Valve. Automatic modulating thermal expansion valve driven by a ball that floats on a liquid reservoir to maintain a constant level.

Functional Performance Tests. Test procedures to determine as-installed capacity, and performance.

Gateway. A means of connecting two or more systems or networks and converting dissimilar signals or protocols.

Gear Drives. A transmission that is placed between the motor shaft and the impeller rotor to increase the speed of the impeller relative to the motor.

Globe Valves. In a globe valve flow is controlled by a circular disk forced against or withdrawn from an annular ring, or seat that surrounds an opening through which flow occurs. Valves are typically used where throttling of flow is required.

Greenhouse Effect. Process where short wave radiant energy from the sun is absorbed and converted to long wave radiant energy and is effectively trapped in the atmosphere.

Groove Volume. The volume between the rotor, gaterotor, and casing of the compressor.

Heat Transfer Coefficient. Heat transmission in unit time through unit area of a materiel or construction and the boundary films, induced by unit temperature difference between the environment on each side.

Hydraulically Independent. The flow or pressure change in one pumping circuit does not affect the flow or pressure in the other circuit even though they share a common pipe.

Hydraulically Remote Coils. Coils that are the furthest from the pump in terms of pressure drop; often but not necessarily the most remote physically from the pump.

Impeller. The blades of a rotor that are used to impart velocity to the fluid.

Inductive Devices. Air flow drawn through the tower because of suction caused by high velocity water flow rather than by a fan.

Integral Combustion Heat Source. Source of heat for absorber is from gas or oil furnace within the machine.

Isolation Valves. Valves that allow a piece of equipment to be isolated from the rest of the system for maintenance or replacement.

Jacket Water. The fluid circulated within an engine used to reject heat.

Lag Machine. In a sequence of operation the machine (chiller, pump, cooling tower, etc.) That is started after another machine (lead machine).

Laminar Flow. Streamline fluid flow where all particles move in substantially parallel paths. Flow that is not turbulent.

Laminar Range. Indicates that fluid flow is streamline and not turbulent.

Lead - Lag. For multiple equipment in parallel, the first machine to be turned on is the "lead" machine, that last is the "lag" machine.

Lead Machine. In a sequence of operation the machine (chiller, pump, cooling tower, etc.) That is started first.

Linear Actuation. Actuation that provides linear movement.

Magnetic Flow Meter. An electronic flow meter that measures the flow by sensing flow induced changes in a weak magnetic field in the medium. These devices are often called "magmeters."

Marine Water Boxes. Special heat exchanger heads that allow access to the heat exchange tubes without disassembling the connecting piping.

Monitor Points. Discrete interface elements to a device or system that provide information about the operation of that device or system.

Multiple Hydronic Circuits. Many different flow paths in a water piping system.

Networked Sensor. A sensor whose value is multiplexed and transmitted across a digital network for use at another node in the network.

Orifice Plate Flow Meter. A device that measures flow by measuring the pressure difference of the flowing fluid across a selected orifice plate.

Paddle Turbine Meter. A device that measures flow by inserting a small paddle wheel in the fluid and measuring the rate of rotation of the wheel.

Part-Load. Operation at less than 100 percent load.

Passes. The number of times the fluid is passed through the heat exchanger before exiting.

Performance Specification. A document that specifies the performance a system is to provide as opposed to prescriptively listing how that performance can be attained.

Piezoresistive. Materials whose electrical resistance change with pressure on the material.

Pneumatic Actuation. Actuation whose force is provided by a air cylinder.

Pneumatic Control. Control devices that utilize pneumatic (compressed air) signals to control inputs and outputs.

Pony Chiller. A chiller that is significantly smaller than the other chillers in the plant, for handling very low loads.

Positive Displacement Machine. Refrigerant compressor that has capability of producing an infinite pressure only as limited by the physical constraints of the materials and equipment.

Power Sensor. A sensor that combines current and voltage sensing to provide a signal based on the power draw of the equipment being monitored.

Precision. The degree of accuracy in measurement or control.

Pressure Taps. Small valved connections in piping of vessels that permit the temporary connection of pressure monitoring equipment for testing or troubleshooting.

Pre-Verification Tests. Primarily "static" checks to verify that equipment has been installed properly, and is ready for further testing. May be done in conjunction with normal start-up procedures.

Process Chilled Water. Chilled water used cool process loads.

Process Loads. Heating or cooling loads that are the result of a manufacturing or other process.

Process or Data Center. Refers to space wherein main frame computers are located and typically have high internal heat gains and/or humidity requirements.

Propeller Fan Towers. Cooling towers with propeller fans, as opposed to centrifugal blowers.

Pumping Head. Total differential pressure (psi) or level (feet of water) created by the pump at a given speed and capacity and includes static head, velocity head, and frictional losses in piping and fittings.

Rangeability. The ability of a system or device to measure or control over various ranges (e.g., the difference between maximum and minimum values)

Refrigerant Head Pressures. The gauge pressure of the hot gas directly downstream of the compressor.

Remote Differential Pressure Controller. Control device consisting of pressure sensors and controller that is used to control a set difference in pressure in a location remote from the cooling plant.

Reset. Control strategy where a temperature (or pressure, etc.) Set point changes as a function of some variable such as outdoor air temperature or piping differential pressure.

Resolution. The capacity of a signal to transmit or receive change in the controlling element.

Reynolds Number. Dimensionless number that indicates how turbulent or laminar fluid flow is in a pipe.

Riding Pump Curve. Term to describe a pump operating without speed control as the flow through it varies.

Rotary Turbine Flow Meter. A device that measure flow by inserting a rotary turbine into the fluid and measuring the rate of rotation of the turbine.

Series Booster Pumps. Pumps piped in series used to raise the pressure of a hydronic circuit.

Series Piping. When pumps are piped in such a way that the discharge of one pump is directed into the inlet of a second pump with the result that the flow rate of each pump is the same but the pressures of each pump are additive.

Solenoids. Valves with actuators that operate in a two position (open/closed) mode.

Start-Up Transients. Initial unstable condition at start-up.

Step Reset. Reset that changes incrementally in steps rather than smoothly.

Storage Tank. Container plus all contents used for storing thermal energy.

Suction Diffusers. Device mounted at the inlet to a pump to straighten water flow, in lieu of providing long runs of straight pipe. May optionally house a strainer.

Sump. Collection basin for water after it has gone through the cooling tower.

Synchronized Timing Gears. Gears that allow twin rotors to rotate at exactly the same speed.

Temperature Bin Method. Energy calculation method, usually used in prediction, in which the annual (or monthly) energy use of a building is calculated as the sum of the energy used for all of the outdoor temperature bins.

Temperature Tolerance. The tolerance of a device or system to be exposed to external temperatures

Thermal Siphon. Flow that occurs due to fluid density differences caused by temperature differences within the same system or pipe.

Three-Pass Evaporator. Heat exchanger where the fluid is directed through the heat exchanger three times.

Three-Way Valves. Valve having either a single inlet and two outlets (diverting) or two inlets and a single outlet (mixing), in which either one or the other is open. Valve can modulate flow in infinite combinations between either inlet or outlet as the case may be.

Throttling Device. A device that separates the high-pressure liquid refrigerant in the condenser from the low-pressure gaseous refrigerant in the evaporator. This device can be an orifice, float valve, or thermal expansion valve.

Tolerance Limit. The point at which the reliability of a device or system may be affected by exposure to certain conditions.

Trend Logs. Record of events taken on a regular schedule or equal time intervals.

True RMS. True Root Mean Square. This is the mathematical relationship that is used to determine the value of AC voltages and currents.

Turn-Down. Operation at part load.

Two-Pass Evaporator. Heat exchanger where the fluid is directed through the heat exchanger twice.

Two-Way Isolation Valves. Automatic control valves that have two positions, open and closed, used for controlling flow to equipment.

Ultrasonic Flow Meter. A device that measures flow by sensing its effect on an ultrasonic signal transmitted into the fluid.

Under-Floor Systems. Systems that supply air from an under-floor plenum rather than overhead

Unloading. Process by which a refrigerant compressor reduces its capacity usually by opening the intake valve on one or more of the pistons.

Valve Authority. The ratio of the maximum to minimum pressure drop across a valve over the range of operations.

Valve Flow Coefficient. The number of GPM of water at 60°F that will pass through a fully open valve. Flow coefficient symbol is "CV."

Verification Tests. Series of test procedures to verify that equipment/systems are operating in accordance with the design intent, and contract documents.

Volumetric Flow Characteristics. Changes in the mass flow of refrigerant.

Vortex Flow Meter. A device that measures flow by the movement of a small target that produces a vortex in the fluid.

Water Temperature Reset. Changing the temperature setpoint of the water leaving the evaporator.

Y-Strainer. Device used to separate solids from the liquid containing them, similar to a filter. Filter basket is located at an angle to the fluid stream.

A. SAMPLE CHILLER BID SPECIFICATION

General

Project Description

A. Project Summary

1. The Sample New Building is a new 600,000 ft^2 office tower in San Francisco. The HVAC system will include a chiller plant of 1,100 tons total capacity.

2. The project is a plan & spec' project and currently is in design development. Construction will begin in 1999.

B. Bid and Product Selection Process

1. Chillers will be selected based on a life-cycle cost analysis over a 15-year study period. The analysis will include the impact of chiller efficiency at both design and off-design conditions, variations in tower water entering temperature, and pressure drop across both evaporator and condenser as a function of flow rate. The building and chiller plant will be simulated using a DOE-2 computer model. While this analysis will be done by the design team, some of the operating data are provided here to assist the vendor in selecting chillers that might be the most life-cycle cost effective. The vendor is encouraged to provide up to four selections for the design team's consideration since the energy analysis is complex and the "winning" chiller combination may not be readily apparent.

2. The purpose of this specification is to establish equipment pricing. However, this is not the final bid. The final bid will be obtained by the contractor when the project is put out to bid.

It is the design team's intention that the best products and final selection be made as part of the design process. Vendors should offer their best products and prices now, so that value-engineering or product substitution does not occur when the contractor solicits the final bids.

3. To ensure that vendors provide the best pricing now, as opposed to later when contractors solicit bids, the design team stipulates the following:

   a. All pricing will remain strictly confidential until after contractor bid. Only the design team will see prices. Prices will not be included in any written reports to owners prior to bid. (Prices will be disclosed to contractors and others after the contractor bid.)

   b. [The engineer should choose one of following:]

   Alternate 1: The chillers selected by this life-cycle cost analysis will be pre-purchased and assigned to the successful mechanical contractor for installation.

   Alternate 2: The chillers selected by this life-cycle cost analysis will be flat-spec'd. Substitutions by the contractor during the bid process will not be allowed.

Alternate 3: The chillers selected by this life-cycle cost analysis will not be flat-spec'd and will not be pre-purchased. However, they will be the final products specified and the basis of design. Substitutions by the contractor during the bid process will be allowed only as alternate bids. Unless savings are substantial (including first costs, energy costs, and redesign costs), the alternate products will not be selected. Hence, it is extremely likely that the product selected as part of this analysis will be purchased and installed. Vendors who price their product during the final contractor bid at a price lower than the one provided as part of this pre-bid will jeopardize their relationship with the design engineers on future projects. *[This third alternate is for projects where the owner and construction manager do not trust the consulting engineer to get competitive prices.]*

4. Historical maintenance costs, factory and dealer support, and general reputation for reliability will also be considered in selection.

Scope

A. Work Included:

1. Centrifugal and/or screw chillers.
2. Ancillary equipment such as starters and controls as specified herein.
3. Factory performance testing.
4. Field start-up and testing.
5. Freight to jobsite.
6. All sales taxes, including that on shipping if applicable.
7. Operation Instructions.
8. Warranty.

B. Work Excluded:

1. Refrigerant detectors.
2. Installation. (The design team will estimate net installation costs of vendor proposals. Please advise in proposal if there are any installation advantages over competitors' products that are not self-apparent.)

Bid Instructions

A. Pricing Options:

1. Vendors are encouraged to provide pricing for up to four different chiller combination options.
2. Voluntary Alternates: Vendors are encouraged to provide alternates/exceptions to these specifications to reduce costs or add value.

B. Proposals. Two (2) copies of bid proposals shall be provided and include all of the following:

1. Completed bid form (see attached).

2. Equipment performance data sheets (see attached) for each proposed chiller. Additional data may be provided at bidders' option. Data in electronic format (spreadsheet on diskette) is <u>required</u>. An electronic copy of the data sheets can be sent to vendors upon request. Note: Data must be based on an ARI-certified computer program **without** credit for ARI allowed tolerance. (See section 2.10 below regarding testing.)

3. Acoustical performance: sound power levels 1 meter from point of highest sound level.

4. Pre-submittal of each proposed chiller showing physical dimensions, electrical requirements, wiring diagrams, etc.

5. Names of references on projects where the proposed products have been installed.

Design and Operating Conditions

A. General Conditions and Constraints

1. The plant runs continuously year-round due to ancillary computer room loads. Primary loads are air handling systems with outdoor air economizers.

2. The chilled water pumping system is primary only, variable flow. Controls (by others) will be provided to maintain minimum flow rates through chillers.

    a. Because of the primary-only pumping configuration, the chiller will experience variable flow on both sides but especially on the evaporator side. The CHW flows can drop within seconds by as much as 30%. By bidding on this project, the chiller manufacturer guarantees that the chiller can handle such variations without nuisance tripping, or they shall provide control sequences with the bid that will allow the chiller to operate through such sudden flow disturbances. (The drop in flow will occur any time a new chiller is brought on line. Since this is an occurrence that is known by the control system prior to it occurring, the controls may be able to minimize potential tripping problems by, for instance, demand-limiting the chiller before the flow is allowed to drop. Specify what may be required with bid.)

    b. At least two chillers must be provided.

    c. No redundancy is required, i.e., the total of chiller loads equals the total design plant load.

    d. Chillers need not be the same size. If chillers are unequally sized, the minimum condenser and chilled water flow of the large chiller must be less than the design flow of the small chiller so that the small pumps can operate with the large chiller in case of pump or chiller failure. (Pumps will be headered to allow either to serve either chiller.)

3. Head pressure control: Cooling towers will have fan speed control only, no bypass. The fan will be controlled at whatever minimum setpoint the vendor specifies (see performance bid form). If additional head pressure controls are required for the lead chiller (the one that will operate during cold ambient conditions for computer room loads), they shall be provided by the vendor. Also provide head pressure control for other chillers if necessary (it should not be). The vendor is entirely responsible for head pressure controls (e.g., controller, wiring excluding field wiring, etc.) except motorized butterfly control valves will be provided, installed, and wired by others.

B. Design Conditions:

1. Chiller:

   a. Capacity: 1100 tons total of all chillers.

   b. Design entering water temperature: 42°F

   c. Design leaving water temperature: 57°F

   d. Fouling factor for performance calculations: 0.00025

2. Condenser:

   a. Design entering water temperature: 73°F

   b. Design leaving water temperature: 85°F

   c. This is an average condition at peak design load. The actual leaving water temperature from each chiller may vary. For instance, if the lead chiller benefits significantly from high flow (lower head), the lag chiller(s) may be selected at lower flow to compensate. (Pump energy will be taken into account, so there are diminishing returns to this strategy.)

   d. Minimum entering water temperature: as limited by vendor (see bid form). Specify as a fixed minimum temperature or as a differential above the leaving chilled water supply temperature. This will be used in the modeling as a limitation on cooling tower control.

   e. Fouling factor for performance calculations: 0.0005

3. Compressor

   a. 460 V, 60 Hz., 3

C. Selection considerations:

1. Chillers may be piped in series or in parallel. (If series piping is selected, specify in quote. The chilled water temperatures on the bid performance form will need to be modified.)

2. At least one chiller will operate 8760 hours per year to serve a small process loads. The load profile is approximately as shown in the table and chart below (drawn from the DOE-2 computer model):

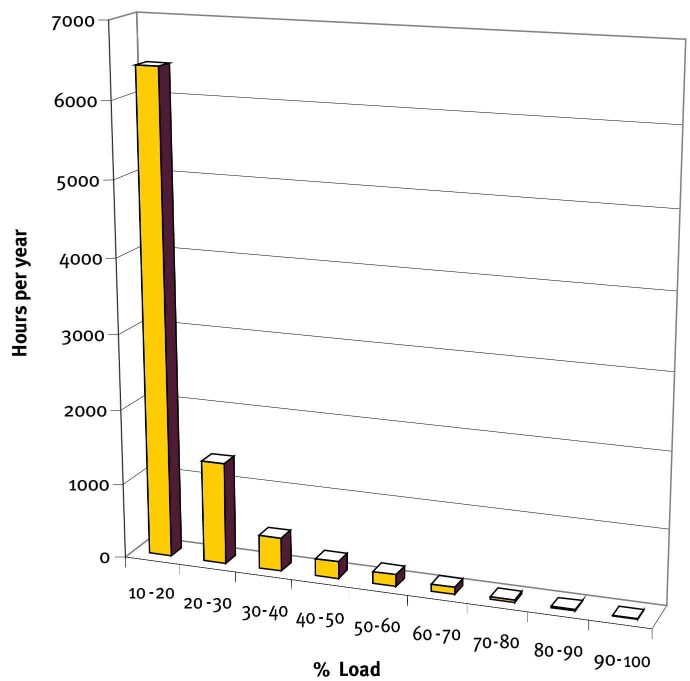

Figure A1-1:
Annual Load Profile

Table A1-1:
Annual Load Profile

Load Range (%)	0-10	10-20	20-30	30-40	40-50	50-60	60-70	70-80	80-90	90-100
Hours per year	0	6420	1357	447	231	159	95	31	14	6

Clearly, from this profile, it is imperative that the lead chiller be capable of operating at 100 tons to 200 tons efficiently. The plant seldom operates above 50% load.

3. Chilled water is variable flow and thus flow rate can increase above the nominal level during low load and mild weather conditions when delta-T degrades. Chillers that have a high maximum flow rate will therefore be favored for the lead chiller. (The lead chiller should be capable of operating at 200% of design flow.)

4. Pump energy impacts will be considered in the life-cycle cost analysis. (The more chillers provided, the lower the pump energy costs.)

5. The towers are capable of maintaining entering condenser water temperature below design conditions except for very few hours in the summer. The towers will be very efficient and thus the likely control scenario is to maintain as cold a condenser water entering temperature as possible. (See bid form for location to specify limiting leaving water conditions.) If the vendor feels that another control sequence will be preferred, please specify with bid.

6. Selection ideas: The following are possible size/selection scenarios that the vendor should consider. However, the vendor is not limited to these selections and they may not even be the best design concepts for the project. The vendor may suggest any chiller size/quantity combination and may propose from one to four alternatives.

a. Suggestion 1:

    i. CH-1: 550 tons, very high efficiency dual compressor

    ii. CH-2: 550 tons, medium/low efficiency dual compressor

b. Suggestion 2:

    i. CH-1: 200 tons, very high efficiency

    ii. CH-2: 900 tons, high efficiency dual compressor

c. Suggestion 3:

    i. CH-1: 200 tons, very high efficiency

    ii. CH-2: 350 tons, high efficiency

    iii. CH-3: 550 tons, medium/low efficiency

d. Suggestion 4:

    i. CH-1: 200 tons, very high efficiency, variable speed drive

    ii. CH-2: 350 tons, high efficiency

    iii. CH-3: 550 tons, medium/low efficiency

Materials

General

A. Trane, Carrier, York, McQuay.

B. The unit shall be completely factory packaged including evaporator, condenser, compressor, motor, starter, lubrication system, control system, and all interconnecting unit piping and wiring. Any field installation that is required, other than piping and normal control and power wiring, shall be clearly identified with the bid or shall be included in the vendors pricing.

C. Performance shall be certified in accordance with ARI Standard 550/590-1998.

D. Cal-code and UL listed.

Compressor

A. Screw and/or centrifugal.

B. Single or multiple compressor. Multiple compressor chillers need not have separate refrigerant circuits. (Redundancy is provided by multiple chillers.)

C. Acceptable refrigerants: R-123 and R-134a.

Condenser and Evaporator

A. Tubes may be high-efficiency, internally enhanced type. Each tube shall be roller expanded into the tube sheets and be individually replaceable.

B. R-134a machines: the refrigerant side shall be designed, tested and stamped in accordance with ASME Boiler and Pressure Vessel Code, Section VIII- Division 1.

C. The evaporator shall have a spring loaded refrigerant pressure relief device sized to meet the requirements of ASHRAE 15 Safety Code for Mechanical Refrigeration.

D. Include refrigerant pump-out/reclaim connections.

Water Boxes

A. Cleanable shell and tube type. Designed for 125 psig design working pressure and tested at 200 psig minimum.

B. Stub-out water connections having Victaulic grooves shall be provided. Vent and drain connections with plugs shall be provided on each water box.

C. Water box connections that are not from the same end of the chiller will be given a piping cost penalty in the life-cycle cost comparison.

Purge system (R-123)

A. "Near Zero" high efficiency, air-cooled, able to operate independently from chiller.

B. Unit shall have lights to indicate running, fault indication, and service-needed; and an elapsed time meter.

Factory Insulation

A. Insulation shall factory installed.

B. Minimum 3/4 in. thick closed cell foam insulation (e.g., Armaflex II).

C. Exclude insulation of water box end section at piping connections which will be field insulated.

Controls

A. Each unit shall be furnished complete with a digital control system in a lockable enclosure, factory mounted, wired, and tested. The control center shall include a minimum 40 character alphanumeric display showing all system parameters in the English language with numeric data in English units.

B. Digital programming of essential setpoints through a color coded, tactile-feel keypad shall include: leaving chilled water temperature; percent current limit; pull-down demand limiting; and remote reset temperature range.

C. All safety and cycle shutdowns shall be enunciated through the alphanumeric display and consist of day, time, cause of shutdown, and type of restart required. Safety shutdowns (manual restart) shall include: high condenser refrigerant pressure; low evaporator pressure; low evaporator-condenser differential pressure; high oil temperature; low oil pressure; high oil pressure; motor overload, and sensor malfunction. Cycling (automatic restart) shutdowns shall include: low water temperature; cooler/condenser water flow interruption; power fault (loss of power, low voltage); and anti-recycle.

D. System operating information shall include: return/leaving chilled water temperatures; return/leaving condenser water temperatures; evaporator/condenser refrigerant pressures; differential oil pressure; percent motor current; evaporator/condenser saturation temperatures; compressor discharge temperature; oil temperature; operating hours; and number of compressor starts.

E. Security access shall be provided to prevent unauthorized changing of setpoints, and to select local or remote control of the chiller, and to allow manual operation of the pre-rotation vanes and oil pump.

F. Control center shall be able to interface with a building automation system to provide:

1. remote chiller start/stop
2. reset of chilled water temperature
3. demand limit
4. chiller fault (shut down on a safety requiring reset)

G. Flow switch: provide for evaporator flow if required for safe operation and freeze protection.

Compressor-Motor Starter

A. Unit mounted starter.

1. Wye-delta, star-delta, or solid state.

B. Variable-speed drives, option, if proposed:

1. Need not be factory mounted if that is not an option. Field wiring costs will be added in the life-cycle cost analysis.
2. Bypass starter not required.

C. The chiller shall have a single point wiring connection at the starter, i.e., the starter shall include factory installed transformers for all lower voltage devices (e.g., purge units, controls) that are provided with the chiller.

Startup and Operator Training

A. A factory trained, field service representative shall supervise final leak testing, charging, and initial startup and conduct concurrent operator instruction.

B. Submit six (6) copies of operations and maintenance manuals. Manuals shall be bound with index and tabs and include the following:

1. Equipment submittals.
2. Equipment performance curves or capacity tables.
3. Operating and maintenance instruction sheets and parts list.

Factory Performance Testing

A. The chiller manufacturer shall conduct a certified performance test in accordance with ARI 550/590-1998 test procedures to be witnessed by a representative of the Buyer.

B. If variable speed drives are proposed, the factory tests and performance guarantees apply to the drive and chiller as a unit.

C. The chiller shall be tested at three operating points randomly chosen by the Buyer one week prior to the test. No tolerance (such as that normally allowed by ARI 550/590) on efficiencies or capacities will be accepted.

D. Chiller manufacturer shall repair or replace equipment, at no cost to the Buyer, until equipment is certified by ARI 550/590 test procedures to meet the performance indicated in the manufacturer's proposal. Should failure of tests result in a delay exceeding two weeks, the chiller manufacturer shall also pay liquidated damages in the amount of $1000 for each day the project is delayed while the chiller is being repaired or replaced.

E. The costs of travel for the Owner's representative for the initial test will be borne by the Owner. However, the chiller vendor shall pay all transportation costs and hotel costs (if required) associated with witnessing any required factory re-testing for one witness originating from the San Francisco Bay Area.

Shipment

A. The unit shall be completely assembled, with all main, auxiliary, and control piping installed, controls wired, leak tests completed, air runs tested, and charged with dry nitrogen (2 to 3 psig). The oil charge and miscellaneous materials shall be packed separately. The refrigerant charge shall be shipped concurrently or separately in cylinders for field evacuation and charging of unit.

B. Include all freight charges to the jobsite.

Warranty

A. Chiller manufacturer is to provide one year parts and labor warranty on entire chiller. The warranty period is defined as one year from date of startup or eighteen months from shipment, whichever occurs first.

B. Warranty shall include replacement refrigerant if charge is lost due to a fault covered by the warranty.

Chiller Bid Form

Date:

Manufacturer:

Chiller System Option 1			Bid Price
	❏CH-1	Capacity: Model:	
	❏CH-2	Capacity: Model:	
	❏CH-3	Capacity: Model:	
	❏Voluntary alternate, add or (deduct):		
Chiller System Option 2			
	❏CH-1	Capacity: Model:	
	❏CH-2	Capacity: Model:	
	❏CH-3	Capacity: Model:	
	❏Voluntary alternate, add or (deduct):		
Chiller System Option 3			
	❏CH-1	Capacity: Model:	
	❏CH-2	Capacity: Model:	
	❏CH-3	Capacity: Model:	
	❏Voluntary alternate, add or (deduct):		
Chiller System Option 4			
	❏CH-1	Capacity: Model:	
	❏CH-2	Capacity: Model:	
	❏CH-3	Capacity: Model:	
	❏Voluntary alternate, add or (deduct):		

Chiller Performance Form

Option:	Model:
Manufacturer:	Delivery Lead Time (Weeks):

Design Conditions			
❏ CHW fouling factor:	0.00025	❏CW fouling factor:	0.0005
❏ Leaving CHWST:	42°F	❏Entering CWST:	73°F
❏ Entering CHWRT:	57°F		
❏ Design CHW flow:	_______gpm	❏ Design CW flow:	
❏ CHWΔP:	_______ft	❏ Design CW flow:	_______gpm
❏ Design kW:	_______kW (w/o ARI tolerance)	❏ CW Δ P:	_______ft
❏ Design capacity:	_______tons	❏ Design kW/ton:	_______kW/ton
Operating Constraints			
❏ Maximum CHW flow rate:	_______gpm	❏ Maximum CW flow rate:	_______gpm
❏ Minimum CHW flow rate:	_______gpm	❏ Minimum CW flow rate:	_______gpm
		❏ Minimum CWST:	_______°F or _______°F above leaving CHWST)

Part Load Conditions

The following data shall be at the proposed <u>design</u> conditions entered above except where noted. Where the conditions are beyond the range of the chiller, <u>leave the entry blank</u>. "Minimum" refers to the minimum condenser water entering temperature to the chiller as listed above. Do <u>not</u> include ARI tolerance.

CHW Supply temperature °F	CW Supply temperature °F	Part Load Ratio (% of design cap.)	Capacity tons	Coincident Power kW
45	75	100%		
45	72.5	90%		
45	70	80%		
45	67.5	70%		
45	65	60%		
45	62.5	50%		
45	60	40%		
45	Minimum:	30%		
45	Minimum:	20%		
45	Minimum:	10%		

Chiller Performance Form

Option:	Model:
Manufacturer:	Delivery Lead Time (Weeks):

Full Load Conditions

The following data shall be at the proposed design conditions entered above except where noted. The capacity and power inputs are for unmodulated operation assuming no power or current limits with all capacity control devices fully open/100% Where the conditions are beyond the range of the chiller, leave the entry blank. Do not include ARI tolerance.

CHW Supply temperature °F	CW Supply temperature °F	Inlet guide vane position	Capacity tons	Coincident Power kW
42	55	100%		
42	60	100%		
42	65	100%		
42	75	100%		
42	85	100%		
42	95	100%		
44	55	100%		
44	60	100%		
44	65	100%		
44	75	100%		
44	85	100%		
44	95	100%		
46	55	100%		
46	60	100%		
46	65	100%		
46	75	100%		
46	85	100%		
46	95	100%		
48	55	100%		
48	60	100%		
48	65	100%		
48	75	100%		
48	85	100%		
48	95	100%		

C. SAMPLE PRELIMINARY COMMISSIONING PLAN

Overview

This document defines the structure and scope of the commissioning process for the chilled water plant for ABC Company. This commissioning plan addresses the roles and responsibilities of all parties involved in the commissioning process, the scope of work, and a preliminary schedule for completion of commissioning tasks.

Commissioning is a systematic process of ensuring that all systems perform interactively according to the design intent and owner's operational needs.

Commissioning is intended to achieve the following objectives:

1. Ensure that equipment and systems are properly installed and receive adequate operational checkout by the installing contractors.
2. Verify and document proper operation and performance of equipment and systems.
3. Ensure that the design intent for the project is met.
4. Ensure that the project is thoroughly documented.
5. Ensure that the facility operating staff are adequately trained.

Project Information

The project is a 15-story, 540,000 ft^2 high rise in San Francisco. Program space is primarily office occupancy, with some assembly space on the ground floor (training rooms), minor retail on the ground floor, a 5,000 ft^2 data center, and a cafeteria on the 7th floor (where there is also a roof deck). The data center, and various small computer server rooms, operate continuously and place roughly a 50-ton base load on the chilled water plant.

Commissioning Scope of Work

The equipment listed below will be included in the scope of commissioning work for this project.

Chillers

1. Chiller #1 365 tons R-134A w/VFD
2. Chiller #2 735 tons R-134A w/VFD

Pumps

1. CWP-1 845 gpm
2. CWP-2 1680 gpm
3. CHP-1 585 gpm
4. CHP-2 1175 gpm
5. CHP-S1 280 gpm
6. CHP-S2 280 gpm

7. CHP-S3 230 gpm
8. CHP-S4 230 gpm
9. CHP-S5 230 gpm
10. CHP-S6 230 gpm
11. CHP-S7 120 gpm
12. CHP-S8 120 gpm
13. CHP-S9 120 gpm
14. CHP-S10 120 gpm

Cooling Towers

1. CT-1 1263 gpm two-speed fan
2. CT-2 1263 gpm two-speed fan

Control System

Commissioning will include control points and sequences of operation for the following:

1. Chillers
2. Pumps
3. Cooling towers
4. Control valves

Project Team Directory

Owner

ABC Company, Inc.
2525 N. Beacon Ave.
San Francisco, CA

	Telephone	Fax	Email
Thomas Tycoon	123-456-7890	456-0987	tt$@abcco.com
James Welloff	123-456-7676	456-3456	jw$@abcco.com

General Contractor

Modern Construction, Inc.
600 Hilltop Rd.
Concord, CA

	Telephone	Fax	Email
Ken Golden	123-456-7890	456-0987	kg@modcon.com

Commissioning Authority

Verified Functionality, Inc.
256 Independence Ave.
Berkeley, CA

	Telephone	Fax	Email
Tim Q. Markham	123-456-7890	456-0987	tqm@vfi.com

Mechanical Engineer

Excellent Engineering
255 Market St.
San Francisco, CA

	Telephone	Fax	Email
Carl H. Wagner	123-456-7890	456-0987	chw@exceleng.com

Electrical Engineer

Excellent Engineering
255 Market St.
San Francisco, CA

	Telephone	Fax	Email
Melissa Coleman-Cox	123-456-7890	456-0987	mcc@exceleng.com

Mechanical Contractor

Mechanical Systems Specialists, Inc.
525 N. Basin Ln.
Oakland, CA

	Telephone	Fax	Email
Gordon Overton	123-456-7890	456-0987	go@mssi.com

Electrical Contractor

All Phase Electric
2525 N. Ampere Ave.
San Francisco, CA

	Telephone	Fax	Email
Frank Sparky	123-456-7890	456-0987	sparky@hivolt.com

Controls Contractor

Or Approved Equal, Inc.

252 First Ave.

San Jose, CA

	Telephone	Fax	Email
Greg Johnson	123-456-7890	456-0987	gj@equal.com

Test and Balance Contractor

Balance Right

325 Grand Ave.

San Francisco, CA

	Telephone	Fax	Email
John Thompson	123-456-7890	456-0987	jt@balanceit.com

Roles and Responsibilities

All parties involved in the design, construction, and operation of the new chilled water plant will be involved in the commissioning process. Although the participants' roles and responsibilities will vary, and in some cases overlap, the team's goal is to provide a chilled water plant that is in compliance with the plans and specifications, meets the design intent, and is operationally and functionally ready for the owner to take over.

Owner

Facilitates and supports the commissioning process, including:

1. Managing contracts of the A/E, General Contractor, and Commissioning Authority;
2. Arranging for O&M personnel to participate in the commissioning process;
3. Reviewing and approving change orders; and
4. Providing final approval and acceptance of the project.

General Contractor

Helps facilitate the commissioning process by:

1. Working with Commissioning Authority to integrate commissioning activities into the construction process and schedule;
2. Ensuring that subcontractors execute their responsibilities, including commissioning requirements;
3. Reviewing test procedures/checklists;
4. Witnessing startup procedures and test procedures;
5. Assembling and reviewing O&M manuals;
6. Remedying deficiencies identified during commissioning;

7. Ensuring completion of punchlist items; and
8. Obtaining final inspection approvals.

Design Professionals

Help facilitate the commissioning process by:

1. Providing information for basis of design and design intent;
2. Responding to requests for clarification generated during the commissioning process;
3. Assisting in the resolution of any system deficiencies;
4. Reviewing verification and functional test procedures;
5. Reviewing and approving O&M manuals and system manual;
6. Reviewing and approving TAB report;
7. Participating in team meetings as required;
8. Developing punchlists; and
9. Reviewing final commissioning documentation.

Contractors

Demonstrate proper system performance, and help facilitate commissioning by:

1. Integrating required commissioning activities into the construction process and schedule;
2. Ensuring cooperation of specialty subcontractors;
3. Providing equipment submittals for review and approval;
4. Participating in test procedures to demonstrate proper system performance;
5. Ensuring participation of equipment manufacturers in startup, testing, and training activities;
6. Preparing and submitting O&M manuals;
7. Remedying deficiencies identified during the commissioning process; and
8. Ensuring completion of punchlist items.

Equipment Vendors

Perform contracted startup work, and help facilitate commissioning by:

1. Participating in training sessions;
2. Assisting in testing as required to demonstrate system performance; and
3. Assisting in resolution of system deficiencies.

Commissioning Authority

Coordinates the commissioning process and facilitates communication, including:

1. Developing commissioning plan;
2. Performing design review;
3. Attending (or leading, as appropriate) project team meetings;
4. Conducting construction observations and reporting on findings;
5. Developing verification and functional test procedures;
6. Witnessing startup and pre-verification procedures;
7. Verifying TAB results;
8. Coordinating and documenting execution of verification and functional test procedures;
9. Assisting in resolution of deficiencies;
10. Reviewing O&M manuals;
11. Assembling systems manual; and
12. Developing commissioning report.

Commissioning Process

This list provides a brief overview of typical commissioning tasks and the general order in which they will occur.

1. Project kickoff meeting.
2. Ongoing throughout – project meetings to cover project observation, coordination, and planning/scheduling issues.
3. Design review.
4. Review equipment submittals, including startup procedures.
5. Periodic site walkthroughs and documentation during construction.
6. Development of any additional startup procedures/requirements.
7. Development of appropriate test procedures.
8. Execution of test procedures.
9. Correction of deficient items and re-testing if required.
10. Submission of O&M manuals.
11. O&M training.
12. Submission of final commissioning report.

Commissioning Schedule

This section outlines a schedule for commissioning activities, based on the estimated schedule for the entire project. Every attempt will be made to minimize conflicts in scheduling as the project progresses.

Task	Start/Finish
Design review	5/1/02 to 8/30/02
Commissioning plan	6/1/02 to 6/30/02
Commissioning meetings	Ongoing
Submittal review	10/15/02 to 11/15/02
Construction site visits	Ongoing
Develop testing procedures	11/1/02 to 2/15/03
Witness equipment startup	6/1/03 to 7/31/03
TAB work	7/1/03 to 9/15/03
Perform test procedures	9/1/03 to 10/15/03
O&M staff training	10/1/03 to 10/31/03
Punchlist completion	10/31/03
Final commissioning report	12/31/03

D. SAMPLE COMMISSIONING SPECIFICATION

Division 01

Part 1 - General

1.01 Description

A. The purpose of the commissioning process is to provide the Owner/operator of the facility with a high level of assurance that the mechanical and electrical systems have been installed in the prescribed manner, meet the design intent, and operate within the performance guidelines. The Commissioning Authority shall provide the Owner with an unbiased, objective view of the system's installation, operation, and performance. This process does not take away or reduce the responsibility of the system designers or installing contractors to provide a finished product. Commissioning is intended to enhance the quality of system start-up and aid in the orderly transfer of systems to beneficial use by the Owner.

1.02 Scope

A. The functions and responsibility of the Commissioning Authority shall include:

1. Responsibility: The primary point of responsibility is to inform the General Contractor (Construction Manager) and the Owner on the status, integration, and performance of systems within the facility.

2. Information: The Commissioning Authority shall function as a catalyst and initiator to disseminate information and assist the design and construction teams in the completion of the construction process. This shall include system completeness, performance, and adequacy to meet the intended performance standards of each system. Services include observing construction, spot testing, verification and functional performance testing, and providing performance and operating information to the responsible parties, i.e., contractors, design professionals, and the Owner.

3. Quality assurance: The Commissioning Authority shall assist the responsible parties in maintaining a high quality of installation and system performance.

4. Observation of tests: The commissioning Authority shall observe and coordinate testing as required to ensure system performance meets the design intent.

5. Documentation of tests: The commissioning Authority shall document the results of the verification and performance testing directly and/or ensure that the appropriate technicians document all testing. The Commissioning Authority shall provide standard forms to be used by all parties for consistency of approach and type of information to be recorded.

6. Resolution of disputes: The Commissioning Authority is to remain an independent party present on the project with specific knowledge of the project. Should disputes arise, the Commissioning Authority shall perform research to determine the scope and extent of the problem and educate the involved parties as to the nature and extent of the problem. This shall include technical and financial

aspects of the dispute, including assistance in help identifying who the responsible parties are to implement corrective action. The Owner shall preside over resolution of the problem.

7. Deficiencies: The Commissioning Authority shall provide technical expertise to oversee and verify the correction of deficiencies found during the commissioning process.
8. Acceptance: The Commissioning Authority shall determine and advise the Construction Manager and Owner of the date of acceptance for each component and system for start of the warranty period.
9. Operations and Maintenance (O&M) manuals: The Commissioning Authority shall provide technical expertise to review and edit the system's operating and maintenance manuals
10. Training: The Commissioning Authority shall schedule and coordinate training sessions for the Owner's operations and maintenance staff for each system.

B. The Commissioning Authority is referred to as an independent contractor in this Division and shall work under a separate contract directly for the Owner.

C. The Commissioning Authority shall not be financially associated with any of the Division 2 through 16 contractors on this project to avoid conflicts of interest.

1.03 Systems To Be Commissioned

A. The following pieces of equipment and systems shall go through commissioning:

1. Chilled water system
2. Condenser water system
3. Hydronic distribution systems
4. Variable frequency drives
5. Motor control centers
6. DDC/EMCS control systems, hardware, software, and documentation

1.04 Coordination

A. The Construction Manager shall be responsible for furnishing a copy of all construction documents, addenda, change orders, and appropriate approved submittals and shop drawings to the Commissioning Authority.

B. The Commissioning Authority shall coordinate directly with each contractor on the project specific to their responsibilities and contractual obligations. If the contractor is a subcontractor to another contractor, the Commissioning Authority shall disseminate written information to all responsible parties relative to the nature and extent of the communication.

C. The Commissioning Authority is primarily responsible to the Owner through the Construction Manager, and as such, shall regularly apprise the Construction Manager and the Owner of progress, including pending problems and/or disputes, and shall provide regular status reports on progress with each system. Any potential change in the contractual and/or financial obligations of the Owner (credits, change orders, schedule changes, etc.) shall be identified and quantified as soon as possible.

1.05 Schedule

A. Commissioning of systems shall proceed per the criteria established in the specific sections that follow, with activities to be performed on a timely basis. The Commissioning Authority shall be available to respond promptly to avoid construction delays.

B. Start-up and testing of systems may proceed prior to final completion of systems to expedite progress. However, the Commissioning Authority shall not perform testing and checkout services that are the primary responsibility of the contractor or vendor in advance of their testing and checkout.

C. Problems observed shall be addressed immediately, responsible parties notified, and actions to correct deficiencies coordinated in a timely manner.

D. Contractor schedules and scheduling is the responsibility of the Construction Manager. The Commissioning Authority shall provide commissioning scheduling information to the Construction Manager for review and planning activities.

1.06 Related Work Specified Elsewhere

A. Commissioning requires support from the contractors. The commissioning process does not relieve any contractors from their obligations to complete all portions of work in a satisfactory manner.

B. Refer to Sections 15995 and 16970 for Division 15 and 16 contractor responsibilities relative to the commissioning process.

Part 2 - Products

2.01 Test Equipment

A. All industry-standard test equipment required for performing the specified tests shall be provided by the Commissioning Authority. Any proprietary vendor-specific test equipment shall be provided by that vendor or manufacturer.

B. Any portable or hand-held setup or calibration devices required to initialize the control system shall be made available by the control vendor (at no cost) to the Commissioning Authority.

C. The Commissioning Authority's instrumentation shall meet the following standards:

1. Be of sufficient quality and accuracy to test and/or measure system performance within the tolerances required.
2. Be calibrated at the manufacturer's recommended intervals with calibration tags permanently affixed to the instrument
3. Be maintained in good repair and operating condition throughout the duration of use on this project.
4. Be immediately re-calibrated or repaired if dropped and/or damaged in any way during use on this project.

Part 3 - Execution

3.01 Commissioning Plan and Schedule

A. The Commissioning Authority shall develop and submit a schedule for the commissioning process that is integrated with the construction schedule. Included shall be the required work by all team members (Commissioning Authority, design team, contractors, and the Owner). This schedule shall include time for test and balance, verification, and functional performance testing.

3.02 Construction Observation

A. This is an additional and separate activity from that provided by the design team. Construction observation is required as part of the commissioning and coordination process to be provided by the Commissioning Authority.

3.03 Test And Balance

A. Air and water balance and equipment performance verification shall be accomplished by an independent test and balance firm. The Commissioning Authority shall spot check this work to verify accuracy of results.

3.04 Verification and Functional Performance Test Procedures

A. Personnel experienced in the technical aspects of each system to be commissioned shall develop and document the commissioning procedure to be used. Procedures shall include a performance checklist and performance test data sheets for each system based on actual system configuration. These procedures shall be reviewed by the appropriate design engineers for technical depth, clarity of documentation and completeness. Special emphasis shall be placed on testing procedures that shall conclusively determine actual system performance and compliance with the design intent.

B. The majority of mechanical equipment requires safety devices to stop and/or prevent equipment operation unless minimum safety standards or conditions are met. These may include adequate oil pressure, proof-of-flow, non-freezing conditions, maximum static pressure, maximum head pressure, etc. The Commissioning Authority shall observe the actual performance of safety shutoffs in a real or closely simulated condition of failure.

C. Systems may include safety devices and components that control a variety of equipment operating as a system. Interlocks may be hard-wired or operate from software. The Commissioning Authority shall verify operation of these interlocks.

D. The Commissioning Authority shall determine the acceptance procedures for each system within Divisions 15 disciplines. The acceptance procedures shall incorporate the commissioning standards and successful testing results as referred to throughout Division 15 and 16 specifications.

   1. In particular, the temperature control system shall have all I/O points individually verified for proper function, calibration, and operation. The Commissioning Authority shall review proposed testing procedures and report formats, and observe sufficient field testing to confirm that all I/O points have been properly tested.

2. All control sequences of operation strategies, alarm generation and reporting shall also be reviewed and proper operation verified by the Commissioning Authority.

3. The central work station graphics, point assignments, alarm messages, and logging functions shall be verified.

E. The appropriate contractor and vendor(s) shall be informed of what tests are to be performed and the expected results. Whereas some test results and interpretations may not become evident until the actual tests are performed, all parties shall have a reasonable understanding of the requirements. The commissioning plan shall address those requirements and be distributed to all parties involved with that particular system.

F. Acceptance procedures shall confirm the performance of systems to the extent of the design intent. When a system is accepted, the Owner shall be assured that the system is complete, works as intended, is correctly documented, and operator training has been performed.

3.05 Software Documentation Review

A. The Commissioning Authority shall review detailed software documentation for all DDC control systems. This includes review of vendor documentation, their programming approach, and the specific software routines applied to this project. Discrepancies in programming approaches and/or sequences shall be reported and coordinated in order to provide the Owner with the most appropriate, simple, and straightforward approach to software routines.

3.06 Operating and Maintenance (O&M) Manuals

A. The Commissioning Authority shall review the draft form of the O&M manuals provided by the Division 15 contractor. The review process shall verify that O&M instructions meet specifications and are included for all equipment furnished by the contractor, and that the instructions and wiring diagrams are specific (edited where necessary) to the actual equipment provide for this project.

Published literature shall be specifically oriented to the provided equipment and shall indicate required operation and maintenance procedures, parts lists, assembly/disassembly diagrams, and related information.

The contractor shall incorporate the standard technical literature into system-specific formats for this facility as designed and as actually installed. The resulting O&M information shall be system specific, concise, to the point, and tailored specifically to this facility. The Commissioning Authority shall review and edit these documents as necessary for final corrections by the contractor.

B. The O&M manual shall be reviewed, edited and finalized prior to Owner training sessions, as these documents are to be utilized in the training sessions.

3.07 Training

The Commissioning Authority shall schedule and coordinate training sessions for the Owner's staff for each system. Training sessions shall be in both a classroom setting with the appropriate schematics, handouts, and audio/visual training aids, and on-site with the installed equipment. The training sessions shall be videotaped for future training needs.

A. The Commissioning Authority shall host each training session with program overview and curriculum guidance.

B. The appropriate installing contractors shall provide training on all the major systems per specifications, including peculiarities specific to this project.

C. The equipment vendors shall provide training on the specifics of each major equipment item including philosophy, troubleshooting, and repair techniques.

D. The automatic control and fire alarm vendors shall provide training on the control system and fire alarm system per their specification section.

3.08 Record Drawings

A. The Commissioning Authority shall review the as-built contract documents to verify incorporation of both design changes and as-built construction details. Discrepancies noted shall be corrected by the appropriate party.

3.09 Exclusions

A. Responsibility for construction means and methods: The Commissioning Authority is not responsible for construction means, methods, job safety, or any construction management functions on the job site.

B. Hands-on work by the Commissioning Authority: The contractors shall provide all services requiring tools or the use of tools to start-up, test, adjust, or otherwise bring equipment and systems into a fully operational state. The Commissioning Authority shall coordinate and observe these procedures (and may make minor adjustments), but shall not perform construction or technician services other than verification of testing, adjusting, balancing, and control functions.

Part 1 - General

1.01 Definition

A. The purpose of this section is to specify the Division 15 responsibilities and participation in the commissioning process.

B. Work under this contract shall conform to requirements of Division 1, General Requirements, Conditions of the Contract, and Supplementary Conditions. This specification covers commissioning of mechanical systems which are part of this project.

C. Commissioning work shall be a team effort to ensure that all mechanical equipment and systems have been completely and properly installed, function together correctly to meet the design intent, and document system performance parameters for fine tuning of control sequences and operational procedures. Commissioning shall coordinate system documentation, equipment start-up, control system calibration, testing and balancing, and verification and performance testing.

D. The commissioning team shall be made up of representatives from the owner, design professionals, major equipment suppliers, and construction trades. The trades represented on the commissioning team shall include, but not be limited to, piping and fitting, controls, test and balance, and electrical. The lead person for each trade who will actually perform or supervise the work is to be designated as the representative to the commissioning team. Responsibility for various steps of the commissioning process shall be divided among the members of the commissioning team, as described in this section.

E. The Commissioning Authority shall have responsibility for coordinating and directing each step of the commissioning process.

F. Mechanical system installation, start-up, testing, balancing, preparation of O&M manuals, and operator training are the responsibility of the Division 15 Contractors, with coordination, observation, verification and commissioning the responsibility of Division 1, Section 01650. The 01650 commissioning process does not relieve Division 15 from the obligations to complete all portions of work in a satisfactory and fully operational manner.

1.02 Scope of Work

A. Commissioning work of Division 15 shall include, but not be limited to:

1. Testing and start-up of the equipment.
2. Testing, adjusting and balancing of hydronic and air systems.
3. Cooperation with the Commissioning Authority.
4. Providing qualified personnel for participation in commissioning tests, including any seasonal testing required after the initial testing.
5. Providing equipment, materials, and labor as necessary to correct construction and/or equipment deficiencies found during the commissioning process.

6. Providing operation and maintenance manuals, and as-built drawings to the Commissioning Authority for verification.

7. Providing training and demonstrations for the systems specified in this Division.

B. The work included in the commissioning process involves a complete and thorough evaluation of the operation and performance of all components, systems, and sub-systems. The following equipment and systems shall be evaluated:

1. Chilled water system

2. Condenser water system

3. Hydronic distribution systems

4. Variable frequency drives

5. DDC/EMCS control systems

C. Timely and accurate documentation is essential for the commissioning process to be effective. Documentation required as part of the commissioning process shall include but not be limited to:

1. Progress and status reports, including deficiencies noted.

2. Minutes from all meetings.

3. Pre-start, and start-up procedures.

4. Training agenda and materials.

5. As-built records.

6. Piping pressure test report.

7. Operation and maintenance (O&M) manuals.

D. Detailed testing shall be performed on all installed equipment and systems to ensure that operation and performance conform to contract documents. All tests shall be directed by the Commissioning Authority. The following testing is required as part of the commissioning process:

1. Verification tests are comprised of a full range of checks and tests to determine that all components, equipment, systems, and interfaces between systems operate in accordance with contract documents. This includes all operating modes, interlocks, control responses, and specific responses to abnormal or emergency conditions.

2. Functional performance tests (FPT) shall determine if the chilled water system is providing the required cooling services in accordance with the finalized design intent. These tests shall also determine the installed capacity of the cooling plant, and the individual heat transfer components.

E. Comprehensive training of O&M personnel shall be performed by the Mechanical Contractor, and where appropriate, by other sub-contractors, and vendors prior to turnover of building to the owner. The training shall include classroom instruction, along with hands-on instruction on the installed equipment and systems. The training sessions shall be videotaped for future reference.

1.03 Roles and Responsibilities

All parties involved in the construction process shall be involved in the commissioning process. Following is a description of the responsibilities of each party:

A. Owner

1. Assign maintenance personnel and schedule them to participate in meetings, training sessions and inspections as follows:

    a. Construction Phase coordination meeting.

    b. Initial Owner training session at initial placement of major equipment.

    c. Maintenance orientation and inspection.

    d. Piping test and flushing verification.

    e. Procedures meeting for Testing, Adjusting and Balancing.

    f. Owner training session.

    g. Verification demonstrations.

2. Provide qualified personnel for videotaping and editing of training sessions.

3. Videotape construction progress, hidden shafts, etc. (optional).

4. Provide any utilities required for the commissioning process.

B. Commissioning Authority

1. Develop the commissioning requirements and all related testing, verification and quality control procedures.

2. Prepare the mechanical commissioning program required as part of the Commissioning Specification. Include list of all contractors for commissioning events by name, firm and trade specialty.

3. Execute the mechanical commissioning program, through organization of all meetings, tests, demonstrations, training events and performance verifications described in the Contract Documents and approved mechanical commissioning program. Organizational responsibilities include preparation of agendas, attendance lists, arrangements for facilities and timely notification to participants for each commissioning event. The Commissioning Authority shall act as chairman at all commissioning events and assure execution of all agenda items. The Commissioning Authority shall prepare minutes of every commissioning meeting and send copies to all attendees and the Owner.

4. Review the project plans and specifications with respect to their completeness in all areas relating to the mechanical commissioning program. This includes ensuring that appropriate commissioning guidelines have been followed, and that there are adequate devices included in the design to ensure the ability to properly test, adjust, and balance the systems, and to document the performance of each piece of equipment and each system. Any items required but not shown shall be brought to the attention of the Contractor prior to submittal of shop drawings.

5. Schedule the Construction Phase coordination meeting within 90 days of the award of the contract, at some convenient location and at a time suitable to the Contractor and Design Professional. This meeting shall be for the purpose of reviewing the complete mechanical commissioning program and establishing tentative schedules for mechanical system orientation and inspections, O&M submittals, training sessions, system flushing and testing, job completion, test, adjust and balance (TAB) work, and verification and functional performance testing.
6. Conduct periodic inspections of work in progress to ensure that all systems and equipment are installed according to specifications.
7. Receive and review the Operation and Maintenance (O&M) manuals as submitted by the contractor, ensuring that they follow the specified outline and format. Insert systems description as provided by the Design Professional.
8. Adequate accessibility for maintenance and component replacement or repair is the Construction Manager's responsibility and shall be checked by the Commissioning Authority.
9. Witness equipment and system start-up and testing. Ensure the results are documented (including a summary of deficiencies), and incorporated in the O&M manuals.
10. Prior to initiating the TAB work, the Commissioning Authority shall meet with the Owner, mechanical Contractor, Design Professional and TAB Contractor. The TAB Contractor shall outline TAB procedures and get concurrence from the Design Professional and Commissioning Authority.
11. Schedule the O&M training sessions. The format shall follow the outline in the O&M manuals. This mechanical system training should include hands on training.
12. Submit detailed verification test procedures and data sheets for review by the Design Professional.
13. Submit detailed functional performance test procedures for review and acceptance by the Design Professional.
14. Upon receipt of notification from the Construction Manager that the mechanical systems have been completed and are operational, the Commissioning Authority shall proceed to verify the TAB report and performance of the control systems in accordance with the Commissioning Specification. The Commissioning Authority shall certify that the mechanical systems are ready for verification testing, and functional performance testing.
15. Conduct verification tests.
16. Submit verification test data to the Design Professional for review and acceptance.
17. Provide detailed checklist data sheets to document verification tests.
18. Provide and install calibrated test instrumentation to monitor and record data as necessary.
19. Conduct functional performance tests. The test data shall be part of the commissioning report.

20. Submit functional performance test report for review to the Design Professional.
21. Re-test if performance deficiencies are found, corrected, and additional testing is requested.
22. Review as-built drawings for accuracy with respect to the installed systems. Request revisions to achieve accuracy.
23. Ensure that the O&M manuals, and all other as-built records have been updated to include all modifications made during the construction phase.
24. Prepare the final commissioning report.

C. Architect

1. Provide support to the Design Professional who must provide a service as a part of the commissioning process. This shall include providing adequate space for equipment installation and maintenance.
2. Include Section 01650 regarding commissioning in Division 1 - General Requirements alerting all parties to the need to participate.
3. Conduct periodic inspections of work in progress to ensure that all systems and equipment are installed according to specifications.

D. Mechanical Design Professional

1. Provide documentation of initial design concepts, and Design Intent.
2. Provide mechanical system design parameters and obtain approval of Owner.
3. Prepare contract documents incorporating the Commissioning Specification requirements, and description of the mechanical system.
4. The Design Professional shall specify and verify adequate maintenance accessibility for each piece of equipment in shop drawings and the actual installation. Periodic inspections are part of the Design Professional's normal construction administration duties.
5. The Design Professional retains responsibility for the system evaluation, adequacy of the system to meet design intent, capacity of the system, quality control check or any of the other elements of the system design.
6. Attend the Owner training sessions. Conduct the mechanical training session pertaining to the overview of the system design, the system design goals and the reasoning behind the selection of equipment.
7. Participate in O&M personnel orientation and inspection at the final construction stage.
8. Attend initial meeting with TAB representative as scheduled by Commissioning Authority.
9. Review verification and functional performance testing procedures submitted by the Commissioning Authority.

10. Review TAB report and verification data sheets for system conformance to contract documents. Issue a report noting deficiencies requiring correction to the Commissioning Authority.
11. Review functional performance testing report for deficiencies in meeting the finalized design intent.
12. Review as-built records as required by contract documents.
13. Review and comment on the final commissioning report.

E. Construction Manager

1. Include cost for commissioning requirements in the contract price.
2. Include commissioning requirements in the mechanical, electrical, and controls contracts, as well as all other sub-contractors, to ensure full cooperation of all parties in the mechanical commissioning program.
3. Ensure acceptable representation, with the means and authority to prepare and coordinate execution of the mechanical commissioning program as described in the contract documents.
4. Remedy deficiencies identified in verification tests.
5. Evaluate any performance deficiencies identified in the FPT report for non-conformance with contract documents.

F. Mechanical Contractor

1. Include cost to complete commissioning requirements for mechanical systems in the contract price.
2. Include requirements for submittal data, O & M data, and training in each purchase order or sub-contract written.
3. Ensure cooperation and participation of specialty sub-contractors such as sheet metal, piping, refrigeration, water treatment, and TAB.
4. Ensure participation of major equipment manufacturers in appropriate training and testing activities.
5. Attend coordination meetings scheduled by the Commissioning Authority.
6. Assist the Commissioning Authority in all verification and functional performance tests.
7. Prepare preliminary schedule for mechanical system orientation and inspections, O&M manual submission, training sessions, pipe and duct system testing, flushing and cleaning, equipment start-up, TAB, and task completion for use by the Commissioning Authority. Update schedule as appropriate throughout the construction period.
8. Conduct mechanical system orientation and inspection at the equipment placement completion stage.

9. Update drawings to the record condition to date, and review with the Commissioning Authority.

10. Gather O&M data on all equipment, and assemble in binders as required by the Commissioning Specification. Submit to Commissioning Authority prior to the completion of construction.

11. Notify the Commissioning Authority a minimum of two weeks in advance, so that witnessing equipment and system start-up and testing can begin.

12. Notify the Commissioning Authority a minimum of two weeks in advance, of the time for start of the TAB work. Attend the initial TAB meeting for review of the TAB procedures.

13. Participate in, and schedule vendors and Contractors to participate in the training sessions as set up by the Commissioning Authority.

14. Provide written notification to the Construction Manager and Commissioning Authority that the work has been completed in accordance with the contract documents, and that the equipment, systems, and sub-systems are functioning as required.

15. Provide a complete set of as-built records to the Commissioning Authority.

G. Test, Adjust, and Balance Contractor

1. Include cost for commissioning requirements in the contract price.

2. Attend commissioning coordination meetings scheduled by the Commissioning Authority.

3. Submit the TAB procedures to the Commissioning Authority and Design Professional for review and acceptance.

4. Attend the TAB review meeting scheduled by the Commissioning Authority. Be prepared to discuss the procedures that shall be followed in testing, adjusting and balancing the HVAC system.

5. Participate in training sessions as scheduled by the Commissioning Authority.

6. At the completion of the TAB work, and submittal of final TAB report, notify the mechanical Contractor and Construction Manager.

7. Participate in verification of the TAB report, which will consist of repeating any selected measurement contained in the TAB report where required by the Commissioning Authority for verification or diagnostic purposes.

H. Automatic Controls and Building Automation System Contractors

1. Include cost for commissioning requirements in the contract price.

2. Review design for controllability with respect to selected manufacturers' equipment;

   a. Verify proper hardware specification exists for functional performance required by specification and sequence of operation.

b. Verify proper safeties and interlocks are included in design.

c. Verify proper sizing of control valves and actuators based on design pressure drops. Verify control valve authority to control coil properly.

d. Verify proper sizing of control dampers. Verify damper authority to control air stream. Verify proper damper positioning for mixing to prevent stratification. Verify actuator vs. damper sections for smooth operation.

e. Verify proper selection of sensor ranges.

f. Clarify all questions of operation.

3. Attend initial commissioning coordination meeting scheduled by the Commissioning Authority.

4. Provide the following submittals to the Commissioning Authority;

   a. Hardware and software submittals.

   b. Control panel construction shop drawings.

   c. Narrative description of each control sequence for each piece of equipment controlled.

   d. Diagrams showing all control points, sensor locations, point names, actuators, controllers and, where necessary, points of access, superimposed on diagrams of the physical equipment.

   e. Logic diagrams showing the logic flow of the system.

   f. A list of all control points, including analog inputs, analog outputs, digital inputs, and digital outputs. Include the values of all parameters for each system point. Provide a separate list for each stand-alone control unit.

   g. A complete control language program listing including all software routines employed in operating the control system. Also provide a program write-up, organized in the same manner as the control software. This narrative shall describe the logic flow of the software and the functions of each routine and sub-routine. It should also explain individual math or logic operations that are not clear from reading the software listing.

   h. Hardware operation and maintenance manuals.

   i. Application software and project applications code manuals.

5. Verify proper installation and performance of controls / BAS hardware and software provided by others.

6. Integrate installation and programming schedule with construction and commissioning schedules.

7. Provide thorough training to operating personnel on hardware operations and programming, and the application program for the system.

8. Demonstrate system performance to Commissioning Authority.

9. Provide control system technician for use during system verification and functional performance testing.
10. Provide system modifications as required.

I. Equipment Suppliers and Miscellaneous Contractors

1. Include cost for commissioning requirements in the contract price.
2. Provide submittals, and appropriate O&M manual section(s).
3. Participate in training sessions as scheduled by the Commissioning Authority.
4. Demonstrate performance of equipment as applicable.

1.04 Documentation

A. The Commissioning Authority shall oversee and maintain the development of commissioning documentation. The commissioning documentation shall be kept in three ring binders, and organized by system and sub-system when practical. All pages shall be numbered, and a table of contents page(s) shall be provided. The commissioning documentation shall include, but not be limited to, the following:

1. The final commissioning plan.
2. A detailed description of the design intent for the project, listing operating parameters, control sequences, occupancy conditions, etc.
3. Approved test and balance report for the building being commissioned.
4. Minutes from all commissioning meetings.
5. All pre-functional performance test checklists, signed by indicated personnel, organized by system and sub-system.
6. All verification and functional performance test checklists/results, signed by indicated personnel, organized by system and sub-system.

Part 2 - Products

2.01 Test Equipment

A. The Commissioning Authority shall furnish test equipment required during the commissioning process that is not part of the mechanical control system. The owner shall furnish necessary utilities for the commissioning process.

B. Test equipment, accuracy. All instruments must be calibrated prior to tests.

1. Voltmeter, ± 1% scale.
2. Ammeter, ± 1% scale.
3. Ohmmeter, ± 0.1% scale for calibrating ±0.4°F resistance temperature sensors, ± 0.25% scale for calibrating ±1°F temperature sensors, ± 1% scale for measuring motor current.
4. Ultrasonic time-of-travel strap-on flow sensor, ± 5% of reading.
5. Other flow sensors, ± 2% of reading.

6. Water pressure gauge, ±1/2% scale, ASME Grade 2A
7. Watt meter, ±1/2% scale, 3 phase split core CTs
8. Refrigerant pressure gauge, as supplied by chiller manufacturer.
9. Temperature, ± 0.2°F.

2.01 Test Equipment - Proprietary

A. Proprietary test equipment and software required by any equipment manufacturer for programming and/or start-up, whether specified or not, shall be provided by the manufacturer of the equipment. Manufacturer shall provide the test equipment, demonstrate its use, and assist in the commissioning process as needed. Proprietary test equipment (and software) shall become the property of the owner upon completion of the commissioning process.

Part 3 - Execution

3.01 General

A. A pre-construction meeting of all commissioning team members shall be held at a time and place designated by the owner. The purpose shall be to familiarize all parties with the commissioning process, and to ensure that the responsibilities of each party are clearly understood.

B. The Contractor shall complete all phases of work so the systems can be started, tested, balanced, and commissioning procedures undertaken. This includes the complete installation of all equipment, materials, pipe, duct, wire, insulation, controls, etc., per the contract documents and related directives, clarifications, and change orders.

C. A commissioning plan shall be developed by the Commissioning Authority. The Contractor shall assist the Commissioning Authority in preparing the commissioning plan by providing all necessary information pertaining to the actual equipment and installation. If Contractor-initiated system changes have been made that alter the commissioning process, the Commissioning Authority shall notify the Owner.

D. Commissioning is normally intended to begin prior to completion of a system and/or sub-systems, and shall be coordinated with the Division 15 contractor. Start of commissioning before system completion does not relieve the contractor from completing those systems as per the schedule.

3.02 Participation in Commissioning

A. The Contractor shall provide skilled technicians to start-up and debug all systems within Division 15. These same technicians shall be made available to assist the Commissioning Authority in completing the commissioning program. Work schedules, time required for testing, etc., shall be requested by the Commissioning Authority and coordinated by the contractor. Contractor shall ensure that the qualified technician(s) are available and present during the agreed upon schedules and of sufficient duration to complete the necessary tests, adjustments, and/or problem resolutions.

B. System performance problems and discrepancies may require additional technician time, Commissioning Authority time, reconstruction of systems, and/or replacement of system components. The additional technician time shall be made available for subsequent commissioning periods until the required system performance is obtained.

C. The Commissioning Authority reserves the right to question the appropriateness and qualifications of the technicians relative to each item of equipment, system, and/or sub-system. Qualifications of technicians shall include expert knowledge relative to the specific equipment involved and a willingness to work with the Commissioning Authority. Contractor shall provide adequate documentation and tools to start-up and test the equipment, system, and/or sub-system.

3.03 Deficiency Resolution

A. In some systems, misadjustments, misapplied equipment, and/or deficient performance under varying loads will result in additional work being required to commission the systems. This work shall be completed under the direction of the Owner, with input from the contractor, equipment supplier, and Commissioning Authority. Whereas all members shall have input and the opportunity to discuss, debate, and work out problems, the Owner and/or Design Professional shall have final jurisdiction over any additional work done to achieve performance.

B. Corrective work shall be completed in a timely fashion to permit the completion of the commissioning process. Experimentation to demonstrate system performance may be permitted. If the Commissioning Authority deems the experimentation work to be ineffective or untimely as it relates to the commissioning process, the Commissioning Authority shall notify the Owner, indicating the nature of the problem, expected steps to be taken, and suggested deadline(s) for completion of activities. If the deadline(s) pass without resolution of the problem, the Owner reserves the right to obtain supplementary services and/or equipment to resolve the problem. Costs incurred to solve the problems in an expeditious manner shall be the contractor's responsibility.

3.04 Additional Commissioning

A. Additional commissioning activities may be required after system adjustments, replacements, etc., are completed. The contractor(s), suppliers, and Commissioning Authority shall include a reasonable reserve to complete this work as part of their contractual obligations.

3.05 Acceptance Procedures

Start-up

A. Prerequisites. The following work shall be completed before start-up

   1. Pre-start inspection.

B. Procedure.

   1. The installing contractor shall be responsible for scheduling and executing start-up. Start-up is best scheduled as soon as possible after substantial completion of the system or subsystem but before balancing work is initiated. The start-up schedule shall be submitted to the Commissioning Authority for review.

2. Chillers shall be started by manufacturer's representatives specifically trained for this work on the particular chiller type installed, unless otherwise specified in the Chiller specification section.

3. Variable speed drives shall be started by manufacturer's representatives, unless otherwise specified in the variable speed drive specification section.

4. The contractor's start-up technician or designated subcontractor or manufacturer's representative shall complete the start-up forms as equipment is available per the start-up schedule. If there is a discrepancy between the performance of equipment and that required, the required corrective action shall be taken by the contractor or his representative. The start-up technician who completes the form shall sign the form when it has been completed.

5. Completed forms shall be submitted to the Commissioning Authority for review and to demonstrate that the work has been completed.

C. Digital control system start-up:

1. Record all tests and file with other start-up documentation forms.

2. Sensor test and calibration:

3. All chilled and condenser water temperature sensors shall be tested and calibrated as required. Factory calibration alone is not sufficient.

4. For sensors with shielded cable, inspect to be sure each shield is grounded only at one end.

5. Follow the procedure below to test and calibrate RTDs, thermistors, and other variable resistance sensors that are directly connected to the DDC system without a transmitter:

6. Disconnect sensing element from loop.

7. Check for a failed signal at DDC system. If system shows a failure, proceed to next step. If not, check for shorts or bad point definition.

8. Connect decade box or other suitable resistance simulation device in place of sensor.

9. Referring to sensor manufacturer's resistance-temperature data, simulate minimum and maximum of desired temperature range.

10. Example: a platinum RTD temperature sensor is 1000 ohms at 70°F with a desired range of 40°F to 240°F and resistance change of 2.2 ohm/°F. The resistance at lower end of range 40°F is [1000 + (40-70)*2.2] = 934 ohms. At 240°F, the resistance is [1000 + (240-70)*2.2] = 1374 ohms. This data could also have been determined from a chart if manufacturer provides one.

11. Check DDC for proper readings. If readings fail to correspond to simulated temperatures, adjust DDC calibration constants (e.g. slope and intercept) as required.

12 Repeat previous step until readings correspond.

13. Reconnect sensor. Allow to reach steady state. Measure temperature with hand-held temperature sensor. Hand-held reading should match DDC reading within accuracy of sensors. If not, replace sensor with new one.

    a. Follow the procedure below to test and calibrate RTDs, thermistors, and other variable resistance sensors that are connected to the DDC system through a transmitter:

14. Disconnect sensing element from loop.

15. Check for a failed signal at DDC system. If system shows a failure, proceed to next step. If not, check for shorts or bad point definition.

16. Connect decade box or other suitable resistance simulation device to the transmitter in place of sensor.

17. Connect an ammeter in series with the signal from the transmitter to the DDC system.

18. Referring to sensor manufacturer's resistance-temperature data, simulate minimum of desired temperature range. Adjust the transmitter zero potentiometer until 4 mA is read by the ammeter.

19. Check DDC for proper readings. Adjust DDC calibration constant (e.g. value at 4 mA) as required to correspond to low end of desired range.

20. Referring to sensor manufacturer's resistance-temperature data, simulate maximum of desired temperature range. Adjust the transmitter span potentiometer until 20 mA is read by the ammeter.

21. Check DDC for proper readings. Adjust DDC calibration constant (e.g. value at 20 mA) as required to correspond to high end of desired range.

22. Reconnect sensor. Allow to reach steady state. Measure temperature with hand-held temperature sensor. Hand-held reading should match DDC reading within accuracy of sensors. If not, replace sensor with new one.

    a. For sensors used to calculate differential temperature (e.g. chilled water supply and return for load calculation), place sensors into a bath of water at a temperature near the expected operating water temperatures. Verify that readings are identical at DDC system. Use hand-held sensor to measure bath as well. If DDC sensors are not equal, recalibrate the one that is furthest from the hand-held reading to match the other.

    b. DDC pressure sensors shall be calibrated to match reading of hand-held gauge when pumps are operating and pressures are within normal operating range.

23. Control valve stroke test

    a. During control valve tests, pumps must be placed in normal operating mode to ensure that valves can perform against normal back pressure.

    b. For each valve:

        i. Command valve closed through DDC system, verify position, and adjust output zero signal as required.

ii. Command valve open, verify position, and adjust output signal span as required.

iii. For analog outputs, command output to 25%, 50%, and 75% of range and observe actual performance. Be sure pumps are running per normal operation. If actuator position does not reasonably correspond to commanded position, replace actuator or (for pneumatic valves) add pilot positioner.

iv. Remove output signal and power source from valve. Observe valve return to normal position. Record and verify position is as shown on control drawings.

Verification Tests

A. Scope of Verification Tests

1. Operating tests and checks to verify that all components, equipment, systems, sub-systems, and interfaces between systems, operate in accordance with contract documents. These tests are to include all operating modes, interlocks, specified control responses, specific responses to abnormal or emergency conditions, and verification of the proper response of the building automation system controllers and sensors.

2. Verify the validity of the TAB report.

B. Participants in Verification Tests

The Commissioning Authority shall be responsible for preparing the scope of these tests. The Commissioning Authority shall schedule the tests and assemble the commissioning team members who shall be responsible for the tests. Participating contractors, manufacturers, suppliers, etc. shall include all costs to do the work involved in these tests in their proposals. Following is a list of tasks and supporting information that shall be required:

1. Mechanical contractor - provide the services of a technician(s) who is (are) familiar with the construction and operation of this system. Provide access to the contract plans, shop drawings, and equipment cut sheets of all installed equipment.

2. Controls contractor - provide the services of a controls engineer who is familiar with the details of the project. Provide details of the control system, schematics, and a narrative description of control sequences of operation.

3. Electrical contractor - provide a foreman electrician familiar with the electrical interlocks, interfaces with emergency power supply, and interfaces with alarm and life-safety systems. Provide access to the contract plans, and all as-built schematics of sub-systems, interfaces, and interlocks.

C. Documentation and Reporting Requirements

1. Provide checklists for each component, piece of equipment, system, and sub-system, including all interfaces, interlocks, etc. Each item to be tested shall have a different entry line with space provided for comments. Separate checklists shall

be prepared for each mode of operation. Provide space to indicate whether the mode under test responded as required or not. Also, provide space for all necessary parties to sign off on each checklist.

2. Data sheets used in verification of the proper operation of the control system shall include each controller to be verified, the system it serves, the service it provides, and it's location. For each controller, provide space for recording the readout of the controller, the reading at the controller's sensor(s), and any comments. Also, provide space for all necessary parties to sign off on each checklist.

3. All test procedures and data sheets shall be submitted to the design professional for review and acceptance.

D. Instrumentation

1. The Commissioning Authority shall furnish all measurement instrumentation for the verification tests. All instruments will have been calibrated within the manufacturer's recommended period prior to these tests.

E. Verification Procedures

1. The Commissioning Authority shall direct and witness the verification operating tests and checks for all equipment and systems.

a. Set the system equipment into the operating mode to be tested, i.e. normal shut down, normal auto position, normal manual position, unoccupied cycle, emergency power, and alarm conditions.

b. The Commissioning Authority shall inspect and verify the position of each device and interlock identified on the checklist. Each item shall be signed off as acceptable (yes), or failed (no).

c. This test shall be repeated for each operating cycle that applies to the mechanical system being tested.

d. Operating checks shall include all safety cutouts, alarms, and interlocks with smoke control and life safety systems during all modes of operation of the mechanical system.

e. If during a test an operating deficiency is observed, appropriate comments shall be added to the checklist data sheet.

f. Verification of the proper responses of monitoring and control system controllers and sensors shall be as follows:

i. For each controller or sensor, record the indicated monitoring and control system reading, and the test instrument reading.

ii. If the initial test indicates that the test reading is outside of the control range of the installed device, the calibration of the installed device shall be checked and adjusted as required. The deficient device shall be re-tested and the results recorded on the checklist data sheet.

g. False loading.

i. In most installations, verification tests are made just after construction and prior to occupancy. Weather conditions during the tests will vary depending on the local climate and time of year. Therefore, there may not be sufficient load to test equipment, so false loading may be required. Means to provide false loading will vary with system type. Some possible measures include:

   a) Using preheat coils at inlets to air handlers. This is an ideal location for false loading since conditions in the spaces served by the air handlers will not be affected.

   b) Using reheat coils at zones. This may not always be practical if spaces served are occupied since overheating is possible.

   c) Opening outdoor air dampers if the weather is warm during tests.

   d) Closing outdoor air dampers if the weather is cold during tests.

   e) Bleeding hot water into the chilled water system if coil changeover valves are present. (Care must be used to ensure that very high chilled water entering temperatures do not result).

ii. To control false loading, temporary control loops may be added to the DDC system to modulate hot water valves or other parameter to maintain a given return water temperature or load.

iii. False loading may not always be practical with some system types. If so, tests may have to be postponed until occupancy or until the weather is warm. It is not recommended that hot and chilled water piping systems be cross-connected purely for the purpose of chiller testing due to the expense and disruption involved.

h. The Commissioning Authority shall direct and witness the field verification of the final TAB report.

   i. The TAB contractor shall be given sufficient advance notice of the date of field verification. However, they shall not be informed in advance of the data points to be verified. The TAB contractor must use the same instruments (by model and serial number) that were used when the original data were collected.

   ii. Failure of an item is defined as:

      a) For all readings other than sound, a deviation of more than 10 percent.

      b) For sound pressure readings, a deviation of 3 decibels. (Note: variations in background noise must be considered).

   iii. A failure of more than 10 percent of the selected items shall result in the rejection of the final TAB report.

i. If deficiencies are identified during verification, the construction manager must be notified, and action taken to remedy the deficiency. The final tabulated checklist data sheets shall be reviewed by the Design Professional and the

Commissioning Authority, to determine if verification is complete, and the operating system is functioning in accordance with the contract documents.

Functional Performance Testing

A. Scope of Functional Performance Testing

1. Functional performance tests shall determine if the chilled water system is providing the required cooling services in accordance with the final design intent. They shall also determine the installed capacity of the cooling plant, and heat transfer components. Following is a list of test examples:

   a. Determine capability of chilled water system to deliver chilled water at the design supply temperature, and required rate of flow.

   b. Determine as-installed operating efficiency (kW/ton) of chillers.

   c. Determine the ability of the HVAC unit to deliver the cooling and/or heating services to the distribution system, at the design supply air temperature, required static pressure, and proper outside air ventilation rate.

B. Submittals

1. Detailed procedures for each series of tests shall be submitted by the Commissioning Authority for review and acceptance. The procedures shall include samples of the data sheets that will be part of the reports.

C. Participants in Functional Performance Tests

1. Participants in the functional performance tests shall be the same as those listed in the verification tests.

D. Instrumentation

1. In addition to the instrumentation requirements detailed under verification, the Commissioning Authority may need to provide data acquisition equipment to record data for the complete range of testing.

E. Functional Performance Test Procedures

1. The Commissioning Authority shall supervise and direct all functional performance tests.

2. Measurements will be required to allow for calculation of total capacity of the system for each mode of operation under test.

   a. Full load performance. Chiller capacity and efficiency at full load may be tested in the field by simulating loads. However, due to field conditions that might lead to variations from design conditions (e.g. variations in flow, fluctuating loads), costs may be reduced and accuracy increased by verifying chiller performance in the factory using a factory-witnessed test in lieu of in-situ testing. (This must be specified before putting chillers out for bids).

   b. Part-load performance. In order to have a benchmark against which actual chiller performance may be compared, predicted chiller performance at various part-load conditions must be obtained. Possible options include:

i. APLV. Application part-load value (APLV) is a system-specific part-load parameter for the chiller typically generated along with chiller selection and performance data from the manufacturer. It is computed as a weighted average of the chiller's energy performance at 4 part-load conditions. The 4 individual performance points are provided along with the other selection data in submittals. These 4 part-load conditions points can be simulated in the field and used for performance verification.

ii. Part-load profile. A large number of data of chiller performance at part-load and off-design operating conditions can be collected during testing. This data can be used to develop regression equations that can estimate chiller performance at a variety of part-load conditions. Various chiller operating conditions can be simulated, or the system operation may be simply observed, and the actual performance compared to the regression estimates.

iii. Minimum load operation. Stability at low load is important to minimize cycling losses and use of hot-gas-bypass (if chiller is so fitted). Therefore, a test at minimal load should be performed to verify the manufacturer's claims. The chiller should be able to operate at the claimed minimum load condition indefinitely.

F. Documentation and Reporting Requirements

1. All measured data and data sheets, as well as a comprehensive summary describing the operation of the chilled water system at the time of testing shall be submitted by the Commissioning Authority.

2. A preliminary functional performance test report shall be prepared by the Commissioning Authority and submitted to the Design Professional for review. Any identified deficiencies need to be evaluated by the Design Professional and Construction Manager to determine if they are part of the contractor's or sub-contractor's contractual obligations. Construction deficiencies shall be corrected by the responsible contractor(s), and the specific functional performance test repeated.

3.06 Operating and Maintenance Manual

A. The operating and maintenance manual shall consist of a sturdy binder with 8-1/2" X 11" sheets containing the following major sections:

1. System Descriptions:

a. Each major system shall be described, typewritten, in general terms, including major components, interconnections, theory of operation, theory of controls, unusual features and major safety precautions. This information should correlate with information provided in the manufacturers' instructions book. This section shall include, but not be limited to, the following data:

i. Detailed description of each system and each of its components showing piping, valves, controls, and other components, with diagrams and illustrations where applicable.

ii. Wiring and control diagrams with data to explain detailed operation and control of each component.

iii. Control sequences describing start-up, all modes of operation, and shut down.

iv. Corrected shop drawings.

v. Approved product data including all performance curves and rating data.

vi. Copies of approved certifications and laboratory test reports (where applicable).

vii. Copies of warranties.

b. System diagrams, described in 3.06 B2 following, shall be incorporated in the appropriate systems descriptions. These should be reduced in size or folded to usefully fit into the manual.

2. Operating Instructions:

a. Condensed, typewritten, suitable for posting, instructions shall be provided for each major piece of equipment. Where more than one (1) common unit is installed, one instruction is adequate. The instructions shall provide procedures for:

i. Starting up the equipment/system.

ii. Shutting down the equipment/system.

iii. Operating the equipment in emergency or unusual conditions.

iv. Safety precautions.

v. Trouble shooting suggestions.

vi. Other pertinent data applicable to the operation of particular systems or equipment.

b. The instructions shall be suitable for posting adjacent to the equipment concerned. The Contractor shall provide instructions for:

i. Chillers and peripherals

ii. Hydronic distribution systems, including pumps.

iii. DDC control systems.

iv. Variable frequency drive.

v. Other specialized or uncommon equipment.

3. Ongoing and Preventive Maintenance:

a. Condensed, typewritten procedures for recommended ongoing and preventive maintenance actions shall be provided for each category of equipment/system listed in 3.06 A2 above. This information shall include, but not be limited to the following:

i. Maintenance and overhaul instructions.

ii. Lubricating schedule including type, grade, temperature, and frequency range.

iii. Parts list, including source of supply and recommended spare parts.

iv. Name, address, and 24 hour telephone number of each subcontractor who installed equipment and systems, and local representative for each type of system.

v. Other pertinent data applicable to the maintenance of particular systems or equipment.

b. These recommended preventive maintenance actions shall be categorized by the following recommended frequencies:

i. Weekly

ii. Monthly

iii. Quarterly

iv. Semi-Annual

v. Annual

vi. Other

B. Posted Operating Instructions and Diagrams:

1. Operating Instructions:

a. Copies of operating instructions provided in the operating manual (3.07 A above) shall be posted in the near vicinity of each piece of applicable equipment. The instructions shall be mounted neatly in frames under Plexiglas, where they can be easily read by operating personnel. Instructions mounted outdoors shall be suitably protected from weather.

2. Posted Systems Diagrams:

a. Simplified one (1) line diagrams of the systems listed shall be developed and transcribed on transparent “D” sized erasable sepia film and posted neatly under Plexiglas in the main or most appropriate equipment room for easy reference by operating and maintenance personnel. These drawings shall be done in a professional manner that is acceptable to the Maintenance Division staff. The diagrams shall show each component including all valves installed in the system, with name and identifying number. If space does not permit valve numbers on the diagrams, valve charts shall be provided. Explanatory notes, where needed, shall be provided.

i. Chiller controls diagram

ii. Hydronic distribution system

3. Other systems as applicable

a. These diagrams shall be suitable for reduction in size and use in the operating manual system descriptions previously covered.

3.07 Operating and Maintenance Training

A. The Mechanical Contractor, and appropriate sub-contractors, shall provide comprehensive operating and maintenance instruction on building systems prior to delivery. The instruction shall include classroom instruction delivered by competent instructors based upon the contents of the operating manual. Emphasis shall be placed upon overall systems diagrams and descriptions, and why systems were designed as they were. This overall systems instruction shall preferably be delivered by the consulting engineers. The classroom instruction shall also include detailed equipment instruction by qualified manufacturer representatives for all equipment listed in 2.04 A for which operating instructions are provided. The manufacturer representative training shall emphasize operating instructions, and preventive maintenance as described in the operating manual. At a minimum, the training sessions shall cover the following items:

1. Types of installed systems
2. Theory of operation
   a. Design intent
   b. Occupied vs. unoccupied or partial occupancy
   c. Seasonal modes of operation
   d. Emergency conditions and procedures
   e. Comfort conditions
   f. Energy efficiency
   g. Other issues important to facility operation.
3. System operations.
4. Use of control system
   a. Sequence of operation
   b. Problem indicators
   c. Diagnostics
   d. Corrective actions.
5. Service, maintenance, diagnostics and repair.
6. Use of reports and logs.

B. Each classroom training period shall be followed by an inspection, explanation and demonstration of the system concerned by the instructors. All equipment listed in 3.07 A shall be started up and shut down, with the exception of sprinkler systems.

C. The contractor shall be responsible for organizing, arranging, and delivering this instruction in an efficient and effective manner on a schedule agreeable to the owner.

D. The contractor shall provide, at or before substantial completion, a proposed agenda and schedule of the above training for approval by the Commissioning Authority and the Owner.

E. SAMPLE VERIFICATION TEST PROCEDURE

Chillers

Chiller Start

With chiller 1 (CH-1) as lead,

Verify that all secondary chilled water pumps are in the normal off mode, then initiate a normal start command for at least one of these secondary pumps. Then check the following:

	Yes	No	Notes
Condenser water isolation valve on CH-1 opens			
Condenser water pump (CWP-1) starts			
Condenser water isolation valve on CT-1 opens			
Cooling tower CT-1 fan starts at low speed			
Chilled water pump CHP-1 starts			
Chiller 1 (CH-1) starts			

With chiller 2 (CH-2) as lead,

Verify that all secondary chilled water pumps are in the normal off mode, then initiate a normal start command for at least one of these secondary pumps. Then check the following:

	Yes	No	Notes
Condenser water isolation valve on CH-2 opens			
Condenser water pump (CWP-2) starts			
Condenser water isolation valves on CT-1 & 2 open			
Cooling tower (CT-1 & 2) fans start at low speed			
Chilled water pump (CHP-2) starts			

Chiller Normal Stop

With chiller 1 (CH-1) as lead,

Shut off any secondary pumps that are operating, then check the following:

	Yes	No	Notes
Chiller 1 (CH-1) stops			
Condenser water pump (CWP-1) runs for two minutes before shutting down			
Chilled water pump (CHP-1) runs for two minutes before shutting down			
Cooling tower (CT-1) fan stops operating			
Condenser water isolation valve on CT-1 closes			
Condenser water isolation valve on CH-1 closes			

With chiller 2 (CH-2) as lead,

Shut off any secondary pumps that are operating, then check the following:

	Yes	No	Notes
Chiller 2 (CH-2) stops			
Condenser water pump (CWP-2) runs for two minutes before shutting down			
Chilled water pump (CHP-2) runs for two minutes before shutting down			
Cooling tower CT- 1 & 2 fans stop operating			
Condenser water isolation valves on CT-1 & 2 close			
Condenser water isolation valve on CH-2 closes			

Chiller Fail

With chiller 1 as lead,

Simulate failure of CH-1 by disconnecting at motor control center, then check the following:

	Yes	No	Notes
Chiller 1 (CH-1) stops			
Chiller 2 (CH-2) starts			
Pumps and cooling tower continue operation			

With chiller 2 as lead,

Simulate failure of CH-2 by disconnecting at motor control center, then check the following:

	Yes	No	Notes
Chiller 2 (CH-2) stops			
Chiller 1 (CH-1) starts			
Pumps and cooling tower continue operation			

With chiller 1 as lead,

Simulate a chilled water setpoint that is 5°F above normal setpoint, then check the following:

	Yes	No	Notes
After 15 minutes, lead switches to CH-2			

With chiller 2 as lead,

Simulate a chilled water setpoint that is 5°F above normal setpoint, then check the following:

	Yes	No	Notes
After 15 minutes, lead switches to CH-1			

Lag Chiller Sequencing

With chiller 1 as lead,

Establish these conditions:

- At least one secondary pump is operating at 100% speed for 10 minutes, and
- CHW reset setpoint temperature is less that 1°F above minimum (at 42°F), or flow rate measured at
- CHW flow meter is above CH-1 maximum flow rate (585 GPM).

Then check the following:

	Yes	No	Notes
Condenser water isolation valve on CH-2 opens			
Condenser water pump (CWP-2) starts			
Condenser water isolation valve on CT-2 opens			
Cooling tower (CT-2) fan starts at low speed			
Chilled water pump (CHP-2) starts			
Chiller 2 (CH-2) starts			

If CHW flow rate falls below the design flow rate of CHP-2, check the following:

	Yes	No	Notes
Chiller 2 (CH-2) stops			
Condenser water pump (CWP-2) runs for two minutes before shutting down			
Chilled water pump (CHP-2) runs for two minutes before shutting down			
Cooling tower (CT-2) fan stops operating			
Condenser water isolation valve on CT-2 closes			
Condenser water isolation valve on CH-2 closes			

If the average CHW temperature difference (return temperature minus the average. of supply water temperatures) is less than 5°F and CHW flow rate is 5% below lead CH-1 maximum flow, check the following:

	Yes	No	Notes
Chiller 2 (CH-2) stops			
Condenser water pump (CWP-2) runs for two minutes before shutting down			
Chilled water pump (CHP-2) runs for two minutes before shutting down			
Cooling tower (CT-2) fan stops operating			
Condenser water isolation valve on CT-2 closes			
Condenser water isolation valve on CH-2 closes			

With chiller 2 as lead,

Establish these conditions:

- At least one secondary pump is operating at 100% speed for 10 minutes, and
- CHW reset setpoint temperature is less than 1°F above minimum (at 42°F), or flow rate measured at the CHW flow meter is above CH-2 maximum flow rate (1175 GPM).

Then check the following:

	Yes	No	Notes
Condenser water isolation valve on CH-1 opens			
Condenser water pump (CWP-1) starts			
Condenser water isolation valve on CT-1 opens			
Cooling tower (CT-1) fan starts at low speed			
Chilled water pump (CHP-1) starts			
Chiller 1 (CH-1) starts			

If CHW flow rate falls below the design flow rate of CHP-2, check the following:

	Yes	No	Notes
Chiller 1 (CH-1) stops			
Condenser water pump (CWP-1) runs for two minutes before shutting down			
Chilled water pump (CHP-1) runs for two minutes before shutting down			
Cooling tower (CT-1) fan stops operating			
Condenser water isolation valve on CT-1 closes			
Condenser water isolation valve on CH-1 closes			

If average CHW temperature difference (return minus avg. of supply water temps.) is less than 5°F and CHW flow rate is 5% below lead CH-2 maximum flow,

	Yes	No	Notes
Chiller 1 (CH-1) stops			
Condenser water pump (CWP-1) runs for two minutes before shutting down			
Chilled water pump (CHP-1) runs for two minutes before shutting down			
Cooling tower (CT-1) fan stops operating			
Condenser water isolation valve on CT-1 closes			
Condenser water isolation valve on CH-1 closes			

Chilled Water Reset

Chilled water supply temperature is reset within a range of 41°F to 48°F, based on the chilled water valve position at the AHU with the highest cooling load.

Check the chilled water setpoints at various valve positions according to the following:

	Yes	No	Notes
Valve 95% open, CHW temperature is 41°F			
Valve 83% open, CHW temperature is 42°F			
Valve 71% open, CHW temperature is 43°F			
Valve 59% open, CHW temperature is 44°F			
Valve 47% open, CHW temperature is 45°F			
Valve 35% open, CHW temperature is 46°F			
Valve 23% open, CHW temperature is 47°F			
Valve 11% or less, CHW temperature is 48°F			

Pumps

Primary Pump Operation

Manual on

With no chiller plant equipment operating, manually turn CHP-1 on.

	Yes	No	Notes
CH-1 and peripherals (CHP-1, CWP-1, CT-1) become lead and start normally			

With no chiller plant equipment operating, manually turn CHP-2 on.

	Yes	No	Notes
CH-2 and peripherals (CHP-2, CWP-2, CT-2) become lead and start normally			

Pump Failure

With CHP-1 as lead pump (also CH-1, CT-1, and CWP-1 operating), "fail" CHP-1 by disconnecting at motor control center. Then check:

	Yes	No	Notes
CH-1 shutdown procedure occurs			
CH-2 start procedure occurs			
CH-2 and peripherals become lead			

With CHP-2 as lead pump (also CH-2, CT-2, and CWP-2 operating), "fail" CHP-2 by disconnecting at motor control center. Then check:

	Yes	No	Notes
CH-2 shutdown procedure occurs			
CH-1 start procedure occurs			
CH-1 and peripherals become lead			

Pump Control by CHW Flow

With CH-1 and peripherals operating alone, manipulate control valves at AHU's to have secondary flow exceed design flow of CH-1 by 10% (CHW flow of 645 GPM).

	Yes	No	Notes
CHP-2 starts and becomes lead pump			
CHP-1 stops and becomes lag pump			

Manipulate control valves at AHU's to have secondary flow drop back down below design flow of CH-1 (CHW flow of 585 GPM or less).

	Yes	No	Notes
CHP-1 starts and becomes lead pump			
CHP-2 stops and becomes lag pump			

Secondary Pump Operation

Pumps CH-S1 through CH-S6 Start

Pumps should start when pump speed signal is greater than 20% for 5 minutes.

Manipulate control valve at each AHU to 25% open to initiate a call for chilled water.

	Yes	No	Notes
Pump CH-S1 starts after 5 minutes			
Pump CH-S2 starts after 5 minutes			
Pump CH-S3 starts after 5 minutes			
Pump CH-S4 starts after 5 minutes			
Pump CH-S5 starts after 5 minutes			
Pump CH-S6 starts after 5 minutes			

Pumps/VFD CH-S1 through CH-S6 operation

Verify that pump/VFD are ramping up and down to meet cooling load.

Manipulate control valve at each AHU to 50% open to call for more chilled water.

	Yes	No	Notes
Pump/VFD CH-S1 ramps up to meet load			
Pump/VFD CH-S2 ramps up to meet load			
Pump/VFD CH-S3 ramps up to meet load			
Pump/VFD CH-S4 ramps up to meet load			
Pump/VFD CH-S5 ramps up to meet load			
Pump/VFD CH-S6 ramps up to meet load			

Manipulate control valve at each AHU to 100% open to call for more chilled water.

	Yes	No	Notes
Pump/VFD CH-S1 ramps up to meet load			
Pump/VFD CH-S2 ramps up to meet load			
Pump/VFD CH-S3 ramps up to meet load			
Pump/VFD CH-S4 ramps up to meet load			
Pump/VFD CH-S5 ramps up to meet load			
Pump/VFD CH-S6 ramps up to meet load			

Manipulate control valve at each AHU back to 25% open to call for less chilled water.

	Yes	No	Notes
Pump/VFD CH-S1 ramps down to meet load			
Pump/VFD CH-S2 ramps down to meet load			
Pump/VFD CH-S3 ramps down to meet load			
Pump/VFD CH-S4 ramps down to meet load			
Pump/VFD CH-S5 ramps down to meet load			
Pump/VFD CH-S6 ramps down to meet load			

Pumps CH-S1 through CH-S6 Stop

Manipulate control valve at each AHU closed to send stop command to pump.

	Yes	No	Notes
Pump CH-S1 stops			
Pump CH-S2 stops			
Pump CH-S3 stops			
Pump CH-S4 stops			
Pump CH-S5 stops			
Pump CH-S6 stops			

Pumps CH-S7 through CH-S10 Start

These pumps only serve fan coil units and should start only when outside air temperature is above 60°F.

With outside air temperature above 60°F (it may be necessary to change setpoint if weather conditions aren't favorable), check the following:

	Yes	No	Notes
CHW control valve at unit opens			
When control valve hits 50% open, pump starts			

Pumps CH-S7 through CH-S10 Stop

Pumps should stop when CHW control valves close.

Change outside air temperature setpoint to be below outside air temperature, then check:

	Yes	No	Notes
CHW control valve at unit closes			
Pump stops			

Cooling Towers

Start/Stop

This verification test is covered with chiller start/stop tests.

Fan Cycling

With CH-1 operating,

Observe tower operation to verify that fan cycles between low speed and off to maintain 15°F difference between CW return temperature and CHW supply temperature (may need to adjust setpoints depending on weather). Check the following:

	Yes	No	Notes
Fan cycles between off and low speed to maintain 15°F differential between CW return & CHW supply			

Observe tower operation to verify that fan cycles between low speed and high speed to maintain a CW supply temperature of 68°F (may need to make adjustments to setpoints depending on weather), then check:

	Yes	No	Notes
Fan cycles between low speed and high speed, maintaining 68°F CW temperature			

With CH-2 operating,

Observe tower operation to verify that fan cycles between low speed and off to maintain 15°F difference between CW return temperature and CHW supply temperature (may need to adjust setpoints depending on weather), then check:

	Yes	No	Notes
Fan cycles between off and low speed, maintaining 15°F differential between CW return & CHW supply			

Observe tower operation to verify that fan cycles between low speed and high speed to maintain a CW supply temperature of 68°F (may need to make adjustments to setpoints depending on weather), then check:

	Yes	No	Notes
Fan cycles between low speed and high speed, maintaining 68°F CW temperature			

Tower Failure

With CT-1 operating,

Simulate failure by disconnecting fan motor at motor control center or local disconnect, then check:

	Yes	No	Notes
CT-2 starts and becomes lead			

With CT-2 operating,

Simulate failure by disconnecting fan motor at motor control center or local disconnect, then check:

	Yes	No	Notes
CT-1 starts and becomes lead			

Tab Report Verification

Flow Rates

Chilled Water Flows

Flow through CHP-1

TAB reading _______ GPM

Verification reading _______ GPM

Flow through CHP-2

TAB reading _______ GPM

Verification reading _______ GPM

Condenser Water Flows

Flow through CWP-1

TAB reading _______ GPM

Verification reading _______ GPM

Flow through CWP-2

TAB reading _______ GPM

Verification reading _______ GPM

Pump Shutoff Head / Impeller Size

CWP-1

TAB reading ________

Verification reading ________

Impeller size ________

CWP-2

TAB reading ________

Verification reading ________

Impeller size ________

CHP-1

TAB reading ________

Verification reading ________

Impeller size ________

CHP-2

TAB reading ________

Verification reading ________

Impeller size ________

CHP-S1

TAB reading ________

Verification reading ________

Impeller size ________

CHP-S2

TAB reading ________

Verification reading ________

Impeller size ________

CHP-S3

TAB reading ________

Verification reading ________

Impeller size ________

CHP-S4

TAB reading ________

Verification reading ________

Impeller size ________

CHP-S5

TAB reading ________

Verification reading ________

Impeller size ________

CHP-S6

TAB reading ________

Verification reading ________

Impeller size ________

CHP-S7

TAB reading ________

Verification reading ________

Impeller size ________

CHP-S8

TAB reading ________

Verification reading ________

Impeller size ________

CHP-S9

TAB reading ________

Verification reading ________

Impeller size ________

CHP-S10

TAB reading ________

Verification reading ________

Impeller size ________