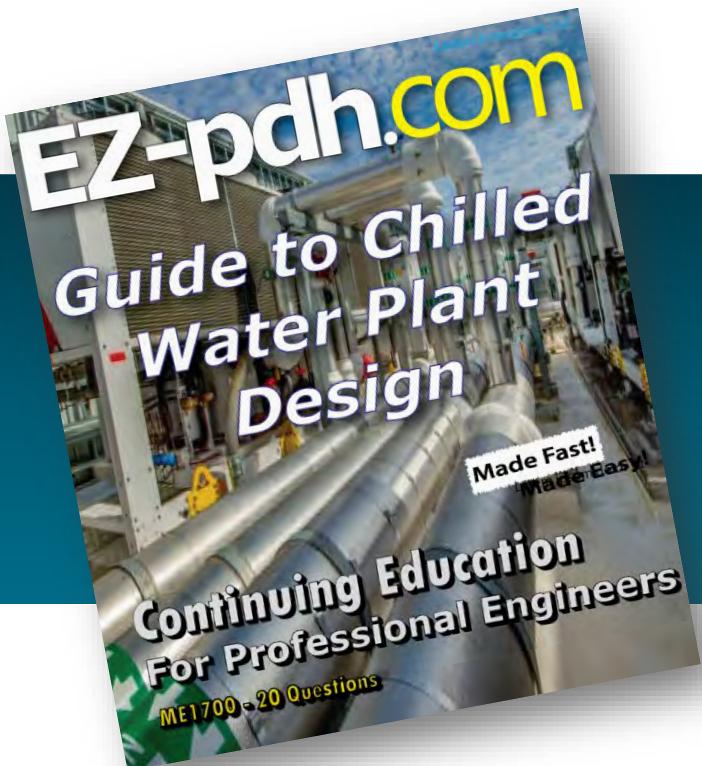


Guide to Chilled Water Plant Design: 5 PDH

Five (5) Continuing Education Hours
Course #ME1700

Approved Continuing Education for Licensed Professional Engineers

EZ-pdh.com
Ezekiel Enterprises, LLC
301 Mission Dr. Unit 571
New Smyrna Beach, FL 32170
800-433-1487
support@ez-pdh.com

Course Description:

The Guide to Chilled Water Plant Design course satisfies five (5) hours of professional development.

The course is designed as a distance learning course that provides detailed guidance on chilled water plant design, covering load estimation, chiller and pump selection, system piping, instrumentation and control strategies, optimization procedures, procurement strategies, and commissioning. Students are also introduced to variable-speed technologies and life-cycle cost evaluations.

Objectives:

The primary objective of the course is to equip engineers with the knowledge and tools to design energy-efficient, cost-effective, and operationally resilient chilled water plants.

Grading:

Students must achieve a minimum score of 70% on the online quiz to pass this course. The quiz may be taken as many times as necessary to successfully pass and complete the course.

A copy of the quiz questions is attached to the last pages of this document.

Table of Contents

Guide to Chilled Water Plant Design

1. OVERVIEW

This course serves as a design guide for chilled water plants. It defines the intended audience, outlines the structure of the content, provides chapter-by-chapter summaries, and offers instructions on how to effectively navigate and apply the material.

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.

This course is written for mechanical engineers who design, redesign or retrofit chilled water plants. The course 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 course also provides useful information for operation and maintenance personnel, mechanical contractors, and building managers.

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.

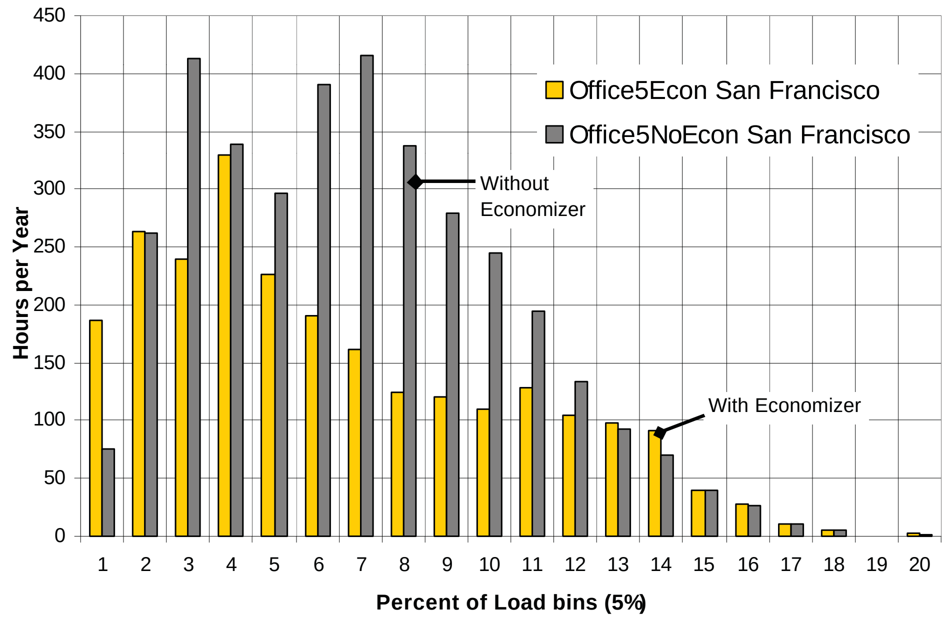
Certain key load parameters affect the cooling load profile and consequently the nature of the plant design. These parameters include:

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.

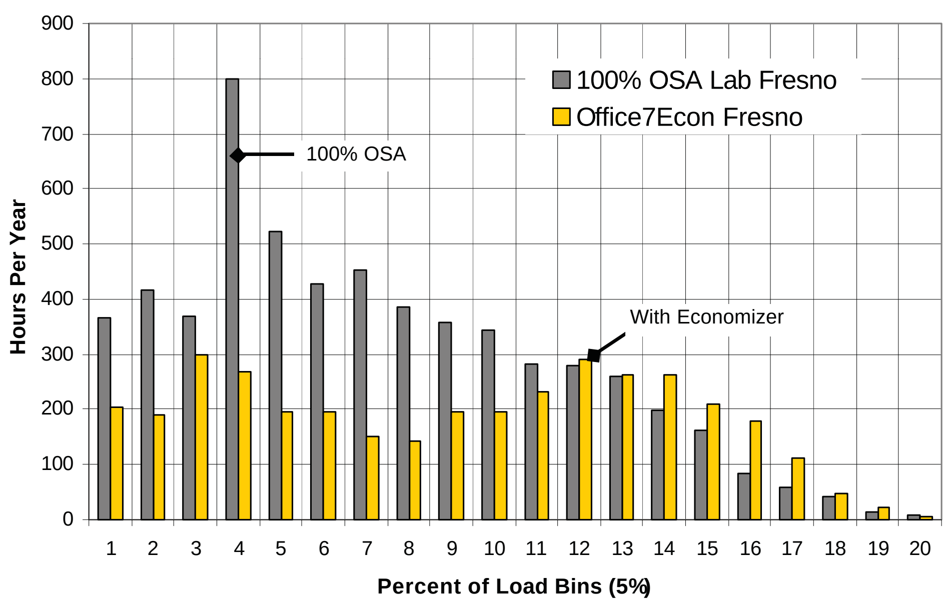
For example, the cooling load profile of a San Francisco office building that operates five days per week was analyzed with and without economizers. As Figure 2-1 shows, the number of hours that the plant operates increases dramatically when an economizer is not used. Additionally the shape of the profile 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.

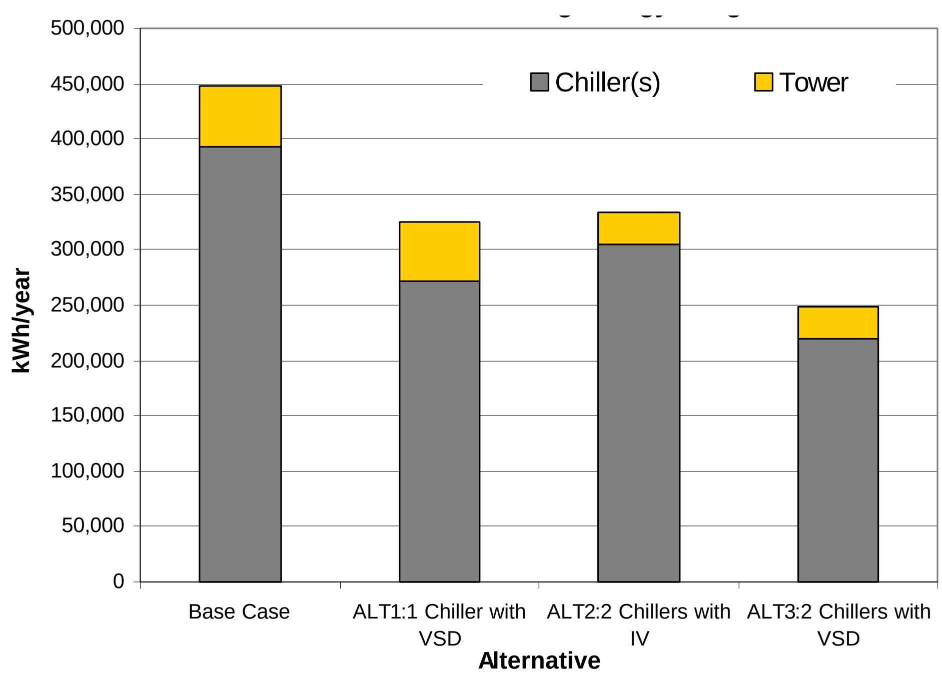
To mitigate problems with oversized plants, the chilled water plant must run efficiently at low loads. The following example from a computer simulation model helps demonstrate the issue of oversizing. In this case, an 800-ton cooling plant serves an office complex that operates on a basic five days per 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,,CLI1DIV12_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

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 course 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:

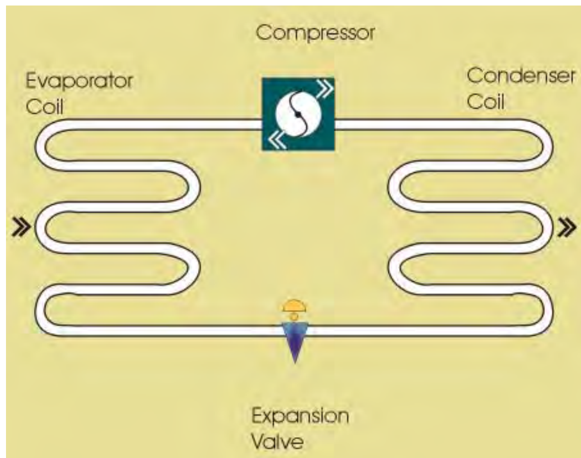
- Carrier Corporation: http://www.commercial.carrier.com/commercial/hvac/general/0,,CLI1_DIV12_ETI1508_MID4369,00.html
- Copeland Corporation (scroll compressors): http://www.copeland-corp.com/
- Danfoss (Turborcor compressor): http://www.turbocor.com/
- McQuay International: http://www.mcquay.com/McQuay/ProductInformation/AllProducts/AllProducts
- 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.

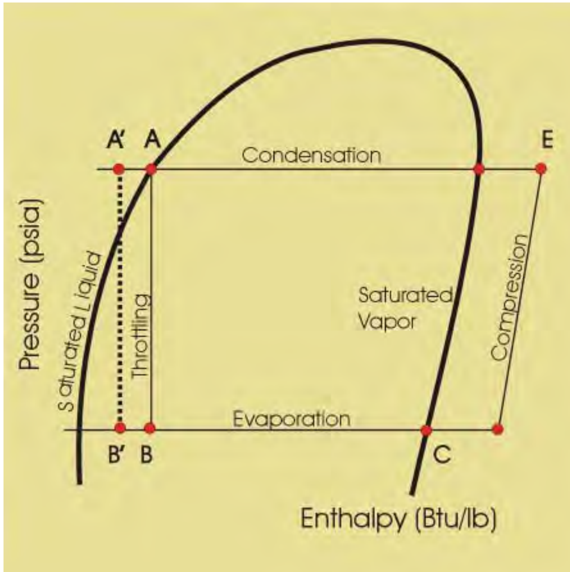


FIGURE 3-2:
PRESSURE-ENTHALPY CHART

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.

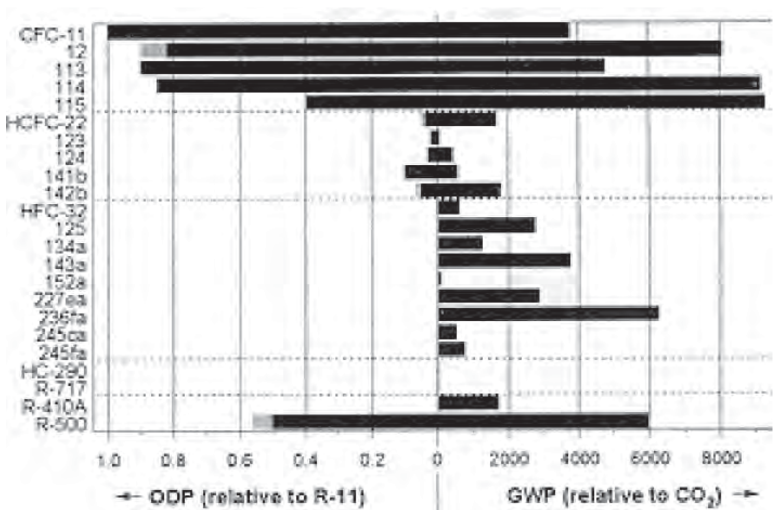
TABLE 3-1:
MONTREAL PROTOCOL

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.

Water Chiller Components

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.

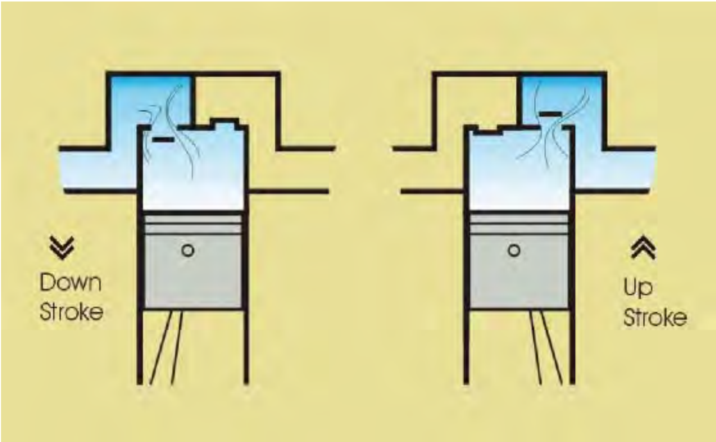
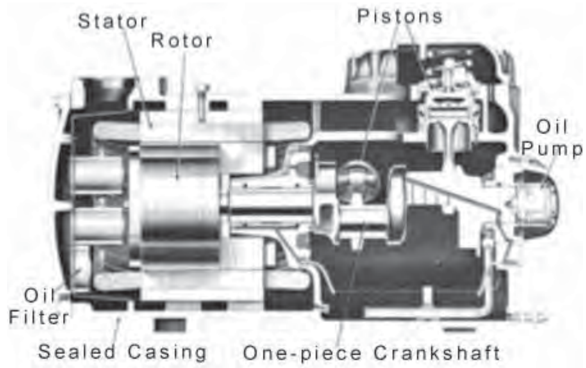


Figure 3-4:
Reciprocating Compressor

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.

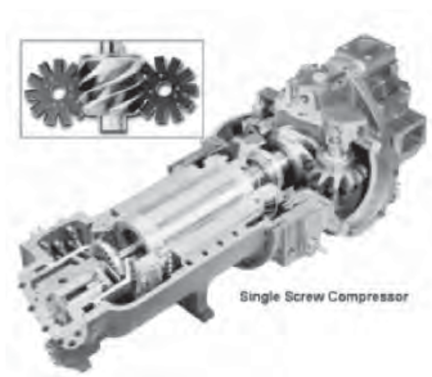


Figure 3-6: The Single Screw Compressor

- **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.

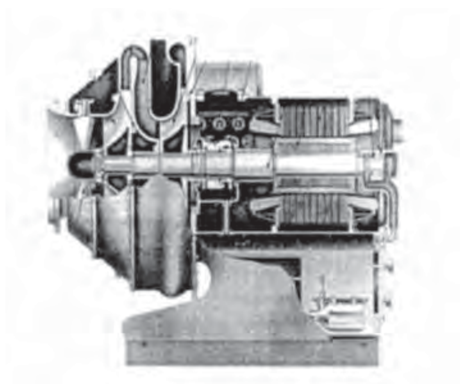


Figure 3-8: Hermetic Centrifugal Compressor

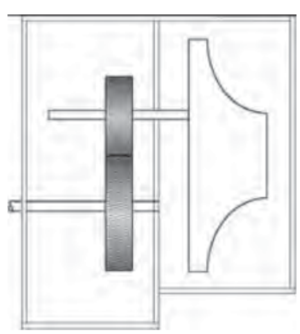


Figure 3-9: Gear Drives

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.

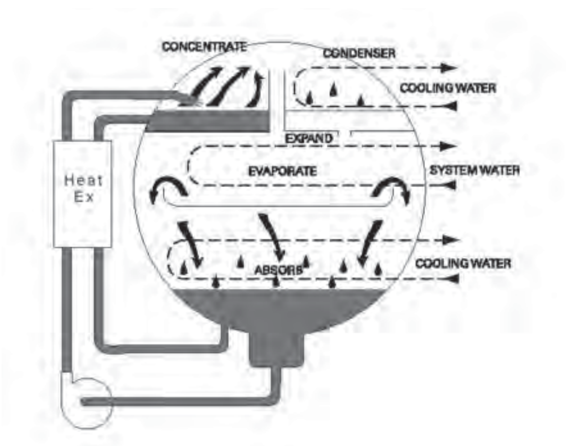
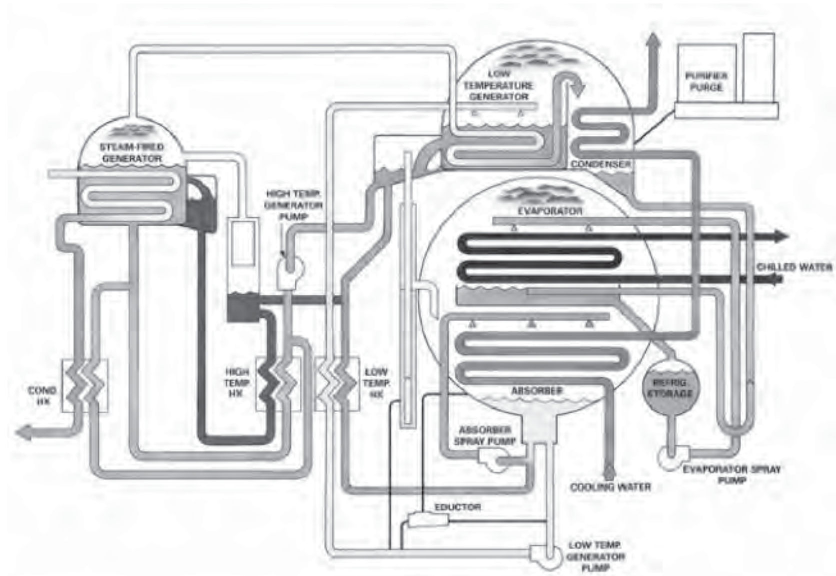


Figure 3-11: Absorption Refrigeration Cycle

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.

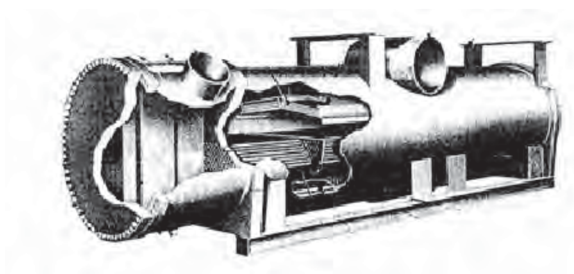


Figure 3-13:
Flooded Shell and
Tube Heat Exchanger

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.

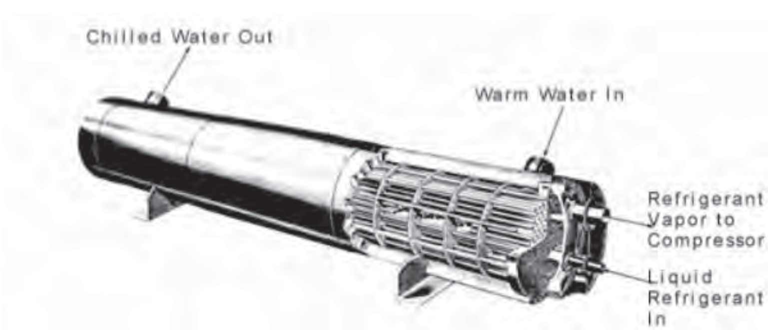


Figure 3-14:
Direct Expansion (DX)

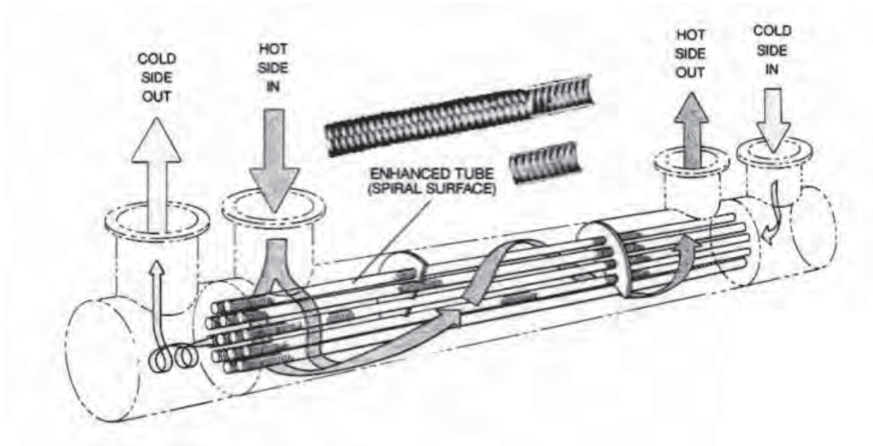


Figure 3-15: Polypropylene Internal Baffles

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.

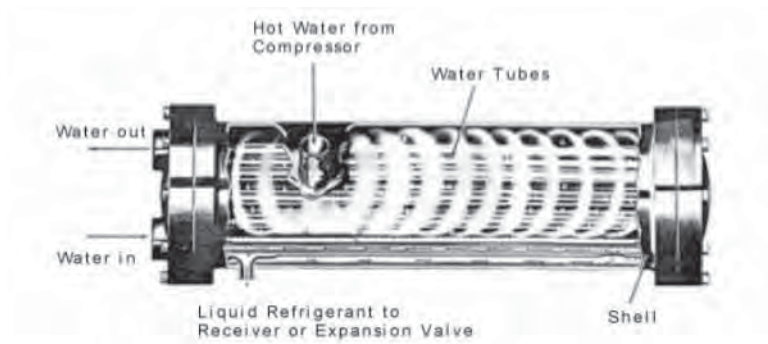


Figure 3-16: Horizontal Shell and Tube Condenser

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

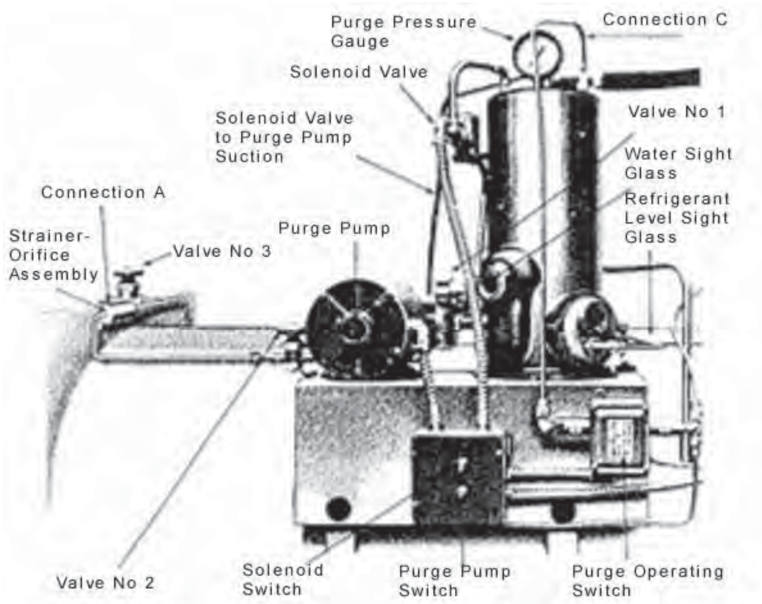
Safety controls protect the unit under abnormal conditions. Safety controls usually trip out the compressor motor and will require manual reset. Typical safety controls include:

- 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



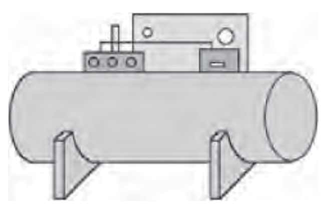
Oil Coolers

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.

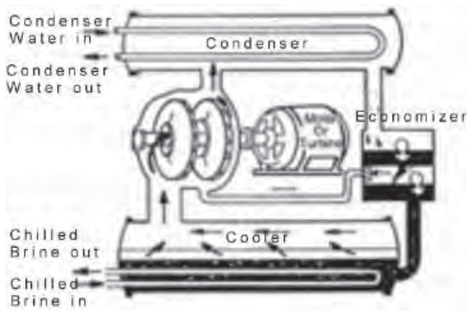


Figure 3-19:
Free Cooling in
Centrifugal Chillers

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:

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

EQUATION 3-1

The higher the number, the more energy efficient the machine. ASHRAE Standard 90.1-2004[7] and Title 24-2005[8] 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:

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-ft²-F/Btu (Evaporator) 0.00025 h-ft²-F/Btu (Condenser)

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

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.

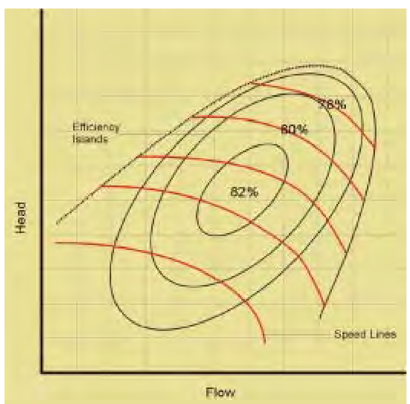


FIGURE 3-20: PERFORMANCE LEVEL

- ***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 course. 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).

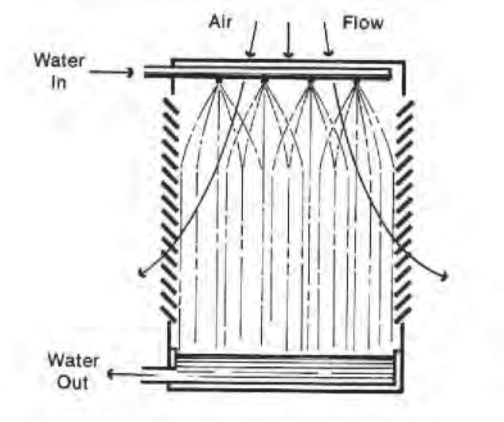
Water surface area is increased by using "fill." Fill can be "splash-type" or "film-type." Film-type fill is most commonly used and consists of closely spaced sheets of PVC arranged vertically. Splash-type fill uses bars to break up the water as it cascades through staggered rows.

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.

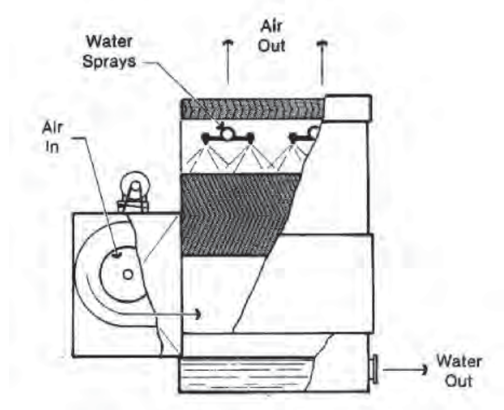
- ***Spray Towers***. Spray towers distribute high-pressure water through nozzles into a chamber where air is induced to flow with the water spray. There are no fans. The air exits out the side of the tower after going through mist eliminators. Spray towers are seldom used. One problem is that the nozzles 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.

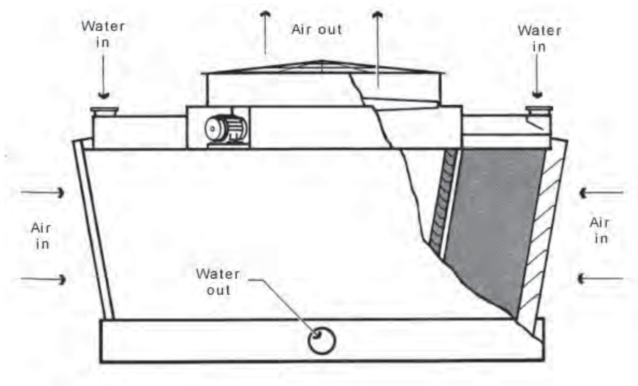


Figure 3-23: Induced Draft Towers

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.

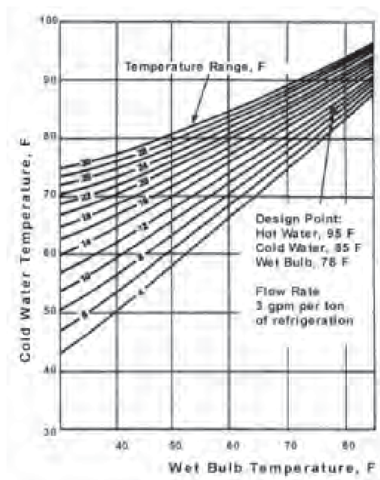
New technologies are being introduced for the treatment and cleaning of cooling towers. One treatment is the introduction of ozone (O_3) into the cooling water. Ozone is a very aggressive oxidizer and when properly applied can be effective at reducing biological growth. One pitfall in the use of ozone 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

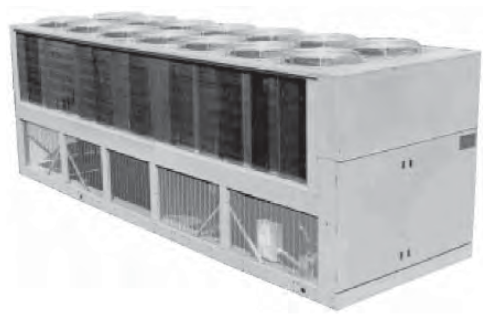
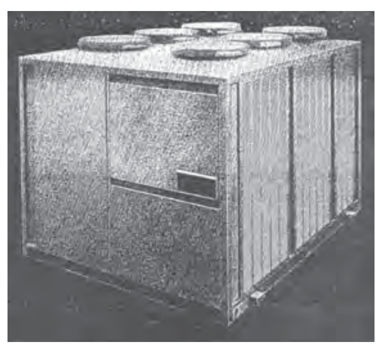


FIGURE 3-26: REMOTE AIR-COOLED CONDENSER



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.

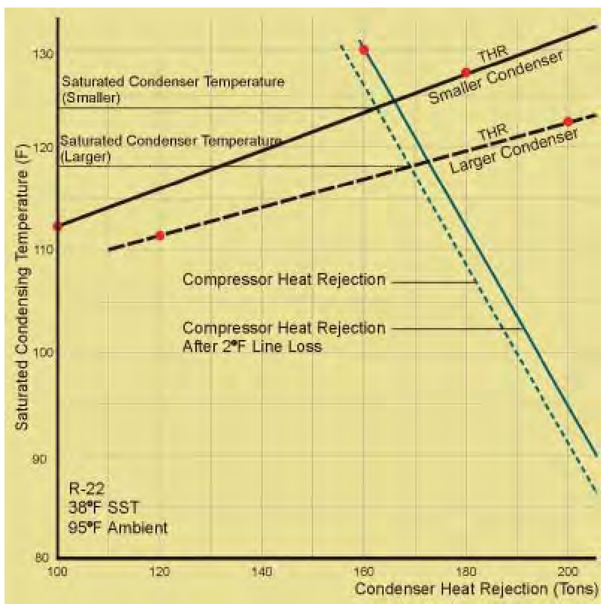


Figure 3-27: Graphical Method (THR of Compressor vs. THR of Condenser Selections)

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.

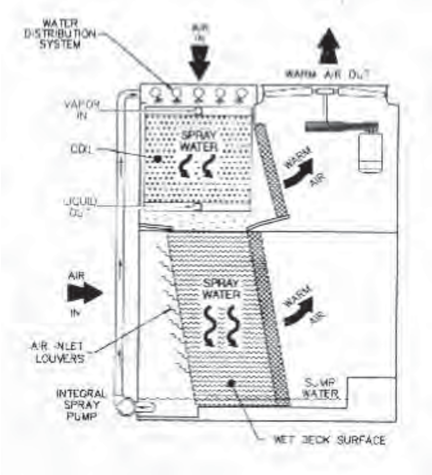


Figure 3-28: Evaporative Condenser

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

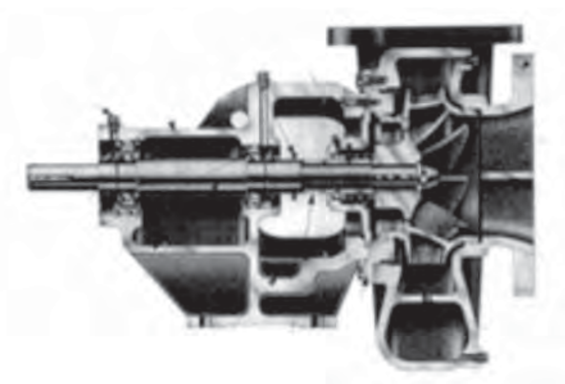
Pump Types

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

Single Suction

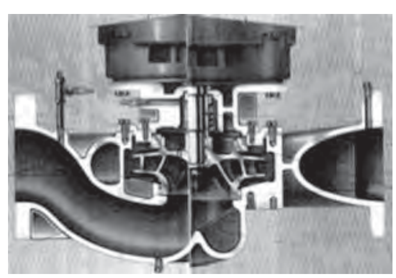
Base-mounted single suction pumps can be either close-coupled or flexible-coupled. Close-coupled pumps use a special motor that has an extended shaft to which the pump impeller is directly connected. The motor and pump cannot be misaligned and they take up less floor space than flexible-coupled pumps. 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.

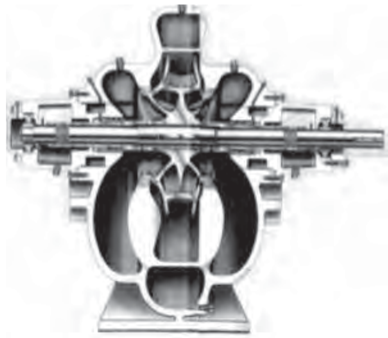


Figure 3-31:
Double Suction Pump

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.

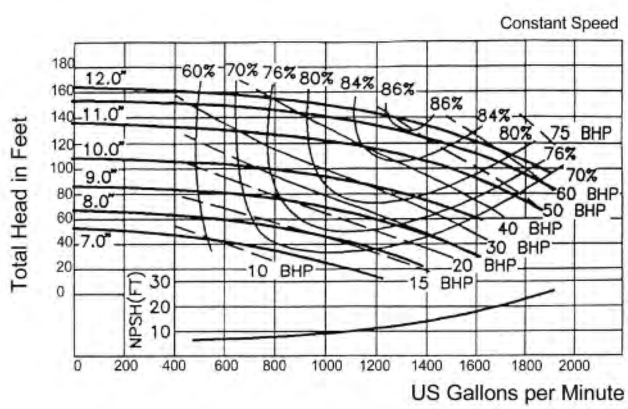


Figure 3-33:
Head Capacity Curve

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.

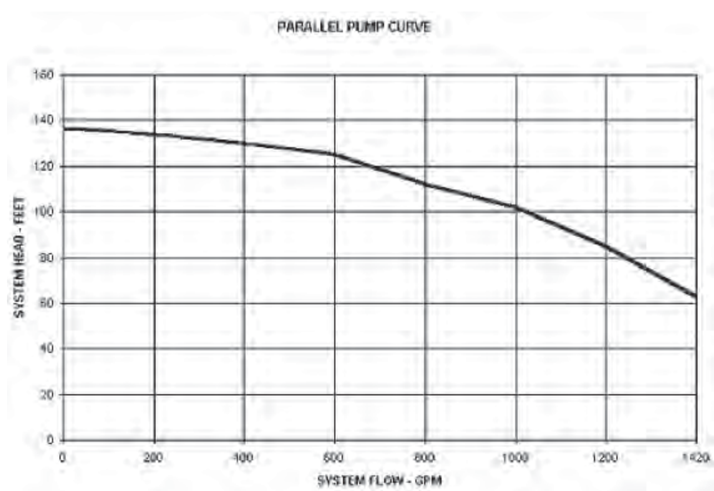


Figure 3-34:
Parallel Pump Curve

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.

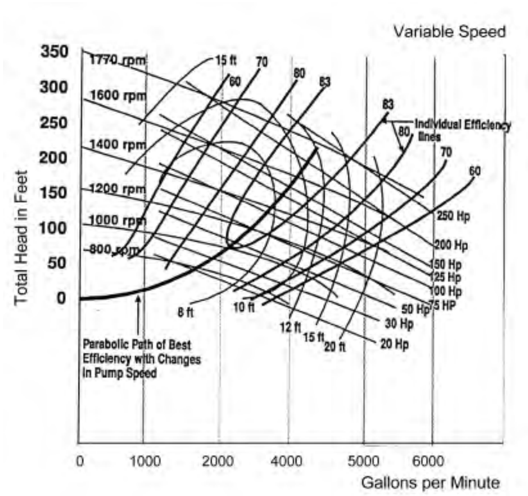


Figure 3-35: Variable-Speed Performance Curve

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

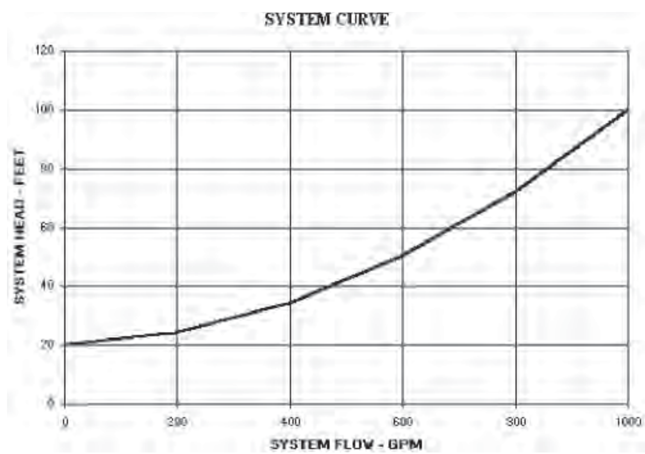
The affinity laws govern the performance of a pump under varying conditions of flow and head pressure. The most important law is:

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.

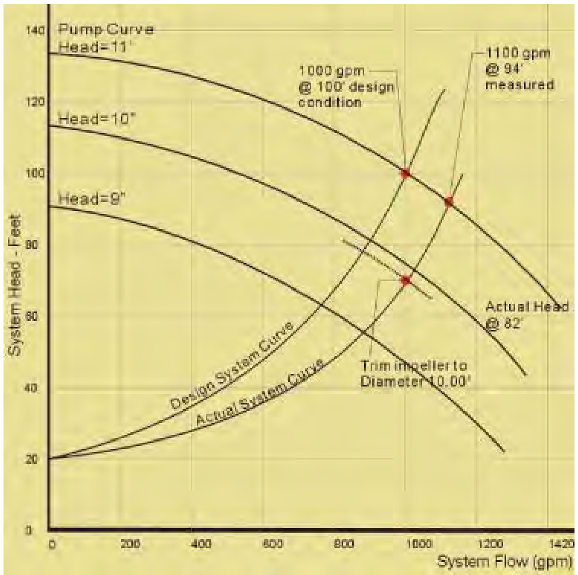


Figure 3-37:
System Curve (for Sizing New Impeller)

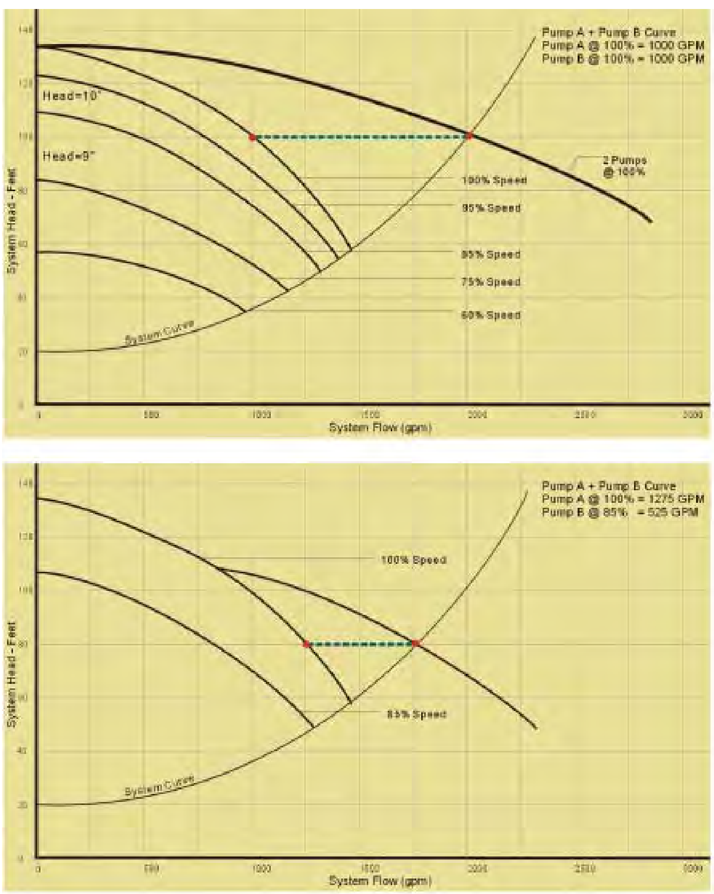
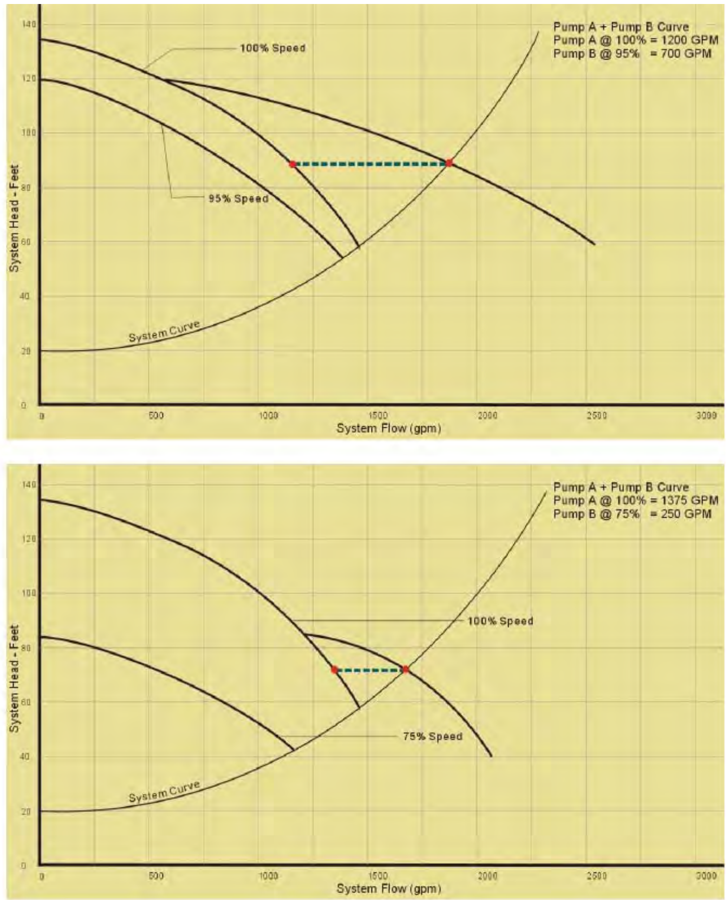


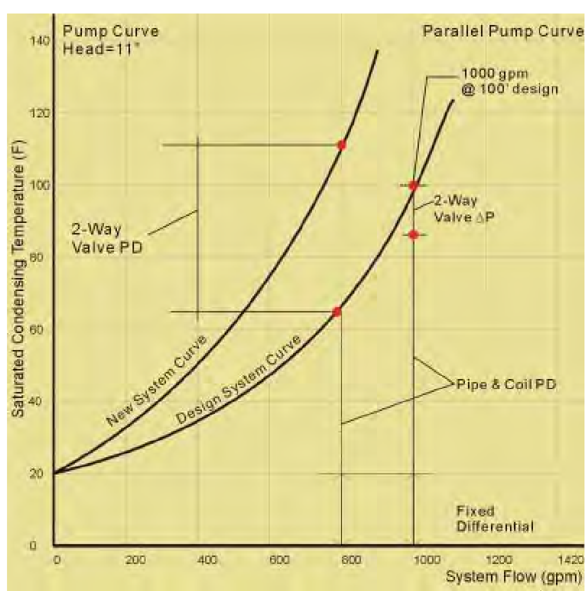
Figure 3-38:
Pump Curves

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.

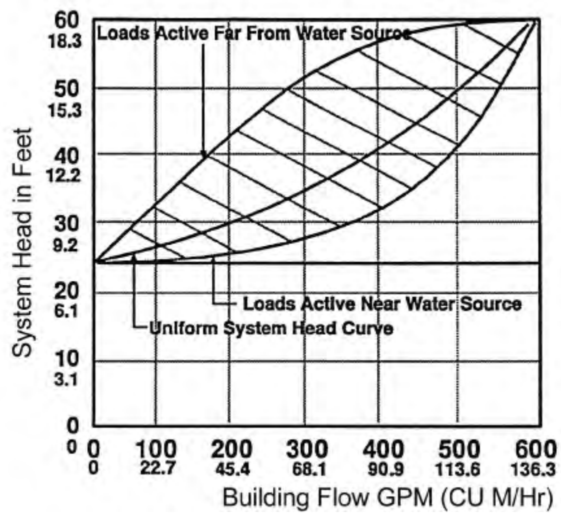


Figure 3-40:
Range of System Curves

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.

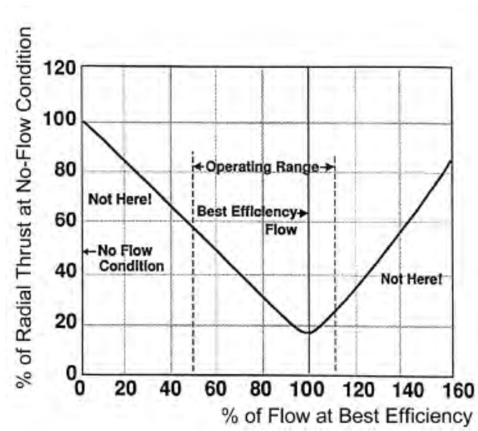
Another pump inlet problem to be avoided is "vortexing" or air entrainment. Any time water is drawn from an open tank or sump there is a potential that a vortex will occur. Vortexing will cause air to enter the pump suction line and will decrease the effectiveness of the pump. Vortexing will occur even 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 course. 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.

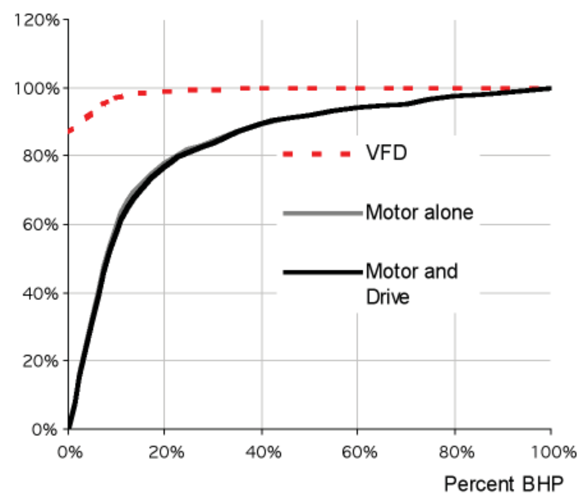


Figure 3-42: Typical Performance, Drive and Motor

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 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.

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.

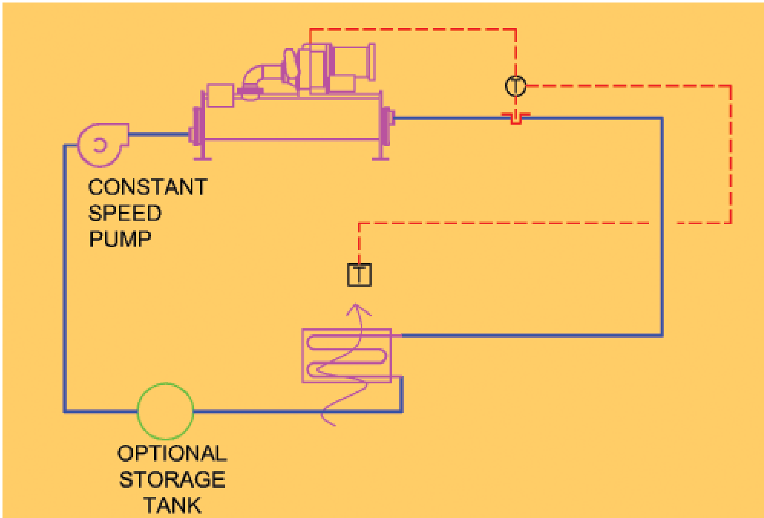


Figure 4-1:
Constant-Flow System,
Single Chiller, Single Coil

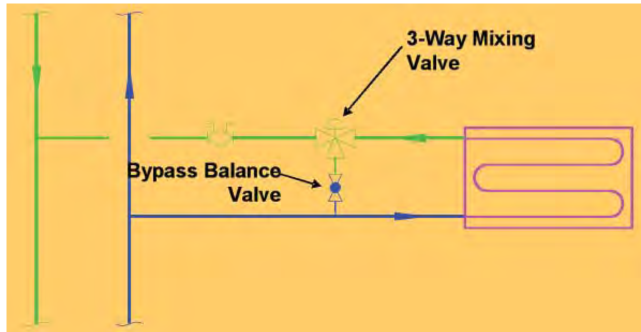
Single Chiller with Multiple Cooling Loads

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

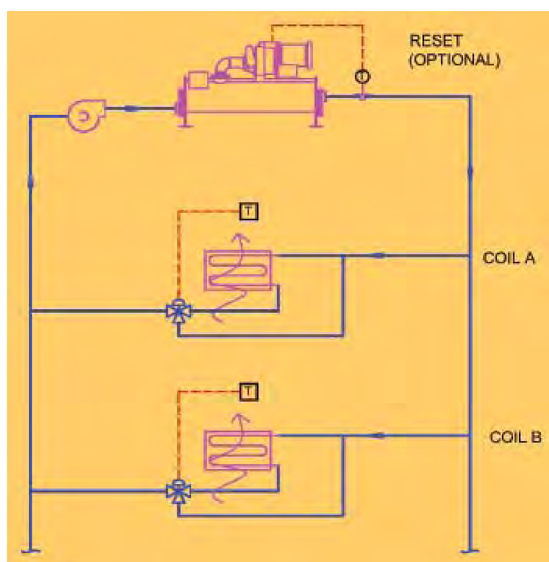


	Pressure Drop @ 100 GPM		
Item	100% to Coil	50% to Coil	0% to Coil
Pipe/Valves	2.0	2.0	2.0
Coil Bypass	8.0	2.0	6.0
Control Valve	10.0	7.5	12.0
Total	20.0	11.5	20.0
GPM @ 20' ΔP*	100	132	100

*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.

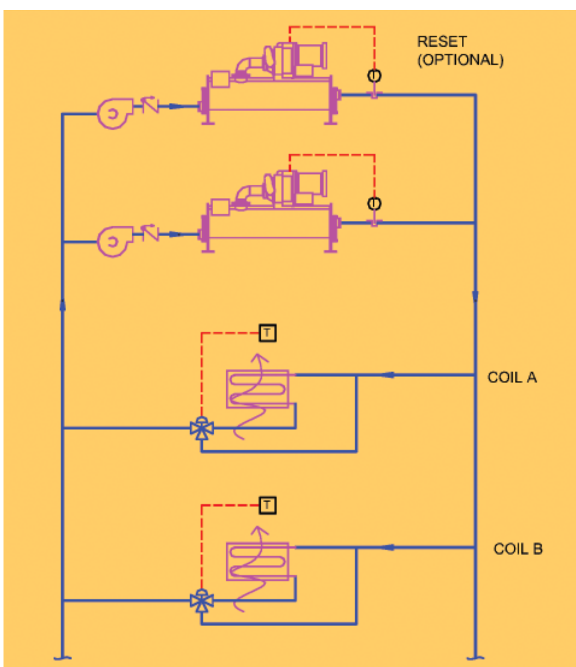


Figure 4-3:
Constant-Flow System,
Multiple Parallel Chillers,
Multiple Coils

Multiple Series Chillers with Multiple Load

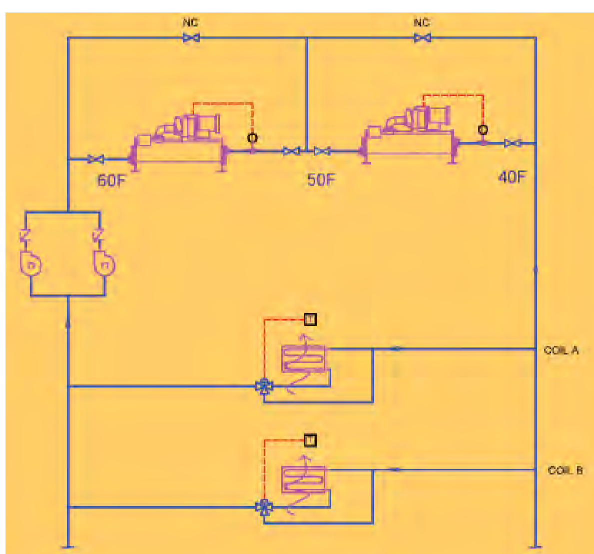
One solution to using a constant-flow system with multiple chillers is to put the chillers in series. In this configuration, the entire flow passes through each machine. This method is effective for systems designed with a high delta-T (15 to 20°F). During off-peak periods, the *lag machine* (second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

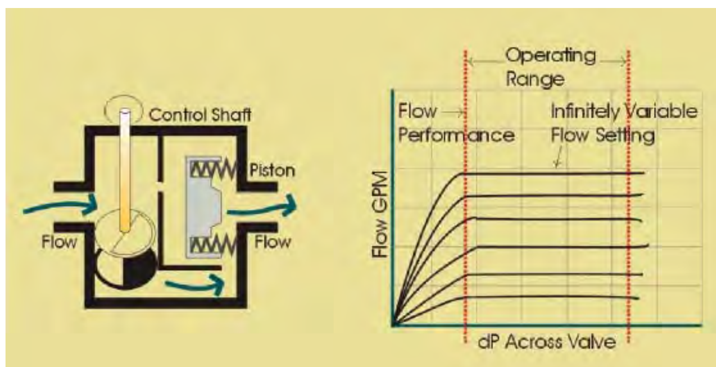
Because of these risks, designers traditionally chose constant flow rates through the evaporators. However, due to the operational advantages (discussed below) coupled with the success of the new chiller internal control strategies, variable flow through the evaporator is rapidly becoming standard practice.

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.

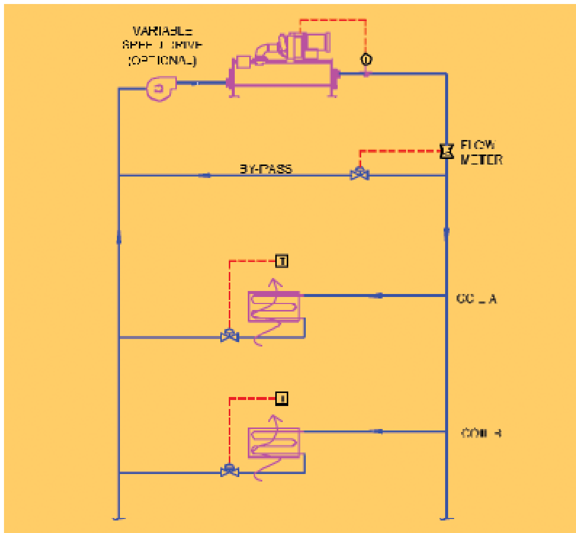


Figure 4-6: Primary-Only Variable Flow Piping, Single Chiller, Multiple Coils

- ***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:

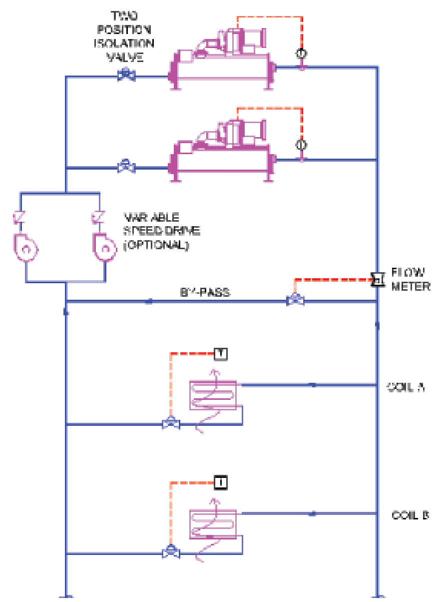
  1. The bypass setpoint changes with the operating chiller(s). For a system with two equally sized chillers there are just two setpoints (one for a single chiller and one for both chillers in operation). For systems with unequally sized chillers, with more than two chillers or with different chillers, 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

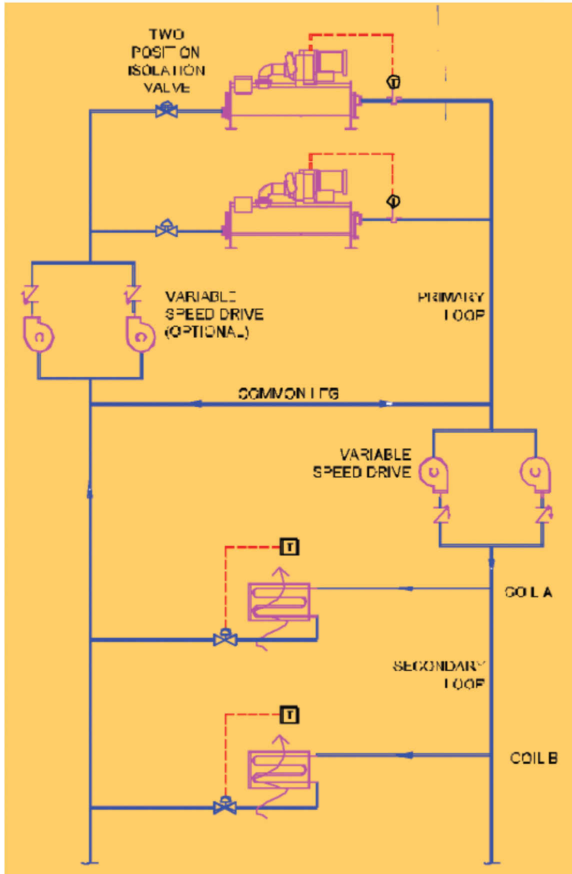
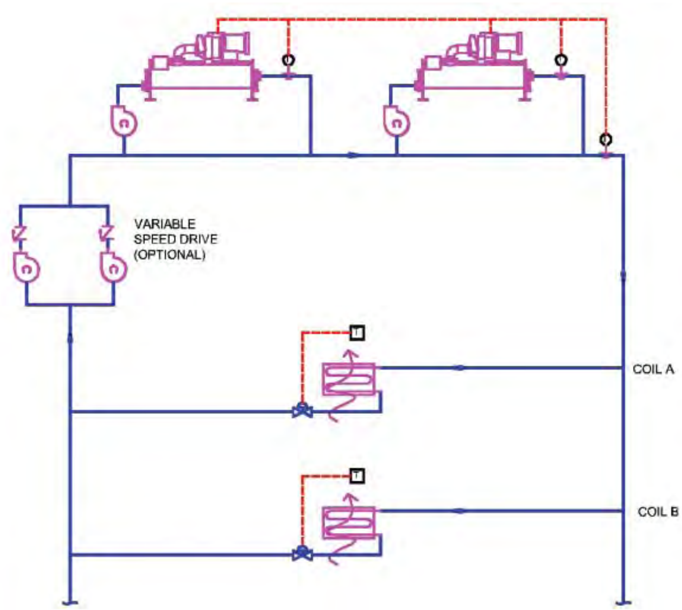


Figure 4-9:
Primary/Secondary Variable Flow Piping, Multiple Chillers, Multiple Coils, Multiple Series Chillers, Multiple Coils



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.

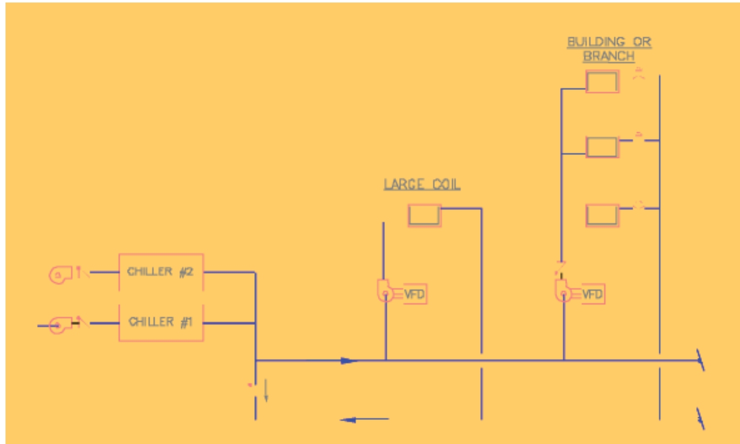
In very large systems (say 5,000 tons or pipes larger than 20 inches), the cost savings achieved by downsizing the common pipe should be considered.

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.

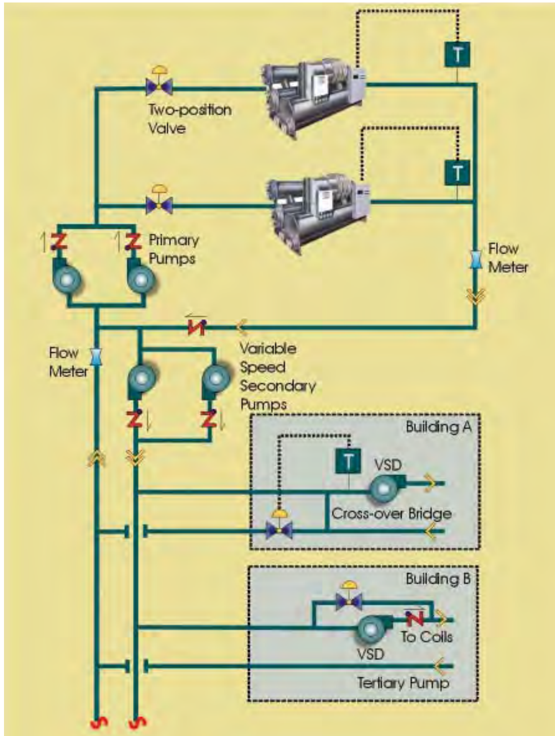


Figure 4-11: Primary/Secondary Variable Flow Piping, Tertiary Pumping

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 .

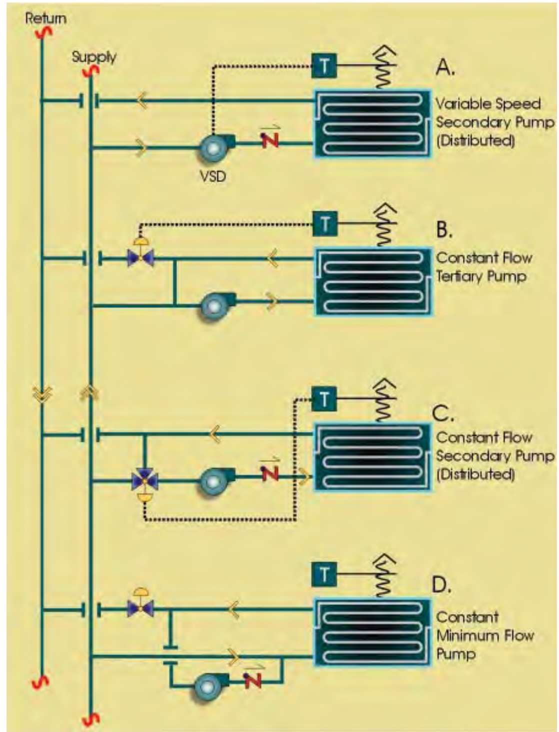
If it is desired to keep constant flow within the coil, a three-way valve can be added to the coil pump circuit that will allow variable flow between the coil and the common pipe (see C). Caution must be applied to this approach because the flow through the pump and coil will vary as a function of the 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.

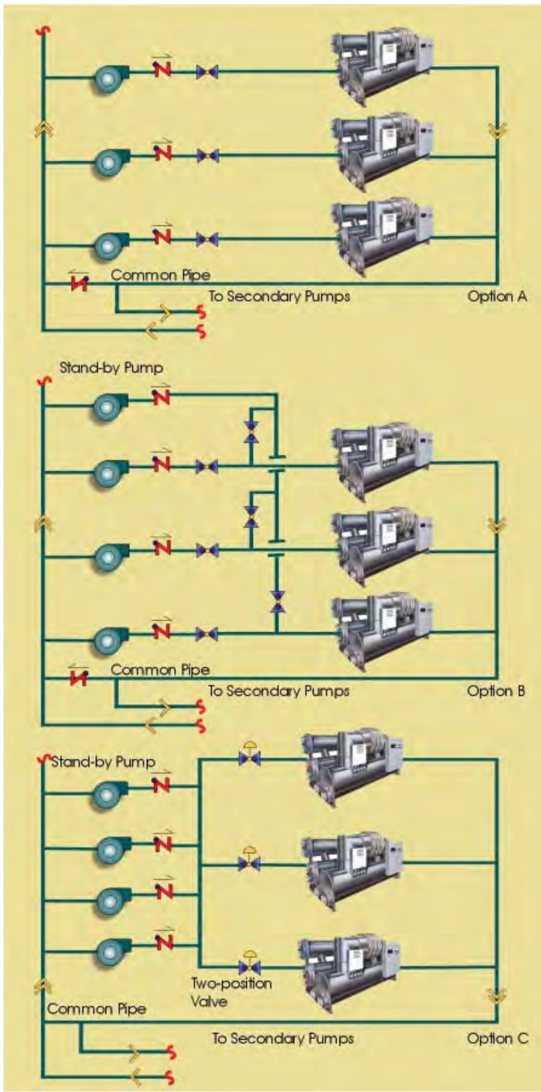


Figure 4-13: Options for Primary Pumps

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

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

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.

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

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		

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:

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

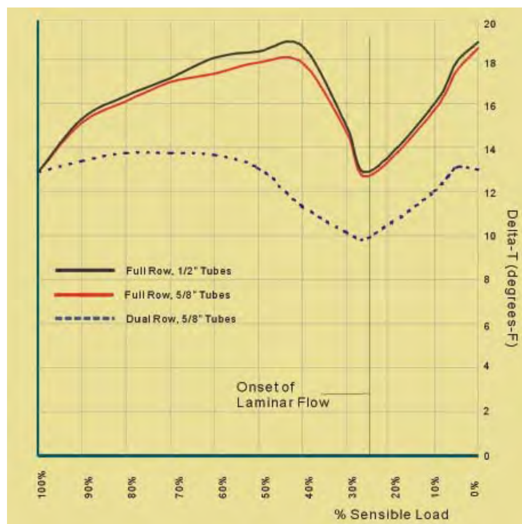
Equation 4-2

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.

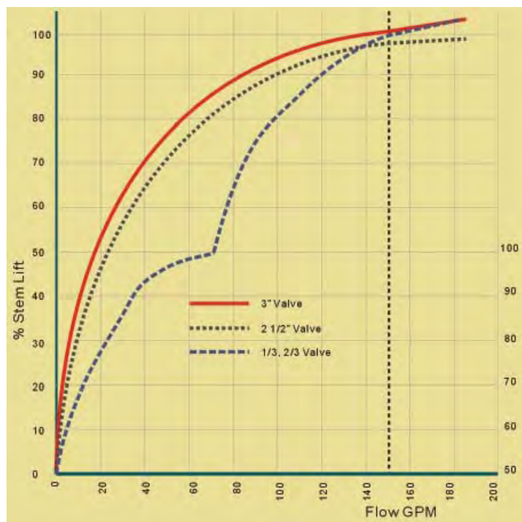
Mitigation. Do not use three-way valves in variable-flow systems. One way to maintain minimum flow in the system is to strategically place two-position bypass valves across the supply and return with flow control valves and open these valves only when the flow at the pumps approaches the minimum flow.

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.

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.

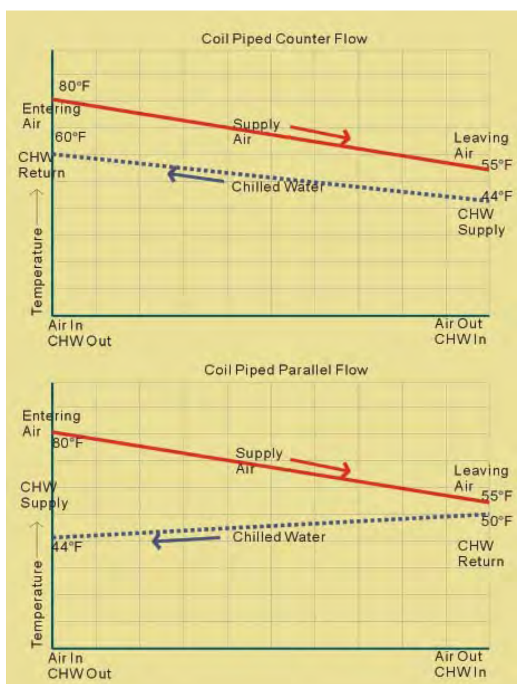
Mitigation. In these cases the only alternative available is to accept that there will be times when low delta-T is inevitable and to design the central plant to accommodate this. This is discussed further in the section, "Designing Chiller Plants to Accommodate Low Delta-T."

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.

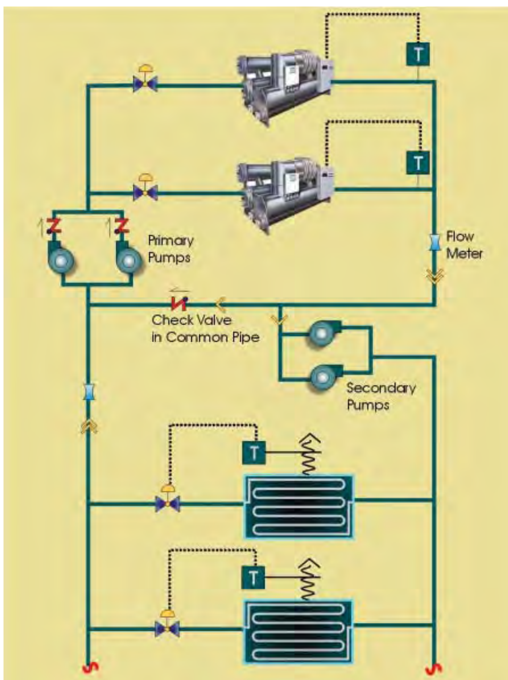


Figure 4-17:
Check Valve in Common Pipe

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

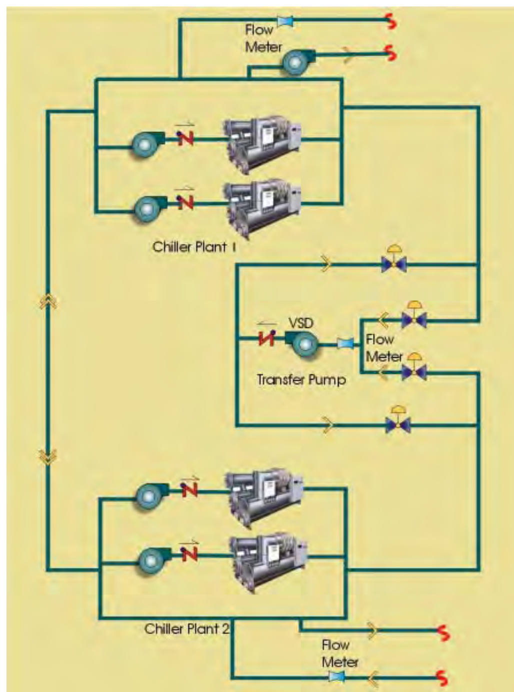


Figure 4-19 shows a typical piping arrangement for connecting several chiller plants from the secondary pump loop. In this diagram several buildings have chillers and one does not have a chiller. In this method, the differential pressure sensors located in the remote piping of each loop are used to control 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.

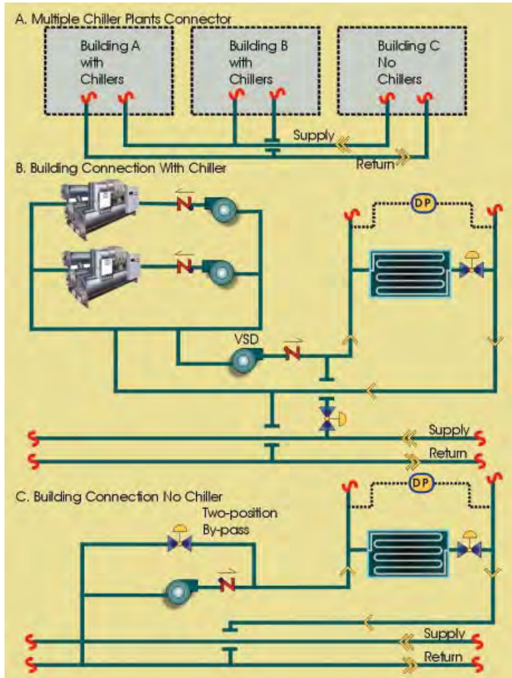


Figure 4-19: Connecting Multiple Chiller Plants on Secondary Side

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

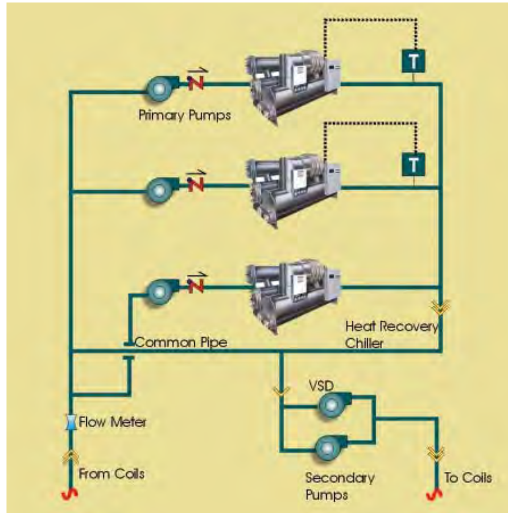
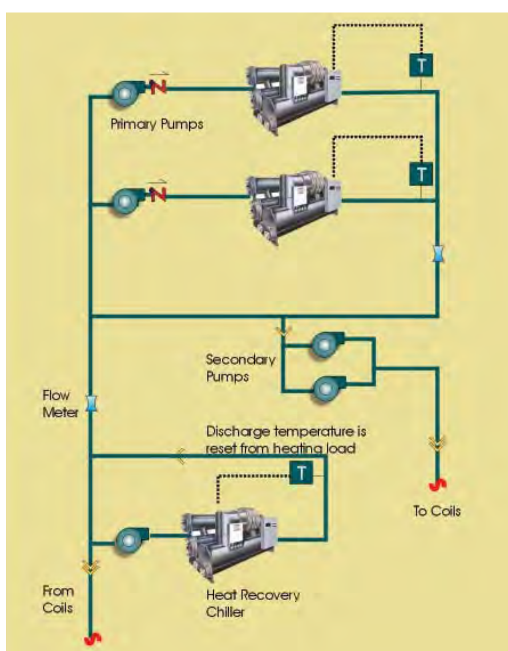


Figure 4-21 shows a configuration for a heat recovery machine that essentially puts the machine in series with the remaining chillers in the plant. Warm return water is pumped to the chiller then back into the return, thereby precooling the inlet water to the other chillers. The heat recovery machine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

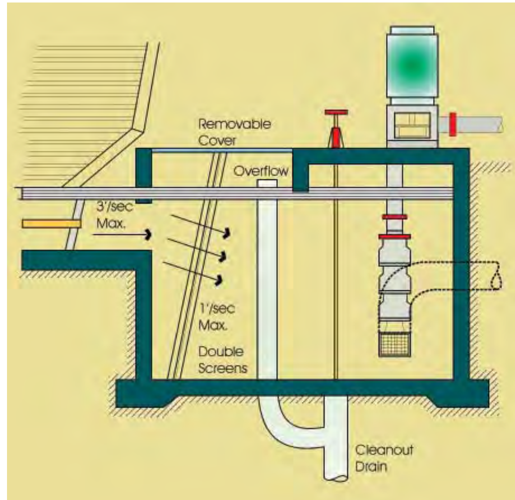
Case 2. Consider the same conditions as Example 1 except that the tower is located at 3,000 ft. The value of Ha for 3,000 ft is 30.5 ft, which is taken from a look-up table. The NPSHA is 22.6 ft as calculated below. There will still be no cavitation.

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

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.

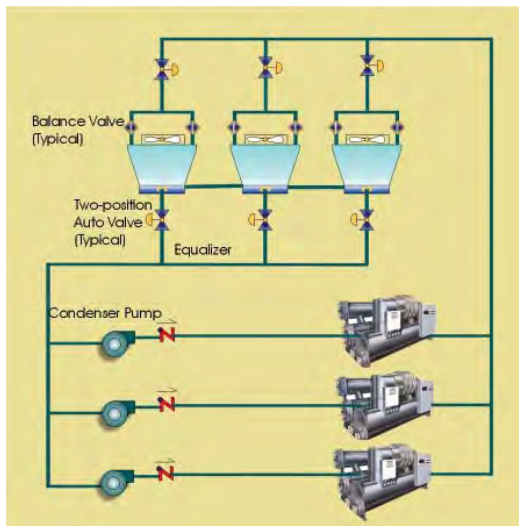
Another condition common at start-up is that the water temperature in the sump is too cold for the chiller. Chillers cannot operate at very cold inlet temperatures. When the machine requires warmer inlet temperatures than are available, three-way bypass valves are sometimes installed in the piping going 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.)

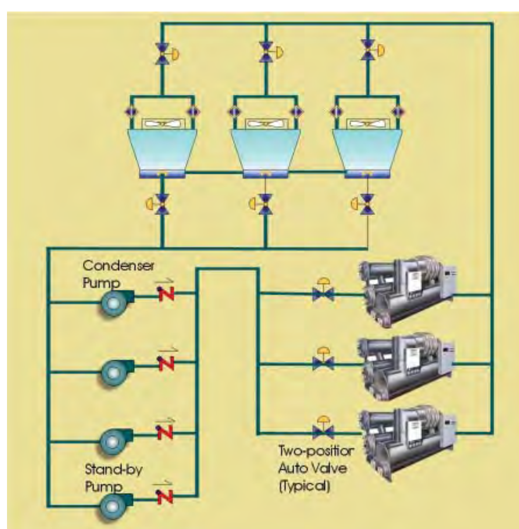
Since each condenser water pump shares the common piping to and from the cooling tower, the flow through the chiller will vary depending on how many pumps are operational. An analysis of pump curves and pressure drop characteristics may be necessary to determine if the flow variation is acceptable. This 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



Example 4-3:
Piping Pressure
Loss Calculation

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

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.

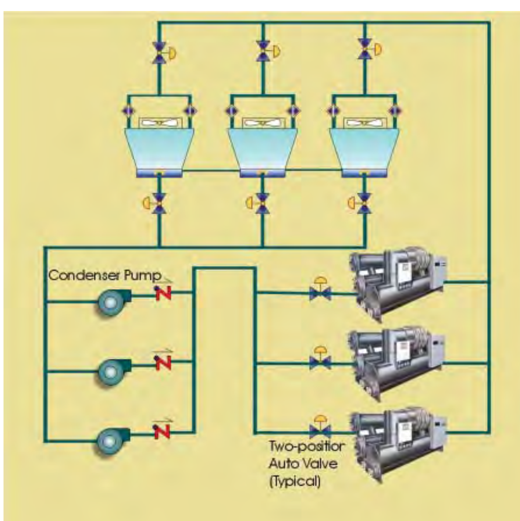
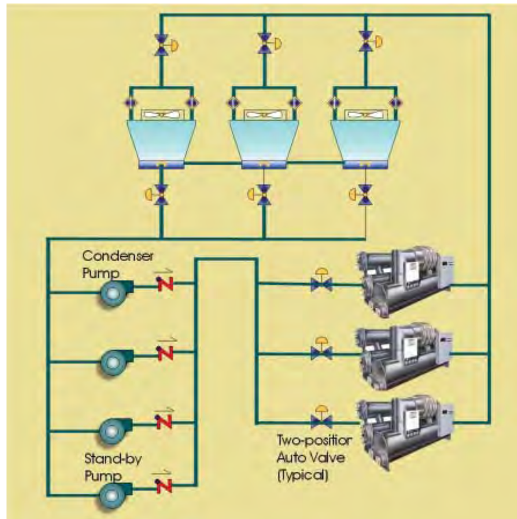


Figure 4-25:
Headered Condenser
Water Pumps

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

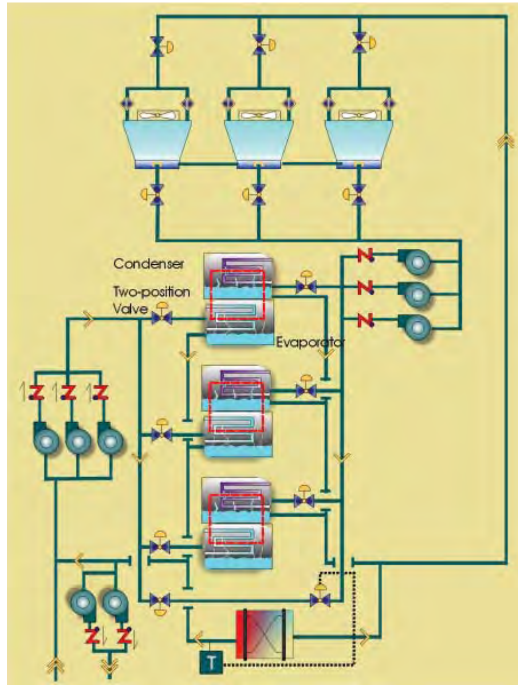
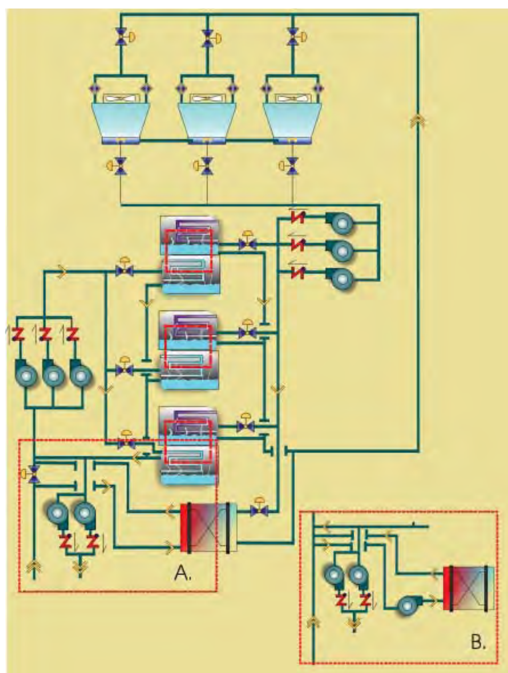


Figure 4-28:
Water Side Economizer,
Integrated



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.

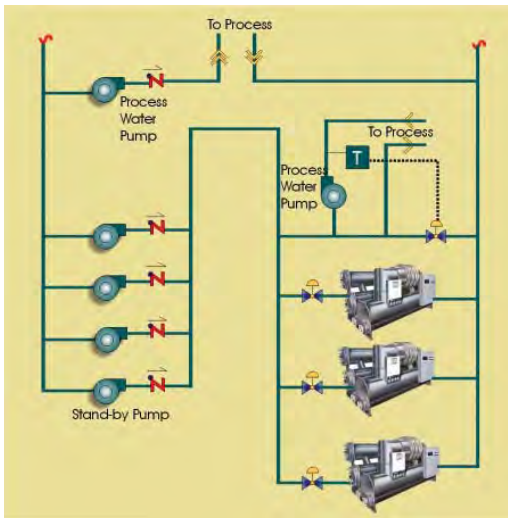


Figure 4-29: Auxiliary Condenser Water Loop

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.

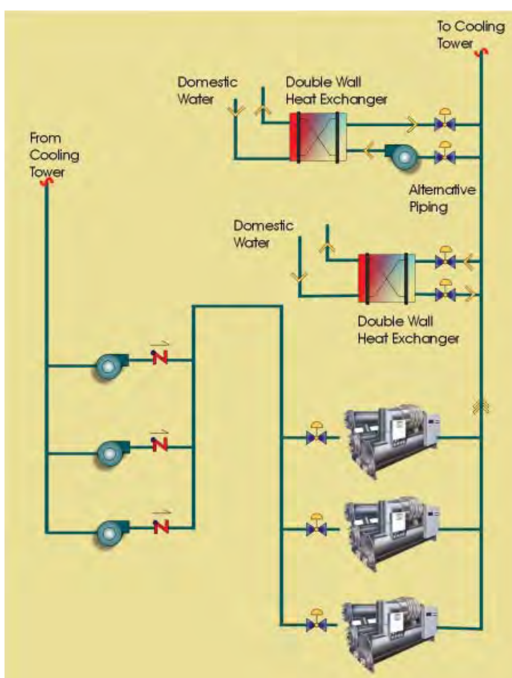
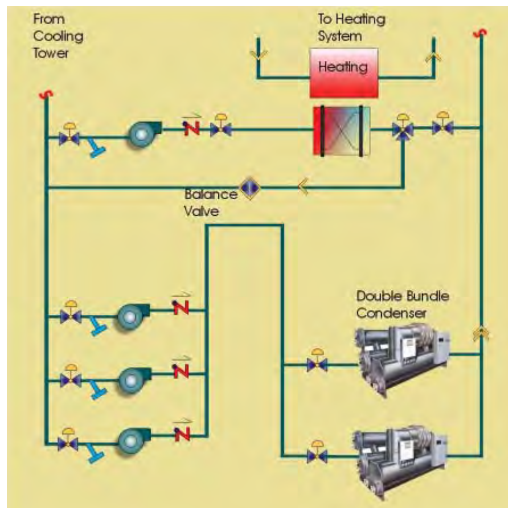


Figure 4-30: Preheat of Domestic Hot Water

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 course.

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	±1ºF
Chilled and Condenser Water Temperature at Central Plant Mains	±0.2ºF
Chilled and Condenser Water Temperature Elsewhere	±0.5ºF
Water Delta-T (supply to return)	±0.15ºF
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. 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. 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.

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.

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 course). 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.

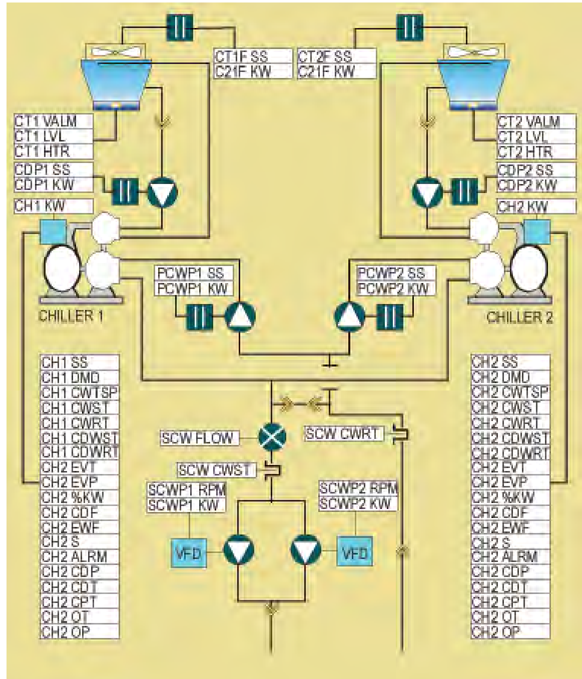


Figure 5-1: Typical Monitoring and Control Point Configuration

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.

1. **What is the primary reason for incorporating economizers into chilled water plant systems?**
    - Reduce hours of mechanical cooling
    - Increase chiller runtime
    - Reduce pipe size
    - Increase compressor speed
    -
2. **Which of the following is NOT a typical method for estimating peak loads?**
    - Site measurements
    - Computer simulations
    - Rules of thumb
    - Control valve calibration
    -
3. **The term "diversity" in load calculations refers to:**
    - Total peak load of all systems
    - Unused cooling capacity
    - Non-simultaneous peak of different load types
    - Equal load distribution across systems
    -
4. **Oversizing chillers without proper control can lead to:**
    - Reduced part-load efficiency
    - Lower construction cost
    - Elimination of peak demand charges
    - Better humidity control
    -
5. **What major environmental protocol influenced the phase-out of CFCs in chillers?**
    - Montreal Protocol
    - Kyoto Protocol
    - Paris Agreement
    - ASHRAE 90.1
    -
6. **Which compressor type uses magnetic bearings and is oil-free?**
    - Scroll
    - Twin screw
    - Reciprocating
    - Turbocor (centrifugal)
    -

7. **Which of the following contributes most to high part-load efficiency in chillers?**
   - Variable-speed drives (VSDs)
   - Hot gas bypass
   - Fixed-speed drives
   - Oversized cooling towers
   -
8. **In a primary-secondary system, the primary loop:**
   - Maintains constant flow through chillers
   - Adjusts to building load variations
   - Runs variable-speed coils
   - Only supports condenser water flow
   -
9. **What is the function of an inlet guide vane in a centrifugal chiller?**
   - Bypass condenser water
   - Control refrigerant flow into impeller
   - Reset chilled water temperature
   - Prevent freezing of tubes
   -
10. **Hot gas bypass is best used for:**
    - Maintaining operation at very low loads
    - Increasing compressor efficiency
    - Optimizing water temperature
    - Replacing economizers
    -
11. **The “lift” in a centrifugal chiller refers to:**
    - System pressure drop
    - Pressure difference between evaporator and condenser
    - Cooling tower head
    - Elevation of pump placement
    -
12. **In absorption chillers, lithium bromide serves as:**
    - Absorbent
    - Refrigerant
    - Heat source
    - Expansion medium
    -

13. Which component rejects heat from the refrigerant into the atmosphere or condenser water loop?

- Condenser
- Compressor
- Evaporator
- Expansion valve

14. One major advantage of a multiple-chiller configuration is:

- Improved part-load efficiency
- Simpler controls
- Lower equipment cost
- Eliminates need for VSDs

15. Oversized cooling towers with VFDs may improve energy performance by:

- Increasing flow rates
- Bypassing heat exchangers
- Reducing fan speeds at part load
- Operating only at full load

16. In commissioning, what does Level 1 typically involve?

- Equipment startup and basic functional tests
- System-wide operational validation
- Owner training
- Warranty reviews

17. Which control method is not energy efficient under low-load conditions?

- Cycling compressors
- Unloaders
- VSDs
- Hot gas bypass

18. What is a major disadvantage of undersizing a chiller plant?

- Higher first cost
- Inability to meet comfort or process loads
- Lower energy consumption
- Lower equipment redundancy

19. Which is a benefit of proper pipe sizing in chilled water systems?

- Reduced pumping energy
- Higher pressure drop
- Improved refrigerant flow
- Faster commissioning
-

20. What is the main function of a heat exchanger in the refrigeration cycle?

- Absorb or release heat to change refrigerant state
- Compress refrigerant
- Reject refrigerant to atmosphere
- Control pressure