# DISTRICT **COOLING** GUIDE

SECOND EDITION

ASHRAE

**Complete Reference**

Planning & System Selection • Central Plants • Distribution Systems • Thermal Storage • System O&M • End User Interface

# District Cooling Guide

Second Edition

The first edition of this publication was developed as a result of ASHRAE Research Project RP-1267, under the auspices of ASHRAE Technical Committee 6.2, District Energy, and Special Project 97. It is not a consensus document.

Supplemental funding for the first edition and funding for the second edition of this publication was generously sponsored by **Empower Energy Solutions, Dubai, United Arab Emirates**.

## REVISERS OF THE SECOND EDITION

**Gary Phetteplace, PhD, PE**
(Principal Investigator and Reviser of all Chapters and Appendices)
GWA Research LLC
Lyme, NH

**John Andrepont**
(Chapter 6)
The Cool Solutions Company
Lisle, IL

**Brian P. Kirk**
(Chapter 8)
Consultant
Tuckahoe, NY

**Steve Tredinnick, PE, CEM**
(Chapters 1, 2, 3, 5)
Burns & McDonnell Eng. Co., Inc.
Downers Grove, IL

## REVIEWERS OF THE SECOND EDITION

**Tariq Alyasi**
Empower Energy Solutions
Dubai, United Arab Emirates

**Samer Khoudeir**
Empower Energy Solutions
Dubai, United Arab Emirates

**Alaa A. Olama**
Independent Consultant
Cairo, Egypt

**Hassan Younes**
Griffin Project Development Consultants
Dubai, United Arab Emirates

## CONTRIBUTORS TO THE FIRST EDITION

**Gary Phetteplace, PhD, PE (Principal Investigator)**
(Chapters 1, 2, 3, 4, 5, 8, 9)
GWA Research LLC
Lyme, NH

**Salah Abdullah**
(Chapter 7)
Allied Consultants Ltd.
Laselky, Maadi, Cairo

**John Andrepont**
(Chapter 6)
The Cool Solutions Company
Lisle, IL

**Donald Bahnfleth, PE**
(Chapter 2)
Bahnfleth Group Advisors LLC
Cincinnati, OH

**Ahmed A. Ghani**
(Chapters 3, 7)
Allied Consultants Ltd.
Laselky, Maadi, Cairo

**Vernon Meyer, PE**
(Chapter 4)
Heat Distribution Solutions
Omaha, NE

**Steve Tredinnick, PE, CEM**
(Chapters 2, 3, 5)
(Volunteer contributor)
Syska Hennessy Group, Inc.
Verona, WI

## PROJECT MONITORING COMMITTEE FOR THE FIRST EDITION

**Steve Tredinnick, PE, CEM (Chair)**
Syska Hennessy Group, Inc.
Verona, WI

**Moustapha Assayed**
Empower Energy Solutions
Dubai, United Arab Emirates

**Samer Khoudeir**
Empower Energy Solutions
Dubai, United Arab Emirates

**Lucas Hyman, PE**
Goss Engineering, Inc.
Corona, CA

**Victor Penar, PE**
Hanover Park, IL

**David W. Wade, PE**
RDA Engineering, Inc.
Marietta, GA

## EX-OFFICIO REVIEWER FOR THE FIRST EDITION

**Kevin Rafferty, PE**
Modoc Point Engineering
Klamath Falls, OR

**Updates and errata for this publication will be posted on the ASHRAE website at www.ashrae.org/publicationupdates.**

# District Cooling Guide

Second Edition

ASHRAE®

Atlanta

ISBN 978-1-947192-15-7 (paperback)
ISBN 978-1-947192-16-4 (PDF)


1791 Tullie Circle, NE
Atlanta, GA 30329
www.ashrae.org


First edition cover design by Laura Haass



---

Library of Congress Cataloging-in-Publication Data

Names: American Society of Heating, Refrigerating and Air-Conditioning Engineers.
Title: District cooling guide.
Description: Second edition. | Atlanta, GA : ASHRAE, [2019] | Includes bibliographical references.
Identifiers: LCCN 2019010443| ISBN 9781947192157 (paperback) | ISBN 9781947192164 (pdf)
Subjects: LCSH: Air conditioning from central stations--Handbooks, manuals, etc.
Classification: LCC TH7687.75 .D47 2019 | DDC 697.9/3--dc23 LC record available at https://lccn.loc.gov/2019010443

---

**ASHRAE STAFF**

**SPECIAL PUBLICATIONS**

**Cindy Sheffield Michaels**, Editor
**James Madison Walker**, Managing Editor of Standards
**Lauren Ramsdell**, Associate Editor
**Mary Bolton**, Editorial Assistant
**Michshell Phillips**, Editorial Coordinator

**PUBLISHING SERVICES**

**David Soltis**, Group Manager of Publishing Services
**Jayne Jackson**, Publication Traffic Administrator

**DIRECTOR OF PUBLICATIONS AND EDUCATION**

**Mark Owen**

# Contents

## *Chapter 3 · Central Plant*

## Chapter 4 · Distribution Systems



## Chapter 5 · End User Interface

## Chapter 6 · Thermal Energy Storage



## Chapter 7 · Instrumentation and Controls

## *Chapter 8 · Operations and Maintenance*



## *Appendix A · Heat Transfer at the Ground's Surface and Subsurface Temperatures*



## *Appendix B · Case Studies*

# Acknowledgments


## FOR THE SECOND EDITION

District cooling (DC) continues to see increased interest worldwide, and significant developments have occurred since the publication of the first edition of the *District Cooling Guide* just over five years ago. This second edition of the *District Cooling Guide* expands on all areas of coverage contained in the first edition.

To capture as much new information and technology as possible at the outset of this project, a workshop to gain input was held in Sharm El-Sheikh, Egypt on Septmember 23, 2017. ASHRAE organized this workshop in cooperation with the United Nations Environment Program and the United Nations Industrial Development Organization during the International Conference on District Energy for Urban Development. This location was chosen to gather as much information as possible from engineers in the Middle East and North Africa region, where the greatest growth in DC is occurring and where all cooling technologies face special challenges due to high ambient temperatures (HAT). Over 40 individuals were in attendance at this workshop and significant input was received regarding all areas of DC system design and operations. This input served as our springboard into the revision of the first edition of this guide, and our team of six well qualified reviewers kept us on track and improved the publication in many ways with their input.

As the principal investigator for the project to update this guide, I would like to first thank ASHRAE for allowing my team the opportunity to undertake this effort and Empower Energy Solutions (Ahmad M. Bin Shafar, CEO) for sponsoring the effort. Special thanks go to W. Stephen Comstock of ASHRAE for getting all the parties together and the project in motion, and to all of the ASHRAE staff involved in the editing and publications process. I would also like to express my gratitude to my team members and subcontractors who added content and expertise in their respective areas of specialty.

And finally, thanks to my loving wife Karen Phetteplace, who not only put up with my long and unscripted hours in my office, but also proofread the entire document.

Gary Phetteplace
March 2019

## FOR THE FIRST EDITION

The principal investigator and authors would like to thank the members of the Project Monitoring Subcommittee (PMS) for their patience through the long process of creating

this document. This includes many discussions of scope and content as well as the actual review. This document has benefited tremendously from the careful review of the PMS members and their many suggestions based on the vast and diverse knowledge of district cooling that their composite experience represents.

The chair of the PMS, Steve Tredinnick, deserves special recognition for the countless hours he has invested in this effort both in his role as the PMS chair and as a major unpaid contributor and a sounding board for the principal investigator.

Gary Phetteplace
January 2013

# Acronyms

| | |
|---|---|
| AHRI | Air-Conditioning, Heating, and Refrigeration Institute |
| BMS | building management system |
| CEN | European Committee for Standardization |
| CFD | computational fluid dynamics |
| CHP | combined heat & power |
| CHW | chilled water |
| CHWS | chilled-water supply |
| CHWR | chilled-water return |
| COD | chemical oxygen demand |
| COWS | central operator workstation |
| CPVC | chlorinated polyvinylchloride |
| CT | cooling tower |
| COP | coefficient of performance |
| CS | constant speed |
| CTI | Cooling Tower Institute |
| CUP | central utility plant |
| DC | district cooling |
| DCP | district cooling plant |
| DCS | district cooling system |
| DPS | distribution piping system |
| DSM | demand-side management |
| EPRI | Electric Power Research Institute |
| EMS | energy monitoring/control system |
| ETS | energy transfer station |
| EOR | engineer of record |
| FM | factory mutual |
| FRP | fiberglass-reinforced plastic |
| HDPE | high-density polyethylene |
| HMI | human-machine interface |
| IEA | International Energy Agency |
| IDEA | International District Energy Association |
| ICMS | integrated control and monitoring system |
| LTF | low temperature fluid |

| | |
|---|---|
| MEP | mechanical, electrical, and plumbing |
| MF | micron filter |
| NIST | National Institute of Standards and Technology |
| NOVEM | Netherlands Agency for Energy and Environment |
| PVC | polyvinylchloride |
| PICV | pressure independent control valve |
| PSV | pressure sustaining valve |
| PVC | polyvinyl chloride |
| RTD | resistive temperature detector |
| TES | thermal energy storage |
| VFD | variable frequency drive |
| VS | variable speed |

# 1 Introduction

## PURPOSE AND SCOPE

The purpose of this guide is to provide guidance for all major aspects of district cooling system (DCS) design. The guidance is organized to be of use to both the inexperienced designer of DCSs as well as to provide a comprehensive reference to those immersed in the district cooling industry. In addition to design guidance, information on operations and maintenance (O&M) has also been included.

## DISTRICT COOLING SYSTEM COMPONENTS

DCSs consist of three primary components: the central plant(s), the distribution network, and the consumer systems or customer's interconnection (i.e., energy transfer station [ETS]); see Figure 1.1. In the central plant (see Chapter 3), chilled water is produced by one or more of the following methods:

- Absorption refrigeration machines (i.e., thermal energy driven)
- Electric-driven compression refrigeration equipment (reciprocating, rotary screw, or centrifugal chillers)
- Gas/steam turbine or engine-driven compression refrigeration equipment
- Combination of mechanically driven systems and thermal-energy driven absorption systems

The choice of refrigeration method and equipment will be driven by many factors, including cost and availability of fuels, electric system capacity, electricity cost, availability of water and alternatives, cogeneration potential, as well as obvious factors such as scale.

The second component is the distribution or piping network that conveys the chilled water to the buildings served (see Chapter 4). The piping may be the most expensive portion of a DCS. Chilled-water piping usually consists of uninsulated or preinsulated directly buried systems. These networks require substantial permitting and coordinating with nonusers of the system for right-of-way if the networks are not on the owner's property. Because the initial cost is high, it is important to maximize the use of the distribution piping network. Where future loads are expected, planning is extremely important as modifications to the buried piping systems are essentially impossible once initially constructed (see Chapter 2).

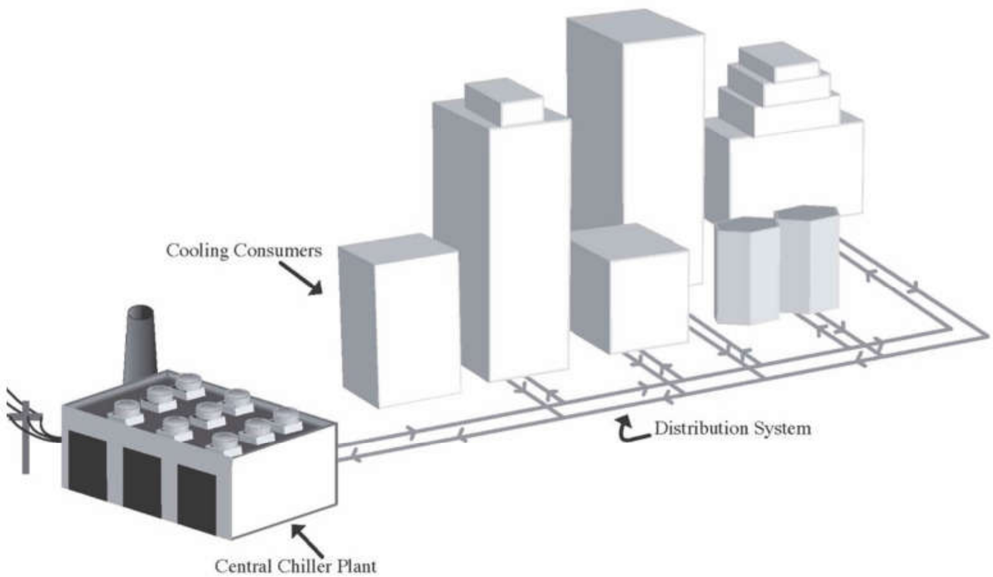



**Figure 1.1** District cooling system.

The third component is the consumer interconnection to the district cooling distribution system, which includes in-building equipment. Chilled water may be used directly by the building systems or isolated indirectly by a heat exchanger (see Chapter 5). While conversion of an existing building with cooling systems to cooling provided by a DCS is possible, extra care will be required in the design of the conversion, just as it would be necessary for the in-building equipment for a building being designed from the outset for connection to a DCS. Improper design in either case can have drastic consequences on the performance of the entire DCS, notably the dreaded and often encountered "low $\Delta T$ syndrome" (see Chapters 2 and 5).

## DISTRICT COOLING HISTORY AND CURRENT STATUS

District cooling (DC) normally distributes thermal energy in the form of chilled water from a central source to residential, commercial, institutional, and/or industrial consumers for use in space cooling and dehumidification. Thus, cooling effect comes from a distribution medium rather than being generated on site at each facility.

Whether the system is a public utility or user owned, such as a multibuilding campus, it has economic and environmental benefits depending largely on the particular application. Political feasibility must be considered, particularly if a municipality or governmental body is considering a DC installation. Historically, successful DCSs have had the political backing and support of the community. In some societies, the government may, for environmental or energy supply reasons, dictate, or through regulation strongly encourage, the use of DCSs owing to their many advantages that will discussed in this and later chapters.

Early attempts at DC date back to the 1880s (Pierce 1994). By the 1930s commercial systems were being built (Pierce 1994). The development of DC was confined mostly to North America—primarily the United States—for a number of decades. However, in recent years there has been increased activity outside of North America, notably in the

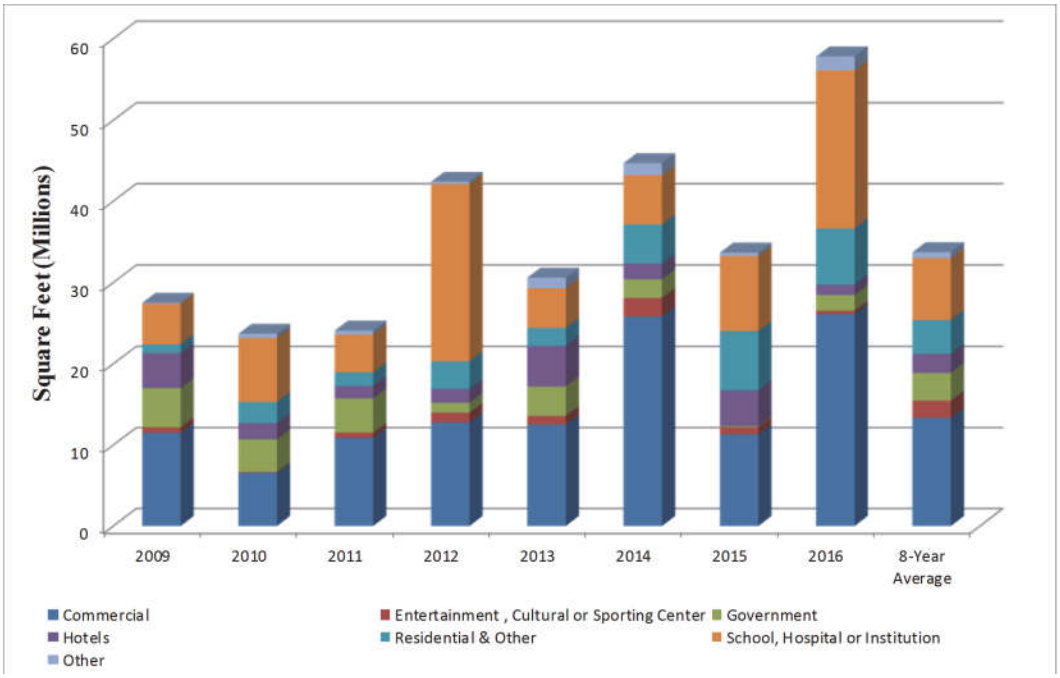



**Figure 1.2** Combined district heating and cooling growth in North America in annual added building area.
*Data from the International District Energy Association*

Middle East, Indonesia (Singapore), and Europe. The International District Energy Association (IDEA), which represents both heating and cooling utilities, reports that in 2016 approximately 65% of the conditioned building space added by its members was added outside of North America. All of that growth was DC in the Middle East and Singapore (IDEA 2017). The size of the buildings added to DCSs as reported by IDEA (2017) is 150% greater on average in terms of area for buildings added by their members outside of North America, despite the fact that DCSs see little use in the residential sector in North America compared to the Middle East where it is more common. Tredinnick et al. (2015) provides breakdowns of the sectors where district energy systems are showing the most growth, based on data from IDEA (2017), Figures 1.2 and 1.3 are updated versions of those from Tredinnick et al. (2015). By examining Figures 1.2 and 1.3, it can be seen that within North America the most growth is seen in the commercial sector and the combined sector of university and heath care, while outside of North America the greatest growth was in the commercial and residential sectors. It should be noted that the data upon which Figures 1.2 and 1.3 are based are comprised only of that reported by the membership of IDEA, which includes few members from Europe and Asia. Werner (2017) estimates that there are 150 DCSs operating in Europe. In terms of actual deliveries of cooling effect, Werner (2017) provides useful estimates for context, estimating that around 300 PJ per year are delivered worldwide. Of that, Werner (2017) estimates that about 200 PJ are delivered in the Middle East, 80 PJ in the US, 14 PJ in Japan, and 10 PJ in Europe.

Growth in air-conditioning, already representing 20% of total electric use in buildings, is expected to increase rapidly due to the economic and demographic growth in the hotter regions of the world (IEA 2018). Increased efficiency of the air-conditioning process is essential to curb the expected demand growth, and DCSs can play a large role in achieving increased efficiency. DCSs are especially applicable where cooling is first

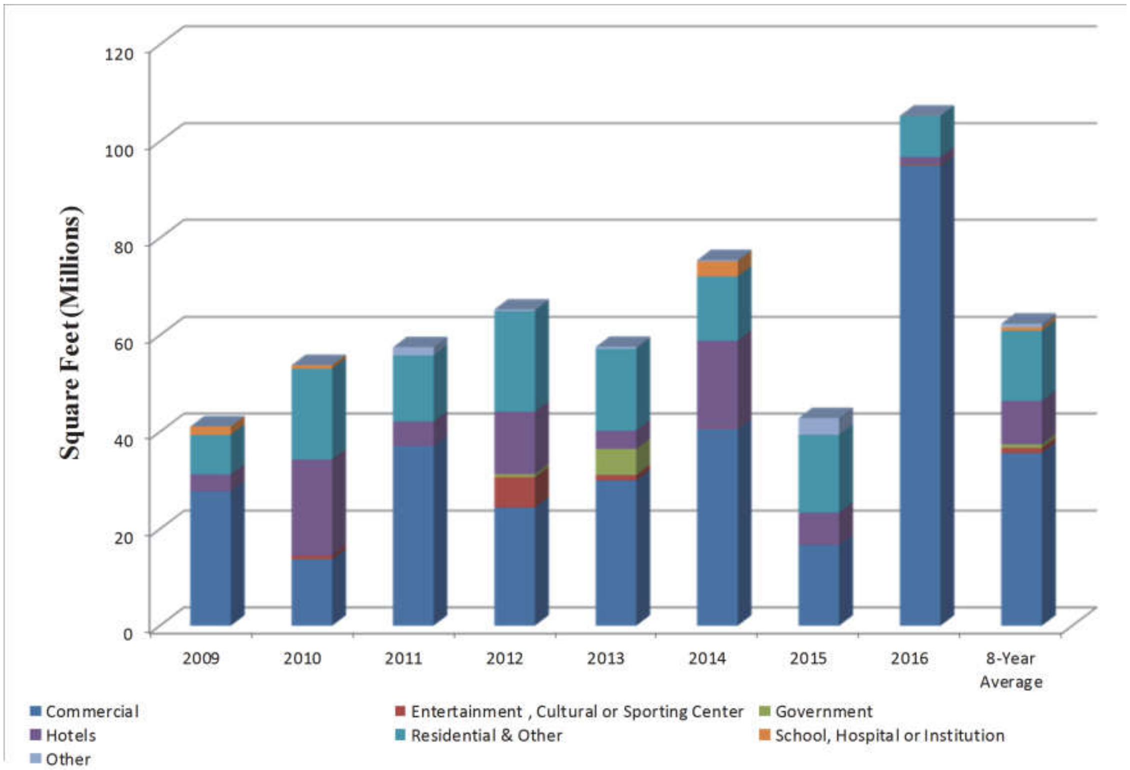



**Figure 1.3** Combined district heating and cooling growth outside of North America in annual added building area.
*Data from the International District Energy Association*

being introduced due to their capital-intensive nature. Without increases in air-conditioning efficiency, energy demand from air-conditioners will more than triple by 2050 with an attendance increase in electrical demand to a level equivalent with China's current total demand for electricity (IEA 2018).

Global warming can be expected to result in increased demand for space cooling, even with reductions in $CO_2$ emissions. IEA (2012) provides several scenarios for expected average global temperature rise by 2050 with what they call the 6°C Scenario (6DS) being essentially a continuation on our current path, including pledges for reductions that have already been made. A more aggressive scenario that would cut global energy related $CO_2$ by more than 50% from 2009 levels is referred to as the 2°C Scenario (2DS). While, under this scenario, IEA (2014) predicts energy efficiency measures would reduce cooling energy consumption from the business-as-usual case, there would still be growth largely driven by nations where comfort cooling is not currently in widespread use. Specifically, IEA (2014) estimates combined growth from 2010 levels of less than 1 EJ to 6 EJ by 2050 for China, India, Latin America, and the Association of Southeast Asian Nations (ASEAN), which includes Indonesia, Malaysia, the Philippines, Singapore, and Thailand. DC is posed to play a large role both in these countries as well as those counties where comfort cooling is more common place but higher efficiencies must be sought. The United Nations Environment Programme (UNEP 2015) has endorsed district energy (district heating and district cooling) for its potential role in achieving energy efficiency and renewable energy use in cities.

## APPLICABILITY

DCSs are best used in markets where the thermal load density is high and the number of equivalent full load hours of cooling (or operating hours) is high. A high load density is needed to cover the capital investment for the transmission and distribution system, which usually constitutes a significant portion of the capital cost for the overall system, often amounting to 50% or more of the total cost. This makes DCSs most attractive in serving densely populated urban areas and high-density building clusters with high thermal loads, especially tall buildings. Urban settings where real estate is very valuable are good places for DCSs since they allow building owners to make maximum use of their footprint by moving most of the cooling equipment off-site. Low-density residential areas have usually not been attractive markets for DC. The equivalent full load hours of cooling are important because the DCS is capital intensive, and maximum use of the equipment is necessary for cost recovery. A DCS is a good choice where high reliability of cooling supply is needed, such as health care facilities, elderly housing, and data centers. DCSs have also found wide acceptance in moderate load density institution settings, such as colleges, universities, and prison complexes where they are favored for their high reliability, improved efficiency, and reduction in staffing requirements as opposed to operating many in-building cooling plants. DCSs, along with their district heating system counterparts, have long been favored solutions for college campuses, and that trend continues today with both expansion and adoption of nonconventional technologies (Nolder and Pollard 2018).

DC has seen a wide variety of applications, and the reader is referred to IDEA (2017) for additional examples. These applications span all major sectors of the building market: residential, commercial, institutional, and industrial. For many of the applications, such as college campuses and military bases, the loads are captive. At the opposite end of the spectrum are DCSs that operate as commercial enterprises in urban areas competing with in-building equipment for cooling loads. Between these two extremes are many other business models, such as DC providers, who operate under contract, or a plant owned by the developer of a real estate project. For information on business models and business development for DC enterprises, see IDEA (2008).

## BENEFITS

### Environmental Benefits

Generating chilled water in a central plant is normally more efficient than using in-building equipment (i.e., decentralized approach) as discussed below, and thus the environmental impacts are normally reduced. The greater efficiencies arise primarily due to the larger, more efficient equipment and the ability to stage that equipment to closely match the load yet remain within the equipment's range of highest efficiency. DCSs may take advantage of diversity of demand across all users in the system and may also implement technologies such as thermal storage more readily than individual building cooling systems. For electric-driven district cooling plants, higher efficiency becomes the central environmental benefit since in-building plants are normally electric driven as well. There may be additional environmental benefits from cooling supplied from a large central plant, such as the ability to use treated sewage effluent as cooling tower makeup water, options that would not be feasible for in-building systems. Other environmental advantages may come from the ability to use sea water or fresh surface water for heat rejection directly—again, an option that is likely not feasible on a single building scale. Furthermore, in some instances, deep sea water or deep fresh surface water may be cold enough to provide cooling directly without the use of refrigeration equipment.

Environmental benefits will normally also be realized because of the increased efficiency that comes with vigilant maintenance and operational optimization realized when the cooling source is operated as a business itself rather than an often neglected and minor ancillary to the core business.

The ability to handle refrigerants in a safer and more controlled environment is a major environmental advantage of DCSs. Beginning with the Montreal Protocol in 1987, the use of chlorofluorocarbons (CFCs) and hydrochlorofluorocarbons (HCFC) refrigerants found to be harmful to the environment has become increasingly regulated. This process began in developed countries and is being followed on a delayed schedule in developing countries. Refrigeration working fluids have since been in a constant state of flux due to the preponderance of equipment already in use as well as the lack of readily available alternatives. Phase out of the most harmful compounds has largely been by replacement, using less harmful, but by no means harmless, refrigerants. For users of cooling equipment, this has meant additional requirements in terms of handling of existing materials. Additionally, the decision on how to proceed when equipment replacement is needed and multiple options exist can be very confusing to all except those totally immersed in the refrigeration/refrigerants industry. All these requirements that may be onerous on a single building owner are much better addressed by DCS operators who are immersed in the industry and the issues of refrigerant use in this era of great flux. Chapters 2 and 3 provide additional discussion on these refrigerant issues.

Waste heat from industrial processes or electric power generation along with geothermal energy are potential energy sources for heat-driven DCSs (See Chapters 2 and 3), which are much easier to use on the scale of DCSs as opposed to smaller in-building equipment. When fuels are burned to generate cooling via absorption or gas/steam turbine and/or engine-driven chillers, emissions from central plants are easier to control than those from individual plants, and on an aggregate generate less pollutants due to higher quality of equipment, higher seasonal efficiencies, and higher level of maintenance. A central plant that burns high-sulfur coal can economically remove noxious sulfur emissions, where individual combustors could not. Similarly, the thermal energy from municipal wastes can provide an environmentally sound system, an option not likely to be available on a building scale system.

Refrigerants and other chemicals can be monitored and controlled more readily in a central plant. Where site conditions allow, remote location of the plant reduces many of the concerns with the use of ammonia systems for cooling.

## Economic Benefits to Building Owners

A DCS offers many economic benefits. Even though the basic costs are still borne by the central plant owner/operator, because the central plant is large, the customer can realize benefits of economies of scale.

### Operating Personnel

One of the primary advantages for a building owner is that operating personnel for the HVAC system can be reduced or eliminated. Most municipal codes require operating engineers to be on site when high-pressure boilers, as would be used to drive absorption chillers, are in operation. Some older systems require trained operating personnel to be in the boiler/mechanical room at all times. When chilled water is brought into the building as a utility, depending on the sophistication of the building HVAC controls, there will likely be opportunity to reduce or eliminate operating personnel.

### Insurance

Both property and liability insurance costs may be significantly reduced with the elimination of boilers, chillers, pumps, and electrical switch gear from within the building because risk of a fire or accident is reduced.

### Usable Space and Reduced Noise

Usable space in the building increases when a boiler and/or chiller and related equipment are no longer necessary. The noise and vibration associated with such in-building equipment is also eliminated. In retrofit applications, this space cannot usually be converted into prime office space, however it does provide the opportunity for increased storage or other use.

### Equipment Maintenance

DCSs deliver cooling to a building as a "finished product," eliminating the need for problematic in-building cooling equipment, such as cooling towers and their attendant threat of *legionella* outbreaks. With less mechanical equipment, there is proportionately less equipment maintenance, resulting in less expense and a reduced maintenance staff. The need to carry an inventory of spare parts is greatly reduced.

### High Reliability

DCSs provide unparalleled reliability of supply, owing to equipment redundancy and the high level of operational supervision and maintenance that is prevalent in DCSs. On average, IDEA (2008) reports that DCSs have reliability exceeding 99.94%.

### Higher Efficiency

A larger central plant can achieve higher thermal and emission efficiencies than several smaller units. When strict regulations must be met for emissions, water consumption, etc., control equipment is also more economical for larger plants. Partial load performance of central plants may be more efficient than that of many isolated small systems because the larger plant can modulate output and operate one or more capacity modules as the combined load requires. While a recent study (Thornton et al. 2008) found that actual operating data on in-building cooling plants is scarce, the limited data the study uncovered indicate that in-building systems were operating at an average efficiency of 1.2 kW/ton (2.9 COP). Another study (Erpelding 2007) found that central district cooling plants can have efficiencies of 0.85 kW/ton (4.1 COP) under less than optimal design/operation. Thus, the efficiency of chilled-water generation in a central plant is approximately 40% greater than an in-building chiller plant. Others (IDEA 2008) have suggested much greater efficiency improvements relative to air-cooled, in-building cooling systems, i.e., approximately 1.65 kW/ton (2.1 COP) for air-cooled in-building systems versus approximately 0.70 kW/ton (5.0 COP) for electric-driven district cooling with thermal storage, an increase in efficiency of nearly 140% for district cooling.

Similarly, typically industrial-based controls systems are used that offer a higher level of controls and monitoring of the overall system and efficiency as compared to commercial decentralized cooling systems. Furthermore, with the higher level of monitoring comes additional scrutiny in operations to optimize the system performance and efficiency resulting in reduced operating costs.

Additional gains in overall delivered cooling efficiencies are also possible when thermal energy storage (see Chapter 6) is implemented with district cooling due to the ability to run cooling equipment at higher loads during nighttime hours when conditions are more favorable for heat rejection. As an energy storage method for short-term load shifting, thermal energy storage currently is more economically favorable than energy storage by other means such as pumped hydro, compressed air, or batteries as discussed in more detail in Chapter 6. Thermal energy storage (TES) is a good example of a technology that is more feasible on the large scale prevalent in a DCS.

### Available Primary Energy

While on a building scale it may not be practical to generate chilled water via absorption chillers, this is possible in larger central chiller plants, and larger central absorption chiller plants may even use fuels such as coal or refuse or multiple fuels. The use of high-voltage chillers may also be impractical from all but the largest in-building chiller plants.

## REFERENCES

Erpelding, B. 2007. Real efficiencies of central plants. *Heating/Piping/Air-Conditioning Engineering*: May 1, 2007.

IDEA. 2008. *District cooling best practices guide*. Westborough, MA: International District Energy Association.

IDEA. 2017. *District energy space 2016*. Westborough, MA: International District Energy Association.

IEA. 2012. *Energy technology perspectives 2012*. Paris: International Energy Agency.

IEA. 2014. *Heating without global warming, market development and policy considerations for renewable heat*. Paris: International Energy Agency.

IEA. 2018. *The future of cooling, opportunities for energy-efficient air conditioning*, Paris: International Energy Agency.

Nolder, E., and S. Pollard. 2018. Underneath the ivy: Early adopters of district energy, Ivy League schools continue expanding and improving their underground distribution networks. *District Energy*: First Quarter. Westborough, MA: International District Energy Association.

Pierce, M. 1994. Competition and cooperation: The growth of district heating and cooling. *Proceedings of the International District Energy Association*, 1182–917.

Thornton, R., R. Miller, A. Robinson, and K. Gillespie. 2008. *Assessing the actual energy efficiency of building scale cooling systems*. Report 8DHC-08-04, Annex VIII. Paris: International Energy Agency, IEA District Heating and Cooling Program. www.iea-dhc.org.

Tredinnick, S., D. Wade, and G. Phetteplace. 2015. District energy enters the 21$^{st}$ century. *ASHRAE Journal* 57(7):48–56.

UNEP. 2015. District energy in cities, unlocking the potential for energy efficiency and renewable energy. Paris: United Nations Environmental Programme.

Werner. 2017. International review of district heating and cooling. *Energy* 137:617–31. Elsevier.

# 2 Alternative Development and System Planning

## INTRODUCTION

For buildings, the cost of utility services and infrastructure needed to provide these services is significant and may even exceed the cost of the buildings themselves over their lifetimes. Planning has the potential to reduce both the initial and future costs. The objective of planning should be to guide decision making such that cost savings in providing utilities over the life expectancy of the building(s)/campus/system are realized. Due to their capital intensive nature, the need for planning is significant for DCSs, and the current rate of growth of these systems has made system planning a topic of much current interest.

The term master plan will be used here, but there are several levels of planning. A master plan covers all levels of planning and is typically integrated with the planning of the development of both new and existing sites. Throughout this section, the development of utility master plans will typically be applicable to private, public, and utility owned systems. Differences in planning for the different systems will be identified whenever they are significant.

For totally new systems in a greenfield project, master planning is an essential first step to assuring that the owner's requirements are fulfilled in the delivered system. Appropriate planning will have significant impacts, not just on the first cost of the project, but also the future operations and maintenance (O&M) costs. The master plan will also provide valuable information to those responsible for the O&M of the system, and it is thus essential that the individuals who will be responsible for the O&M of the system be involved in the planning process. For sites with existing systems/other utilities, involvement of the system operators in the development of a master plan is essential; there is no substitute for corporate knowledge when dealing with buried systems. A plan properly prepared will also benefit the users served by the system, and it may be prudent to engage these representatives in the planning process. Potential stakeholders in the development of a master plan are:

- Building/campus/site owner
- Owner's project engineer
- Site master planners
- Utility system operators
- Operators/engineers of other utilities on site or within utility right-of-way
- Potential contractors

- District cooling customer (the building's users)
- Adjacent residential neighborhood associations and business enterprises

This list is by no means exhaustive; special circumstances could result in many different and varied stakeholders. The planning process should include everyone involved in the design, construction, operation, and use of the facilities provided with district cooling. Master utility plans for existing facilities also must consider correcting deficiencies built into them initially or during periods of rapid expansion. Utility plans for existing systems also must account for the aging of the existing infrastructure and must also consider appropriate timing for replacement and upgrading efforts. Furthermore, building functions can change over the life of the building, (i.e., college dorms converted to offices, etc.) therefore the service lines to the building and distribution mains must be of adequate size to accommodate future facility remodeling or expansion.

Planning is a difficult task that takes time and creative energy to explore the many variables that must be considered to develop a system that will operate in accordance with design intent and is: sustainable, reliable, energy efficient, environmentally friendly, supports expansion to serve new chilled-water (CHW) loads, and is easily operated and maintained. One additional variable, that may be the most important of all considerations, is the development of a concept-level budget estimate (opinion of probable cost). Due to poor preliminary cost estimates, many systems have to be modified with lower quality materials during the bidding process before construction begins. This ultimately is detrimental to the functionality, efficiency, and life expectancy of the system during its entire operational lifetime. Early planning and concept designs of adequate detail used to develop more accurate cost estimates can mitigate many of these issues. In this chapter, guidance will be given for developing a system master plan that is sustainable, meets the owner's performance needs and desires, and can be constructed within the owner's constraints for overall capital cost and cash flow.

A note of caution is required with regard to retaining an engineering firm to plan, design, supervise construction, and commission a DCS or major alterations/additions to an existing DCS. While the engineering of a DCS is not more difficult than many other similar design tasks, there are unique aspects that only an experienced designer of DCS will know. Use of this guide will help acquaint a designer new to DCS, but there is no substitute for experience. A detailed discussion on how to select an engineering firm for a combined heat and power (CHP) systems can be found in Meckler and Hyman (2010), which can also serve as a guide for selecting a design firm for a DCS.

What should a utility master plan provide an owner? At minimum, a utility master plan should provide a prioritized program for long term guidance for building, expanding, and upgrading the district systems, which are typically built incrementally. A good master plan serves as a technically sound marketing tool for the owner's engineers to present needs and solutions to management or to prospective customers. Unfortunately, many owners view utility master plans as an interesting technical exercise with a life of one or two years. When this has been an owner's experience, it usually results from one or more of the following reasons:

- Failure to involve the owner's staff
- Failure to provide intermediate owner reviews
- Use of an unverified database
- Lack of creativity in developing technically sound system alternatives for screening and final selection by the owner
- Inaccurate cost estimation, often related to overly optimistic estimation using unit costs that do not include all elements of the systems

The process for developing a utility master plan may be likened to a pyramid (Figure 2.1) (Bahnfleth 2004). The success of the plan depends on the foundation, a strong database that includes discovery and verification. The development of a strong and accurate database is the foundation on which all other aspects of a master plan stand and get their credibility. With the database in place, the identification of alternatives and the preliminary estimates of cost (screening grade) for each alternative are used to select (with the owner) the most promising alternatives. These alternatives are then subjected to more intense analysis before making the final decisions of how the new plant (or expansion of an existing plant) will be developed as well as how plans for future projects will be laid out. Thus, the pinnacle of the pyramid is a prioritized, priced list of projects needed to both keep pace with the physical growth of a facility and to provide replacements and upgrades to the existing system, if there is one.

## ESTABLISH AND CLARIFY OWNER'S SCOPE

Before the planning process can begin in earnest, as with every project, the owner's scope and expectations for the project must be fully clarified. While a scope may have been written in the preliminary stages of the owner's development of the project, often this scope will need clarification and refinement. Meeting with the owner, the staff who will inhabit the facilities, those performing the O&M, and those responsible for budgeting the project and its O&M costs will help clarify and refine the scope and ultimately can eliminate many potential issues and challenges later. In fact, such open meetings can bring new insights to the owner, sometimes resulting in a totally new scope being developed. During the initial scope review, other factors affecting the system planning can be determined through proper interaction with those present. Among the things that should be learned are the level of system quality expected, the budget and cash flow limitations, and the anticipated potential long-term expansion of the system. With the present emphasis on sustainability, this is an appropriate time to discuss the owner's interest and desires with respect to this important aspect of system design and development.

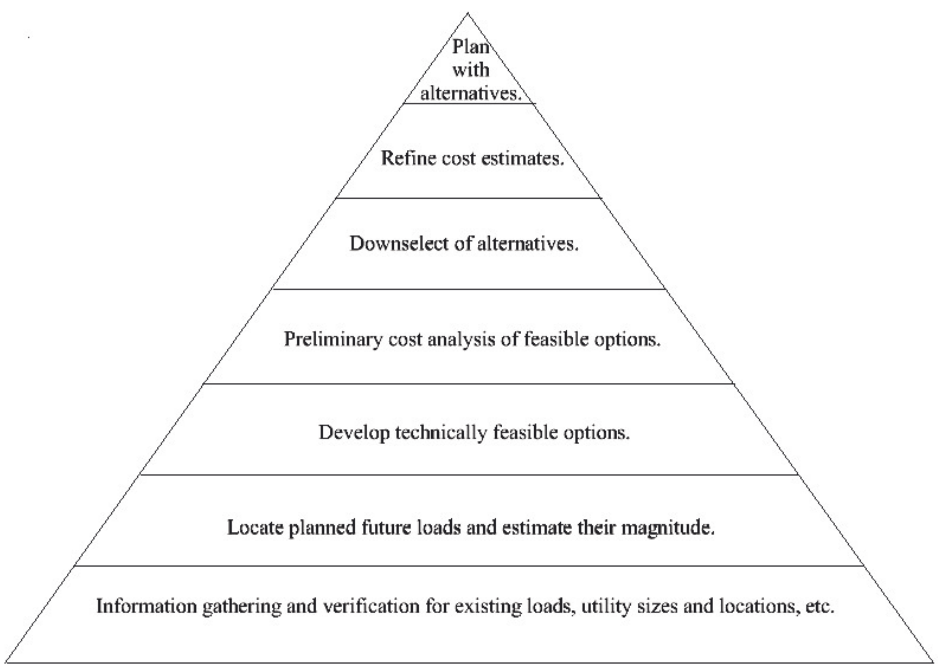



**Figure 2.1** The master planning pyramid.

Throughout the process of establishing and refining the scope, it is paramount that all parties recognize that every system is subject to three basic constraints: budget (owner's), scope limitations (jointly developed by the engineer and the owner), and quality versus cost (a champagne quality system cannot be built on a beer budget).

## DEVELOPMENT OF THE DATABASE

Because gathering information for developing a suitable system-planning database takes some time from the owner's staff who already have their regular jobs to do, it is helpful at the outset to provide a list of information and data that will be required to complete a good system plan. The list includes whatever cooling-load data exists, either in the form of estimated loads for new buildings to be served by the DCS or recorded-load data in existing systems. When considering historic cooling-load data it is particularly important to bear in mind the bias in recent years towards increased cooling requirements that have resulted from the combined impacts of increased ventilation rates; increasing electronics use in offices, dorms, and other spaces; as well as tightening of building envelopes by retrofit measures.

In existing systems, inventories of available equipment that will become a part of the new or expanded system must be obtained from the owner's records and/or by gathering data from available shop drawings and equipment nameplates. Additionally, condition assessments of equipment that may be reused in a new system should be conducted. Equipment condition can be found in maintenance records; code inspections, where required; and from operating personnel interviews. Site utility maps showing the locations of existing utilities are essential for planning distribution system routing to avoid interferences and to minimize the expense of installing tunnels, shallow trenches, or direct burial piping. The site electrical drawings will identify the location of existing electrical system feeders and available substations. A single-line electrical drawing showing system loads and feeder capacities among other things should be obtained when available. As the planning process proceeds, each of these databases will be expanded by the planning team, but early identification of the need for them can assist in meeting time deadlines established by the owner.

Estimating cooling loads to establish the CHW production capacity needed during the life ascribed to the master plan is one task that too often is complicated by attempts to develop computer-based load profiles for each and every building. But for large systems serving tens and even hundreds of buildings, experience indicates that computer-based load analyses of each building may not be necessary and can be very deceptive. It is an example of precision exceeding the accuracy needed, especially when the future addition of large numbers of minimally-defined buildings are to be added to the mix. Furthermore, too much credence can be given to loads calculated by a computer program without adequate scrutiny for reality. Good examples of such additions were new chemistry facilities in the 1990s and the large number of biological research buildings under construction today that have extremely high-load densities. The use of proper unit-load densities in square feet per ton for example, often provides estimates of adequate accuracy when consideration is given to the mix of buildings being served. Hyman (2010) provides a summary of the options for obtaining load data:

- Energy metering data from an energy monitoring/control system (EMS)
- Meter readings at the building or equipment level
- Analysis of utility bills
- Computer energy modeling; requires calibration for existing buildings
- Installed equipment capacity

- Load densities for capacity per unit area, i.e., unit area per ton

While all of these methods may be applicable to existing buildings, only computer modeling and the use of norms per unit area may be readily applied to planned future buildings. For preliminary planning, Table 2.1 provides unit area load data. Such data must be used with extreme caution given the variability of loads across facility types, or even within a given type of facility, due to factors such as occupancy, climate, building construction, etc.

Regardless of the method(s) used to establish individual building loads, it must be recognized that diversity, which is often a judgment call based upon experience, plays an important part in the process as well. A district energy system's diversity factor varies from 0.5% to 0.95% of summing the individual building peak loads. The diversity factor is directly dependent upon the number and character of the group of buildings being served, recognizing that different building functions will not peak at the exact same time of day. Establishing the correct diversity factor is more of an art than a science and its development should involve the system owner and operators so they are aware of the assumptions created. The diversity factor also does not have to be accurate to the third decimal point because accuracy to the last ton is not useful in accommodating future growth and is often misleading. If the district energy system already exists and each building or customer has an energy meter, then the diversity for a specific year or years can be calculated by determining the peak load day at the central plant and comparing it to when each building actually peaked during the season.

Finally, when determining plant peak capacity, it is also important to include the load that is placed on the chiller plant by heat gains or losses to the distribution system. Chapter 4 covers the calculation of heat gains from both insulated and uninsulated CHW distribution piping.

## ALTERNATIVE DEVELOPMENT

### Codes and Standards

Once loads have been established, both existing and planned, and all data on existing system(s), if any, and other utilities have been gathered, identification of alternatives can begin. Identifying potential sites for new central plant(s) will be the first task. This task should begin with a review of codes, standards, and regulations. Early review of voluntary and mandatory codes, standards, and regulations is a necessary step in planning DCSs that will preclude potential conflicts that could result in wasted effort and resources in planning and design. Codes include local and state codes applicable to the following:

- Construction of the central utility plant (CUP) or satellite plant
- Construction of piping systems above and below ground
- Introduction of loops in the distribution system to reduce system pressure drop and increase redundancy
- Determining limits of emissions
- Preparation of construction and operating permits
- Selection of equipment to meet limits on emissions, noise, wastewater quality, etc.
- System performance with emphasis on safety and energy efficiency as it relates to sustainability

**Table 2.1** Approximate Unit-Area Cooling-Load Values (ASHRAE 2009)

| Application | Occupancy | | | | | | Lighting | | | | | | Refrigeration Load[1] | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | $ft^2$/person | | | $m^2$/person | | | W/$ft^2$ | | | W/$m^2$ | | | $ft^2$/ton | | | $m^2$/kW | | |
| | Low | Avg. | High | Low | Avg. | High | Low | Avg. | High | Low | Avg. | High | Low | Avg. | High | Low | Avg. | High |
| Apartment, High Rise | 325 | 175 | 100 | 30.2 | 16.3 | 9.3 | 1 | 2 | 4 | 11 | 22 | 43 | 450 | 400 | 350 | 11.9 | 10.6 | 9.3 |
| Auditoriums, Churches, Theaters | 15 | 11 | 6 | 1.4 | 1.0 | 0.6 | 1 | 2 | 3 | 11 | 22 | 32 | 400 | 250 | 90 | 10.6 | 6.6 | 2.4 |
| Educational Facilities (Schools, Colleges, Universities) | 30 | 25 | 20 | 2.8 | 2.3 | 1.9 | 2 | 4 | 6 | 22 | 43 | 65 | 240 | 185 | 150 | 6.3 | 4.9 | 4.0 |
| Factories: *Assembly Areas* | 50 | 35 | 25 | 4.6 | 3.3 | 2.3 | 3[2] | 4.5[2] | 6[2] | 32[2] | 48[2] | 65[2] | 240 | 150 | 90 | 6.3 | 4.0 | 2.4 |
| *Light Manufacturing* | 200 | 150 | 100 | 18.6 | 13.9 | 9.3 | 9[2] | 10[2] | 12[2] | 97[2] | 108[2] | 129[2] | 200 | 150 | 100 | 5.3 | 4.0 | 2.6 |
| *Heavy Manufacturing*[3] | 300 | 250 | 200 | 27.9 | 23.2 | 18.6 | 15[2] | 45[2] | 60[2] | 161[2] | 484[2] | 646[2] | 100 | 80 | 60 | 2.6 | 2.1 | 1.6 |
| Hospitals: *Patient Rooms*[4] | 75 | 50 | 25 | 7.0 | 4.6 | 2.3 | 1 | 1.5 | 2 | 11 | 16 | 22 | 275 | 220 | 165 | 7.3 | 5.8 | 4.4 |
| *Public Areas* | 100 | 80 | 50 | 9.3 | 7.4 | 4.6 | 1 | 1.5 | 2 | 11 | 16 | 22 | 175 | 140 | 110 | 4.6 | 3.7 | 2.9 |
| Hotels, Motels, Dormitories | 200 | 150 | 100 | 18.6 | 13.9 | 9.3 | 1 | 2 | 3 | 11 | 22 | 32 | 350 | 300 | 220 | 9.3 | 7.9 | 5.8 |
| Libraries and Museums | 80 | 60 | 40 | 7.4 | 5.6 | 3.7 | 1 | 1.5 | 3 | 11 | 16 | 32 | 340 | 280 | 200 | 9.0 | 7.4 | 5.3 |
| Office Buildings[4] | 130 | 110 | 80 | 12.1 | 10.2 | 7.4 | 4 | 6[2] | 9[2] | 43 | 65[2] | 97[2] | 360 | 280 | 190 | 9.5 | 7.4 | 5.0 |
| Private Offices | 150 | 125 | 100 | 13.9 | 11.6 | 9.3 | 2 | 5.8 | 8 | 22 | 62 | 86 | | | | | | |
| Stenographic Department | 100 | 85 | 70 | 9.3 | 7.9 | 6.5 | 5[2] | 7.5[2] | 10[2] | 54[2] | 81[2] | 108[2] | | | | | | |
| Residential: *Large* | 600 | 400 | 200 | 55.8 | 37.2 | 18.6 | 1 | 2 | 4 | 11 | 22 | 43 | 600 | 500 | 380 | 15.9 | 13.2 | 10.0 |
| *Medium* | 600 | 360 | 200 | 55.8 | 33.5 | 18.6 | 0.7 | 1.5 | 3 | 8 | 16 | 32 | 700 | 550 | 400 | 18.5 | 14.5 | 10.6 |
| Restaurants: *Large* | 17 | 15 | 13 | 1.6 | 1.4 | 1.2 | 1.5 | 1.7 | 2 | 16 | 18 | 22 | 135 | 100 | 80 | 3.6 | 2.6 | 2.1 |
| *Medium* | | | | | | | | | | | | | 150 | 120 | 100 | 4.0 | 3.2 | 2.6 |
| Shopping Centers, Department Stores and Specialty Shops | | | | | | | | | | | | | | | | | | |
| Beauty and Barber Shops | 45 | 40 | 25 | 4.2 | 3.7 | 2.3 | 3[2] | 5[2] | 9[2] | 32[2] | 54[2] | 97[2] | 240 | 160 | 105 | 6.3 | 4.2 | 2.8 |
| Department stores: Basement | 30 | 25 | 20 | 2.8 | 2.3 | 1.9 | 2 | 3 | 4 | 22 | 32 | 43 | 340 | 285 | 225 | 9.0 | 7.5 | 5.9 |
| *Department stores: Main Floors* | 45 | 25 | 16 | 4.2 | 2.3 | 1.5 | 3.5 | 6[2] | 9[2] | 38 | 65[2] | 97[2] | 350 | 245 | 150 | 9.3 | 6.5 | 4.0 |
| *Department stores: Upper Floors* | 75 | 55 | 40 | 7.0 | 5.1 | 3.7 | 2 | 2.5 | 3.5[2] | 22 | 27 | 38[2] | 400 | 340 | 280 | 10.6 | 9.0 | 7.4 |
| Dress Shops | 50 | 40 | 30 | 4.6 | 3.7 | 2.8 | 1 | 2 | 4 | 11 | 22 | 43 | 345 | 280 | 185 | 9.1 | 7.4 | 4.9 |
| Drug Stores | 35 | 23 | 17 | 3.3 | 2.1 | 1.6 | 1 | 2 | 3 | 11 | 22 | 32 | 180 | 135 | 110 | 4.8 | 3.6 | 2.9 |
| 5¢ and 10¢ Stores | 35 | 25 | 15 | 3.3 | 2.3 | 1.4 | 1.5 | 3 | 5 | 16 | 32 | 54 | 345 | 220 | 120 | 9.1 | 5.8 | 3.2 |
| Hat Shops | 50 | 43 | 30 | 4.6 | 4.0 | 2.8 | 1 | 2 | 3 | 11 | 22 | 32 | 315 | 270 | 185 | 8.3 | 7.1 | 4.9 |
| Shoe Stores | 50 | 30 | 20 | 4.6 | 2.8 | 1.9 | 1 | 2 | 3 | 11 | 22 | 32 | 300 | 220 | 150 | 7.9 | 5.8 | 4.0 |
| Malls | 100 | 75 | 50 | 9.3 | 7.0 | 4.6 | 1 | 1.5 | 2 | 11 | 16 | 22 | 365 | 230 | 160 | 9.6 | 6.1 | 4.2 |
| Refrigeration for Central Heating and Cooling Plant | | | | | | | | | | | | | | | | | | |
| Urban Districts | | | | | | | | | | | | | 475 | 380 | 285 | 12.6 | 10.0 | 7.5 |
| College Campuses | | | | | | | | | | | | | 400 | 320 | 240 | 10.6 | 8.5 | 6.3 |
| Commercial Centers | | | | | | | | | | | | | 330 | 265 | 200 | 8.7 | 7.0 | 5.3 |
| Residential Centers | | | | | | | | | | | | | 625 | 500 | 375 | 16.5 | 13.2 | 9.9 |

***Note:*** *Refrigeration for applications listed in this table of estimated cooling loads are based on all-air system and normal outdoor air quantities for ventilation, except as noted.*

[1] *Refrigeration loads are for entire application.*

[2] *Includes other loads expressed in W/unit floor area.*

[3] *Air quantities for heavy manufacturing areas are based on supplementary means to remove excess heat.*

[4] *Air quantities for hospital patient rooms and office buildings (except internal areas) are based on induction (air-water) system.*

Standards from organizations such as ASHRAE, the American Society of Mechanical Engineers (ASME), NFPA, and the Air-Conditioning, Heating, and Refrigeration Institute (AHRI) should be consulted for the following purposes:

- To ensure systems and equipment employed in the district cooling plant (DCP) meet minimum construction and performance requirements specified in design documents
- To ensure that performance testing is carried out using accepted test procedures and instrumentation
- To ensure safety of refrigeration systems, including various water chillers and refrigerant storage systems
- To provide adequate ventilation of the building that houses the refrigerants in order to protect operators from exposure to excessive amounts of the chemical
- To select chillers, boilers, hot water generators, and various pressure vessels that meet safety requirements

Regulations that must be followed may be both local and national. Federal regulations from the Department of Energy (DOE) and the Environmental Protection Agency (EPA) may impact the following:

- Emissions—especially applicable to system burning fuels where new sources of emissions may require permitting
- Fuel handling
- Energy efficiency
- Carbon footprint

State and local regulations that may impact the proposed project include the following:

- Building codes and standards covering materials and safety requirements
- Emission limits for heating plants
- Construction and operating permits
- Boiler and pressure piping codes
- Right-of-way for installing piping and electrical systems, where required

The significance of reviewing and becoming very familiar with various codes, standards, and regulations should not be underestimated. Codes, standards, and regulations impact the design and cost of DCSs. They are especially important when meeting special building and operating requirements, such as restrictive emission limits and chemical storage and handling. In addition, codes may significantly impact the aesthetic and mechanical design of a utility plant and its distribution system.

## Special Considerations for DCS

When reviewing codes, standards, and regulations for compliance it is important to consider the unique features of a DCS that may not be familiar to the owner/designer from their experience with other types of buildings/projects. These include the special considerations that come from operation of large cooling towers, such as *Legionella* and other impacts from the drift that is high in dissolved solids, in addition to the concentrated constituents in the tower blowdown. Where treated sewage effluent (TES) is used as makeup water in cooling towers, additional care will be needed. See Chapter 3 for additional information on cooling towers and TSE. As with any plant, noise from equipment may be an issue. Large volumes of refrigerant(s) and water treatment chemicals are other issues encountered with DCSs that may not be found in most building projects. Furthermore, DCSs will often have large quantities of water in storage as a backup to normal water supply for tower makeup. When TES is used, there will be potentially even larger volumes of water in storage that could accidentally be discharged. See Chapter 6 for more information on TES design.

## Local and Institutional Constraints

Institutional/community practices and priorities may also be considered as an additional level of code that must be complied with. For example, local practices may require the central plant be constructed with visual screening, so as to disguise cooling towers. Another important institutional or local factor that must be considered is disruption from the construction/expansion of a buried CHW distribution system. While routing of piping across a common, park, or green might be the most expeditious route, doing so might be unacceptable. Trees are usually sacred, and as well they should be when they are special or rare species, not to mention their role as good oxygen producers. Good practice is to search out the client's forester or arborist, if there is one, before routing large underground piping near trees. Trees and other campus-type monuments may sometimes have greater impact upon pipe routing than congestion from existing active and abandoned utilities. If a forester or arborist is not available, avoiding the tree's drip line is good practice. A good composite utility map is the starting point for pipe routing, but ultimately a three-dimensional representation of the underground utilities is key and it should include a representation of surface factors such as buildings, trees, paved surfaces, etc.

## Integrated Processes

There are many opportunities to integrate production of CHW with either heat production, electric power product, or both. Doing so allows for shared resources as well as potential benefits for more thermodynamically efficient systems. The potential ways in which these processes may be combined are numerous, thus we will not endeavor to treat them all here, nor will those discussed be treated in detail.

Where heating requirements are also being met with a district system, it may be advantageous to combine the processes and consider generation of electricity and cogeneration of heat and power as covered by Chapter 7 of ASHRAE (2016). Absorption-based cooling may be combined with vapor-compression chillers to accomplish the CHW production. If the generation of heat and electricity will be combined with the generation of chilled water, the array of possibilities that must be considered is greatly expanded. In addition, the selection of fuels may be much broader, which will have impacts on plant siting as discussed in the "Central Plant Siting" subsection of this chapter. In the planning stage, consideration should be given to thermal storage as both a means of reducing electric costs but also for the possible chiller capacity reduction and for the backup capacity it provides. Thermal storage is covered in Chapter 6.

The combination of district cooling with inlet-air cooling for combustion turbines used for electric power generation is an obvious combination. The performance of combustion turbines is significantly reduced by increasing inlet-air temperature. For example, an inlet-air temperature increase from the standard rating condition of 59°F to 100°F (15°C to 38°C) will decrease output by 19% and efficiency by 4% (ASHRAE 2016). Since power generation is usually strained by peak air-conditioning periods, this offers a method to mitigate part of the impact, and it is not even necessary for the combustion turbine to be located at the same location as the district cooling plant if the CHW distribution system has adequate capacity. Combining combustion-turbine inlet cooling with thermal storage is also an obvious extension. Where the combustion turbine and thermal storage are not colocated with the district cooling central plant, thermal storage could help relieve peak-load impacts on not only the DCP but also the distribution network.

*ASHRAE Handbook—HVAC Systems and Equipment* has a chapter dedicated to combustion turbine inlet cooling (ASHRAE 2016).

Recovery of waste heat from the condensing process of the CHW central plant is also another possible application. Details on such heat recovery systems are provided in Chapter 9 of *ASHRAE Handbook—HVAC Systems and Equipment* (ASHRAE 2016). Since the normal condensing temperatures in a CHW system are too low to be of direct use for most applications of heat, it is usually necessary to boost temperatures. This can be done by an additional stage of equipment or a double-bundle chiller if the required temperature is low enough (i.e., under approximately 155°F [68°C]). When higher temperatures are required, it will probably be necessary to use a device such as an industrial heat pump. The heat pump would use the condenser water of the CHW plant as a heat source and could provide heating temperatures as high as 220°F (104°C); although, one needs to appreciate that the higher the temperatures desired, the lower the heat pump performance will be. For additional information on heat recovery chillers and industrial heat pumps, refer to Chapter 9 of *ASHRAE Handbook—HVAC Systems and Equipment* (ASHRAE 2016).

An interesting district cooling application that cogenerates power as well as heat is described by Mornhed et al. (1992). The system uses a combustion turbine to drive a 2000-ton (7000 kW) ammonia chiller for district cooling. A direct-expansion ammonia coil is used to cool the natural gas fueled-turbine inlet air. The dual shaft turbine also has a motor/generator so that when cooling demands are lower, excess power from the turbine may be used to generate electricity, or when cooling demands are high the turbine output can be supplemented with electric input to drive the chillers. Heat is recovered from the turbine exhaust for the district heating system in the form of 150 psig (1030 kPa) steam.

Meckler (1997) compares the cost-effectiveness of a conventional cogenerating plant for a large hospital and office building complex with an alternative that uses gas turbine-driven centrifugal chillers as well as a separate gas turbine-driven electric generator. Heat is recovered from both gas turbines and turbine inlet-air cooling is provided by CHW. The system also has thermal storage for heating and cooling effects. For the cooling effect the storage is via encapsulated balls that contain water with a freezing-point enhancing nucleon agent.

For more information on systems that involve the production of electricity, the reader is referred to ASHRAE (2016), which has a chapter dedicated to combined heat and power systems, Meckler and Hyman (2010), Sweetser et al. (2015), or to Orlando (1996).

## Not-in-Kind and Novel Approaches

The discussion above has focused mostly on alternatives that have been adopted in multiple systems, but there are often technologies that can be applied to the special circumstances of a particular application. Examples include systems that use seawater or deep lake water for cooling, either directly or indirectly. Such systems will require not only special engineering but may also require preliminary studies to establish some of the design parameters. We have included several case studies of such systems in Appendix B of this guide. The reader is also referred to UNEP (2015), which has a number of very brief case studies of both district cooling and heating projects.

## Phased Development and Construction

In some instances, phased development will be the logical alternative for the construction of a DCS. Such cases normally occur where the full load is projected over a period of time, service provision is committed, and risk mitigation is crucial.

Phased development and construction can be managed by several options to suit the site conditions:

- Temporary plant(s) and a temporary/permanent distribution network to serve firm cooling capacity requirements until reaching a feasible threshold for building the permanent arrangement
- One permanent network with multiple permanent plants (if feasible), constructed in phases in accordance with load escalation
- One network and one permanent plant properly designed and engineered for initial partial operating load planned to reach the ultimate capacity over a period of time

## Central Plant Siting

Because site constraints are so highly varied, it is difficult to provide general guidance on plant siting. For larger systems, it may be prudent to consider multiple chiller plants, which may include phased construction of the plants. In addition to the code and standards constraints discussed above, the following factors should be considered in choosing a plant site.

### Aesthetics

While ideally the plant would be located at the centroid of the loads to be served, siting will be heavily influenced by the aesthetics and arrangement of the buildings. In evaluating alternative sites, it may be necessary to conduct preliminary hydraulic analysis (discussed later) in order to evaluate the distribution system first-cost and operating-cost impacts of competing plant locations.

### Acoustics

The plant should be sited away from sound-sensitive adjacencies, such as residential areas, music halls, libraries, etc., and all measures should be taken to keep all load equipment noises from transmitting through the central plant building openings. Further scrutiny and calculations relating to acoustical and vibration abatement should be undertaken.

### Topography

Topographical factors may play a large role in plant siting: a plant located at the low point in the system will be subjected to system hydrostatic heads, which may be significant where elevation differences are great. Where a thermal energy storage (TES) system is used, it should be located at a higher elevation than the building equipment served, regardless of if the in-building equipment consists of the heat exchangers of an indirectly connected system or the coils of a directly connected system. An addition planning consideration when TES is used is the additional land area required for the TES system. See Chapter 6 for details on TES system design.

### Fuel Availability, Storage, and Handling

For a chiller plant using electric driven vapor-compression chillers, adequate electric infrastructure will be a major consideration in plant siting. For systems that use engine-driven chillers as well as plants that cogenerate heat and/or electric power, the fuel used may have significant impacts on site selection. Obviously, a natural gas fueled plant will require either location near an adequate existing supply pipeline or construction of a supply pipeline from a nearby main. For systems that use liquid and solid fuels, adequate space must be available on site for fuel unloading and storage. Fuels such as biomass that have relatively low heat content per unit volume will require frequent deliveries or increased storage capacity on site.

Solid fuels such as biomass or coal may be delivered on trucks, or by rail where available, and specialized facilities may be required at the plant site to unload the trucks or rail cars.

For estimating electrical service sizing, a round number to use is 1 kW/ton of refrigeration, or 0.28 $kW_e/kW_{th}$. Variable primary flow plants, plants with higher efficiency chillers, and systems with smaller distribution systems will be less than this value (Refer to Table 3.4).

### Cooling Tower Location

Site selection should consider the many benefits of locating the cooling towers on the ground. The extra effort required and the additional land used on a life-cycle cost basis are typically a good investment. The cost-benefit advantages are ease of tower maintenance, elimination of roof repairs and the cost of roof leakage, ease of increasing capacity, the aesthetic advantage that can be realized, and the control of tower noise.

### Central Plant and Cooling Tower Space Allocation

The size of the plot of land required for the central plant is a major factor in system planning, and several technical aspects influence the required plot size. Because cooling towers are often placed on the central plant roof, local building regulations must be researched if there are limits placed on building height. (Of course, aesthetics could also limit building height.) If cooling towers cannot be placed on the roof of the central plant, the required plot size will be roughly doubled. It is often prudent to oversize the cooling towers to mitigate some unforeseen conditions such as higher than expected wet-bulb temperatures, short circuiting of cooling tower air, fouling of cooling tower fills by sand storms or other blowing debris, etc. Practical limitations on construction practices may also dictate maximum central plant height. Based on experience, approximately 100 ft (32 meters) is a typical maximum building height. See Chapter 3 for design guidance for central plants and their cooling towers and Chapter 8 for a discussion of water treatment.

The source of cooling tower makeup water can have a significant impact on central plant size where nonpotable water alternatives such as seawater or TSE (treated sewage effluent) are used. In addition to any on-site treatment plant requirements, using alternative water sources will normally require larger water storage tanks, increased chemical storage space, etc. Chapter 3 contains guidance on TSE treatment plant design and Chapter 8 has information on water treatment.

If a thermal storage system (TES) is considered, especially the stratified water type, large tanks that have very large space requirements are required. Chapter 6 provides guidance on sizing and design of TES.

If low-voltage chillers are used, the space required for step-down transformers should be considered in initial plant layout and space requirement estimates. See Chapter 3 for chiller selection guidance for central plants.

Central plant plot size will be largely proportional to plant capacity, although large plants will benefit from economies of scale and have somewhat lower space requirements for a given plant thermal capacity.

For approximate sizing of cooling tower spatial requirements, assuming field-erected towers, a good rule of thumb is 0.40 $ft^2$/ton (0.13 $m^2$/kW) for towers located on grade for a minimum size 10,000-ton plant (35.2 MW). Similarly, if the cooling tower is located on the roof of the chiller plant the values increase to 0.5 $ft^2$/ton (0.16 $m^2$/kW) due to proximity of solid or open screen walls for a minimum sized 10,000-ton plant (35.2 MW). However, this value is dependent upon the wet bulb selected and condenser water flow, approach, and range (see Chapter 3).

Because the roof will have many other components (exhaust fans, access hatches, stairs, elevator overruns, exhaust vents, roof drains, etc.), it should be acceptable to use a value closer to 0.9 $ft^2$/ton (0.23 $m^2$/kW) as the roof area and minimum foot print of the plant building for layout purposes. The values also decrease as the plant gets larger and values will increase if packaged units are used. The equipment can then be laid out to determine how many floors the plant should be.

Significant experience has been gained from many district cooling plants located in high-temperature climates such as Dubai, UAE, where capacity density values for concept planning vary from 0.46 to 0.84 ton/$ft^2$ (17 to 32 kW/$m^2$). The variation results from the many issues discussed above and other issues. This approximate range of plant plot sizes will be further developed by the designer in reaching an appropriate design for the site conditions. See Chapter 3 for more information on central plant design.

## Chiller Selection

Chiller selection must include consideration of a number of variables. Among them are anticipated-peak diversified loads; winter-minimum loads, if any; expected-average load or normal operating loads; range of capacity of available water chillers; reliability issues as an increasing number of chillers are put in place; and of course, refrigerant preferences (discussed below) and energy efficiency both at maximum capacity and at in the sweet spot in the 50% to 80% range of capacity. Selection of a modular capacity that meets the variables listed and permits optimization of energy usage is preferred to simplify maintenance and stocking of parts to ensure continuity of service to mission-critical users. A proper selection may permit the plant to provide the much sought after variable primary flow without the complication or expense of special pump drives, see Chapter 3 for detailed design of the chiller plant.

Due to their high installation costs, it is desirable to have chillers selected on a life-cycle basis in lieu of the basis of lowest first cost or best efficiency. Only when the annual maintenance costs, the cost of prime driver utilities, and the replacement and repair costs are included over the life of the equipment along with the first cost, can the most optimum selection be determined.

## Refrigerant Selection

Beginning with the Montreal Protocol in 1987, the use of chlorofluorocarbons (CFC) and hydrochlorofluorocarbons (HCFC) refrigerants found to be harmful to the environment has becoming increasing regulated. This process began in developed countries and is being followed on a delayed schedule in developing countries. Refrigeration working fluids have been in a constant state of flux since the adoption of the Montreal Protocol due to the preponderance of equipment already in use as well as the lack of readily available alternatives. Phase out of the most harmful compounds responsible for destruction of the ozone layer has largely been with replacement, using compounds less harmful to the ozone layer, yet most were found to contribute to global warming. For the users of cooling equipment, this has meant additional requirements in terms of handling of existing materials, and the future will likely see ever increasing levels of regulation.

Due to the ever-changing regulatory environment it is not possible within the confines of this guide to address all the issues surrounding refrigerant choice or offer specific guidance to either those with existing DCSs or those planning new systems. The reader is encouraged to seek sources of information that are reliable and current. There are many issues surrounding development of alternative refrigerants, and much of the effort is now being placed on small unitary equipment as opposed to the chillers that would be used in a DCS. A relatively recent comprehensive summary of these issues, with specific focus on DCS, is provided by Olama (2017). After Olama (2017), there has been one major development in the refrigerants regulatory area and that is an agreement to amend the

Montreal Protocol that was reached at the 28th meeting of the parties to the Montreal Protocol held in Kigali, Rwanda during October 2016. The Kigali Amendment has been ratified by the minimum of 20 parties, so it was adopted and takes effect on 1 January 2019. From the Montreal Protocol, the phase out of the most harmful refrigerants has had separate time lines for “developed” and “undeveloped/underdeveloped” (or Article 5) countries. Under the Kigali Amendment, the Article 5 countries have been further divided into two groups with different phase-out timelines. Locations with high ambient air temperatures (HAT) are especially challenging for air-conditioning and refrigeration systems of any type as a simple consequence of thermodynamics, as a minimum lower performance must be expected with all else being equal. The Kigali Amendment provides for additional relief in the phase-out schedule for HAT countries; however, this only impacts several types of small packaged equipment and does not impact chillers that would be used for DCSs.

## Chilled-Water Distribution Systems

Once the loads at the buildings have been established, the distribution system can be laid out (i.e., a plan view developed). Normally the distribution system is modeled by a simple hydraulic analysis using the flow rates required to satisfy the loads at the design loads, given assumed supply and return CHW temperatures. Numerous commercially-available computer-aided design programs are available for conducting hydraulic analyses. CHW supply and return temperatures impact many aspects of the DCS design and they are discussed throughout this design guide. In the planning phase for the calculation of flows, the engineer may simply need to have an assumed value for the temperature difference between supply and return, often called the system $\Delta T$, or simply $\Delta T$. Chapter 5 discusses typical design $\Delta T$ values and suggests that values in the range of 12°F to 16°F (6.7°C to 8.9°C) at full load may be achieved. With designs that place well deserved emphasis on achieving high $\Delta T$, values as high as 20°F (11°C) or more are possible. However, for systems that have many older buildings connected that cannot be properly retrofitted for connection to a DCS, lower $\Delta T$ values should be assumed, or if experience is available that $\Delta T$ should be used as the basis. Values at the higher end of the $\Delta T$ range will be associated with systems that have mostly newer buildings designed for district cooling and may also have ice-based thermal storage systems, which will often result in lower CHW supply temperature and thus higher $\Delta T$. The importance of maximizing $\Delta T$ cannot be overemphasized; see Chapter 5 for further discussion.

With the assumed $\Delta T$ and the loads, the design flows are calculated and the hydraulic analysis of the distribution network may proceed. Pipe sizes must be chosen and that will be the critical aspect of the planning process. Pipe sizing will have a profound effect on both the operation of the system planned, but also on the ability to expand the system in the future. The choice of pipe size is the classic optimization problem, as described in Phetteplace (1994) for hot-water district heating systems, which are largely analogous to DCSs in this aspect of their design. The basic trade-off is that increasing pipe size results in lower pumping energy, but higher capital costs as well as higher costs associated with heat gain in the case of CHW pipelines. A second order effect is that higher pumping energy consumption results in additional heat load on the system that must be removed by the chillers. For district heating hot water distribution systems, guidelines for pressure losses are given by ASHRAE (2016) as 0.44 psi per 100 ft of pipe (100 pa/m) based on European experience (Bøhm 1988). Other studies have suggested that higher levels of pressure loss may be acceptable (Stewart and Dona 1987) and warranted from an economic standpoint (Bøhm 1986; Koskelainen 1980; Phetteplace 1989). However, where there is high uncertainty in the planning process regarding future loads, over-sizing of pipes is generally preferable to the converse. The prospect of having to excavate and replace an undersized segment of existing piping can be very costly after surfaces, vegetation, and buildings are established and occupied. This replacement is further complicated by the fact that there are buildings connected that must have continuous service and cannot be without cooling for an extended period of time. Very often the cost of increasing pipe size in the first instance to provide low pumping heads (as low as 65 ft [20 m] total dynamic head from the plant to the most remote user) is a good investment. Low primary-system head eliminates many of the control valve problems that sometimes result from overpressure during light load periods. If the system end users depend solely on the secondary pumps to provide the necessary head for required flow rates, reducing pressures with a variable frequency drive (VFD) may create user dissatisfaction.

Hydraulic analyses should also look for the potential to establish loops within the system. Loops within distribution systems are to be favored when the routes are available for them as the loops provide alternate flow paths, which may be used during disruption to flow via the primary path, and when the primary path is in use; secondary paths via loops will reduce flow and thus pressure losses via the primary path. Hydraulic simulation of the network is essential if loops are to be used and their functioning under various load conditions understood. Hydraulic analyses are also useful tools for selecting control valves at end users. In essence, valves that are capable of dissipating the excess available head must be selected at the building interface with the distribution system; see Chapter 5.

In existing systems, the hydraulic analysis should look closely at bottlenecks within the system. The critical flow paths must be identified by the analyses; otherwise efforts to redimension the network for future requirements or to correct existing deficiencies will fail. The addition of more loops will often be the solution to bottleneck problems and to reducing overall system pressure drop.

One other note on underground CHW distribution piping is the benefit of insulating the supply and return piping. The notion that soil/groundwater temperatures approximate those of the chilled water should be cast aside. Soil temperatures are highly variable, as discussed in Chapter 4, not just owing to climate differences, but also the type of surface and burial depth. Despite the fact that insulated piping is significantly more expensive, insulation will often pay for itself during its life cycle. Obviously, savings come from lower heat gain to the piping from the ground, but more importantly there can be significant impacts on system pumping costs as well. This may occur especially during light loads when water in the piping of remote reaches of the system would otherwise be flowing in at low velocities. Due to heat gain in the distribution piping, under these circumstances supply water temperatures may be increased to a level where consumer demands for dehumidification cannot be met. Under those circumstances, it is often necessary to pump much more water than is required bypassing a significant percentage just to achieve adequate flow and supply temperature. This not only increases pumping but it erodes system $\Delta T$ (see Chapters 3 and 5 for a discussion of the impacts of system $\Delta T$). Chapter 4 contains an example calculation on the temperature impacts of CHW piping insulation.

## Unconventional Working Fluids

The desire to reduce the impact that pumping has on district cooling energy consumption has driven the search for fluids that will provide greater heat absorbing potential for a given rate of circulation. Additionally, additives that reduce the frictional losses of the water flowing through the district cooling piping have also been investigated. In addition to reducing flow rates, the other application of such advanced fluids would be to increase the capacity of an existing network. Note that enhancements to district cooling fluids for the purposes of thermal storage are discussed in Chapter 6.

Many studies have been conducted with friction-reducing additives, primarily for district heating systems. The unfortunate side effect of the surfactants used for these purposes is that they also tend to decrease heat transfer rates in heat exchangers. One effort (Weinspach 1996) that included studies on the impacts on heat exchangers looked at possible enhancements to heat exchangers to offset the heat transfer reductions. Using spiral stainless steel springs fixed to the inside of tube-type heat exchangers enhanced the heat transfer yielding results similar to pure water; however, pressure losses increased from 200% to 800%.

The other approach to increasing the heat absorbing potential has been to introduce phase change materials other than water into the circulation, thus yielding a water slurry. Most of the investigation has centered on using ice (Electrowatt-Ekono Oy and FVB District Energy 2002), although some studies have looked at other phase-change materials (Alvarado et al. 2008). The potential for ice slurries is great. For example, according to Electrowatt-Ekono Oy and FVB District Energy (2002), adding an ice fraction of 20% and decreasing the supply temperature to 32°F (0°C) would increase the cooling capacity of a system by nearly four times, compared to the more traditional supply temperature of 44.6°F (7°C), assuming in both cases that the return temperature was 57.2°F (14°C). Of course, simply lowering the temperature to 32°F (0°C) is responsible for doubling the capacity, but the 20% slurry doubles that result yet again. One of the major concerns with pumping ice slurries through district heating networks is that the ice will agglomerate. Electrowatt-Ekono Oy and FVB District Energy (2002) state that studies to date have shown that as long as ice fractions do not exceed 25% and flow velocities are maintained above 1.6 ft/s (0.5 m/s), and preferably above 3.3 ft/s (1 m/s), that agglomeration is not expected. Another issue is stagnation of the ice slurry under which conditions the ice particles will float to the top; however, with ice fractions lower than 20%, it is expected that adequate flow area will remain in the piping system at the bottom of the pipe until the flow becomes turbulent and the ice remixes (Electrowatt-Ekono Oy and FVB District Energy 2002). Flow splitting while maintaining uniform ice concentration and pressure losses are other concerns with ice slurries, and neither of these appear to be major barriers (Electrowatt-Ekono Oy and FVB District Energy, 2002). For additional information on ice slurries the reader is referred to Electrowatt-Ekono Oy and FVB District Energy (2002) and Hansen (2002).

## Construction Considerations and Cost

A key element in the decision-making process in any master planning effort is the cost estimation of equipment, materials, and labor required to implement the proposed alternatives. Thinking like a mechanical contractor during the cost-estimating process is essential if the results are to be appropriation grade estimates. The master plan or, its ultimate implementation, will fail if proper attention is not given to the estimating process. Although systems have not been designed in the master planning process, enough data is generated to take estimates to a fair amount of detail. Experience indicates that estimates made today can be used with cost-escalation data out as far as ten years or more, when they are based on knowledge of the systems proposed and not some generalized unit cost data that is readily available.

The construction costs of the central plant and distribution system depend on the quality of the concept planning and design. Although the construction cost usually accounts for most of the initial capital investment, neglect of future operational and/or maintenance costs could mean the difference between economic success and failure. Field changes usually increase the final cost and delay start-up. Even a small delay in start-up can adversely affect both economics and consumer confidence. It is extremely important that the successful contractors have experience commensurate with the project. It is not unusual for the specifications to require a contractor to have a minimum of five years of experience in similar type projects to ensure that costly novice mistakes are not made. Capital costs of district cooling projects range greatly and are dependent upon local construction environment and site conditions such as the following:

- Labor rates
- Construction environment (slow or busy period)
- Distance for shipping of equipment
- Permits and fees (franchise fees)
- Local authorities (traffic control, times of construction in city streets)
- Soil conditions (clay, bedrock)
- Quality of equipment and controls (commercial or industrial)
- Availability of materials
- Size of piping in distribution system
- Type of insulation or cathodic protection of piping system
- Type of distribution-system installation (direct buried, tunnel, etc.)
- Depth of bury and restoration of existing conditions (city streets, green areas)
- Below-grade conflict resolutions
- Economies of scale

Sample construction cost-unit pricing is summarized by ASHRAE (2016) and is provided in Table 2.2; however, the designer is cautioned that cost can vary widely based on the conditions outlined above. For example, very large chiller plants that have been constructed in the Middle East can cost as little as $1250 per ton ($350/kW) while more typical values for North American chiller plants are in the $3000 to $3500 range for plants of around 10,000 tons (35 MW) capacity. In the case of distribution systems, large diameter piping can increase costs significantly as can urban construction conditions.

Lead time needed to obtain equipment generally determines the time required to build a DCS. In some cases, lead time on major components in the central plant can be over a year. Installation time of the distribution system depends, in part, on the routing interferences with existing utilities. A distribution system in a new industrial park is simpler and requires less time to install than a system being installed in an established business district.

## Consumer Interconnection

The interconnection of the buildings with the distribution system will be a major cost component of the system that must not be ignored in the planning phase. Consumer interconnection costs will vary widely dependent on the type of existing system in the existing buildings (if any) and the type of building interconnection; direct or indirect. Table 5.1 pro-

**Table 2.2** Sample Cost Information (ASHRAE 2016)

| Item | Cost Range per Unit[1] | Unit of Measure |
|---|---|---|
| Chiller plants (including building, chillers, cooling towers, pumps, piping, and controls) | $1,800–$3,500<br>($500–$1,000) | ton (kW) of capacity |
| Direct buried distribution piping (includes excavation, piping, backfill, surface restoration) | $500–$1,250<br>($1,500–$5,000) | ft (m) of trench length |
| Distribution system, for buried inaccessible tunnels | $700–$1,500<br>($2,300–$5,000) | ft (m) of trench length<br>ft (m) of trench length |
| Distribution system, buried walk through tunnels | $3,500–$15,000<br>($11,500–$50,000) | |

[1] *Costs include design fees, contingencies, and taxes.*

vides a summary of the relative merits of direct versus indirect connections. Consumer interconnection costs are usually borne by the consumer for a system that is a commercial venture. However, the magnitude of interconnection costs must be considered in the economic analysis discussed below. High interconnection costs may favor an in-building plant for the customer instead of a DCS.

The consumer interconnection, the consumer's in-building equipment, and their control will ultimately determine the $\Delta T$ that the system will be able to achieve. Chapter 5 discusses the consumer interconnection and its impacts on the all-important system $\Delta T$. This fact cannot be highlighted enough as to its importance. In the planning phase, it is important to recognize the importance that must be placed on proper consumer interconnection and in-building systems, and thus be certain that adequate study of the existing building systems has been conducted and that future buildings are being planned for connection to a DCS. To achieve a high $\Delta T$, as well as to provide proper comfort and control within the building, the building design should follow the recommendations of Chapter 5. For retrofit of existing buildings, Chapter 5 should also be consulted in the planning phase to be certain that adequate resources are allocated to this portion of the system.

Retrofit costs for buildings will vary widely. Factors that will tend to increase cost are:

- Tall buildings
- The need to run piping to the roof to connect to header piping from prior chillers
- Replacing 3-way valves with 2-way valves to improve $\Delta T$
- Implementing variable flow pumping to improve $\Delta T$
- Adding redundancy or oversizing equipment (pumps, heat exchangers, etc.)

Metering is a necessary component in most all consumer interconnections and appropriate allowance should be made for such costs in budget construction cost estimates. While it is tempting to not include metering when there is a common owner for all buildings (i.e., a college, military, or institution campus), experience has shown that troubleshooting both network and building systems operations and any problems therein is greatly enhanced by metering. Metering should include the ability to obtain flow and temperature data as well as energy usage, and many systems will allow for remote monitoring of this data at a central point such as the district cooling central plant. See Chapter 5 for a detailed discussion on metering and remote systems monitoring.

### Energy Cost

Energy will be the major operating cost of a DCS, and, thus, where multiple alternatives are available, each should be carefully considered. For example, the higher capital and maintenance cost of engine-driven chillers could possibly be more than offset by lower energy costs for the engine fuel as opposed to electricity.

Since the prime driver of the chiller (electric motor, steam turbine, gas engine, etc.) is usually the largest motor, it is typically the largest consumer of energy in the CHW system. Pumps for the CHW distribution system water as well as condenser water pumps will also require significant amounts of energy, as will the cooling towers fan motors. Because distribution system pumping is normally accomplished at the chilled water plant, it is conventional practice to quote overall chiller plant performance including all pumping, chillers, and cooling towers in terms of electric energy consumed at the plant per unit of cooling produced, normally in kW/ton ($kW_e/kW_t$). The overall efficiency of the chiller plant expressed in this manner will vary significantly depending on many factors, including the climate, chillers, system $\Delta T$, load density of the users, etc. The expected range of overall chiller plant performance at peak-load conditions will normally fall within the range of 0.80 to 1.2 kW/ton (0.23 to 0.34 $kW_e/kW_t$).

As part of the central-chiller plant operating energy costs, it is important to also include the heat gain within the CHW distribution system; calculating the heat gains and their associated costs is covered in Chapter 4. While distribution system heat gains may add as little as a few percent to load when the system is at capacity, the impact on annual energy consumption will be greater as the heat gains will persist with little or no reduction at times when CHW demand by the consumers is significantly lower.

### Operations and Maintenance Costs

Aside from the major cost of energy, there will be other significant O&M costs that should be considered. At the planning stage, a detailed analysis of these costs is normally not warranted. A method of making a first order accounting for these costs is to assume they are a percentage of the capital cost of the system on an annual basis. For heat distribution systems Phetteplace (1994) has suggested 2% of the initial capital cost as the annual costs, excluding the cost of heat loss and pumping energy. On a similar basis, O&M costs for a chilled water distribution system are probably lower, perhaps 1% to 1 1/2% of initial capital cost. For central plant O&M costs, a higher percentage of capital costs may be justified on an annual basis, perhaps 2% to 3% exclusive of energy costs; see Example 2.1 later in this chapter and refer to Chapter 3 for chiller maintenance costs.

Equipment lifetimes will be an important factor in an economic analysis. Chapter 37 of ASHRAE (2015) provides data on service life estimates for chillers, cooling towers, pumps, and other major equipment as would be used in a district cooling central plant, as well as the expected life of alternative systems to district cooling such as air-cooled chillers or through the wall air-conditioners. As ASHRAE (2015) points out, much of this data is quite old and recent efforts to update it have yet to provide a large enough database to provide high confidence estimates for many types of equipment. Currently ASHRAE maintains an online database where additional data can be added by users and the database can be queried to find the most recent statistics; this database may be found at www.ashrae.org/database (ASHRAE 2013). At the time that this guide was prepared, there were significant amounts of data for chiller life, for example, and even a limited sample of data on the lifetime of DCSs. In the case of the latter, it is assumed that this is from a building owner's perspective rather than a district cooling provider's perspective, which of course in the case of campus systems, for example, may be identical.

## TYPICAL RESPONSIBILITY OF DISTRICT COOLING PARTICIPANTS

There are typically three main parties or participants in a district cooling system—the provider, the customer, and the design engineer. All parties must work together in order for the system to be successful. Because contracts with customers vary from installation to installation, there are no fixed responsibility guidelines of which participant provides what aspect of the system, but the following topics are generally accepted industry norms.

### Responsibility of the DCS Provider

The district cooling provider is responsible for providing consistent and reliable chilled-water service to each customer. Typically, the items they are responsible for providing:

- A properly designed and energy efficient chilled-water generating plant (chillers, pumps, cooling towers, etc.)

- The piping from the DCP to the customer's entry point (point of delivery)
  - For an indirect connection using a plate heat exchanger (PHE), the mid-point of the PHE is considered the point of delivery
  - For a direct connection, the point of delivery could be the location of the decoupler or another convenient location prior to the customer's pumps
- The building isolation valves on district side of the interconnection
- The chemical treatment and makeup water for district side of system
- The heat exchanger and valving at the customer's interconnection
- The controls, including return water temperature control valve, energy meter, and temperature and pressure transmitters to control and monitor the flow through the interconnection. The DCP typically provides the temperature transmitters on the customer's side of the interconnection
- Commissioning of all DCP installed equipment and devices
- Proper maintenance and calibration of all DCP-owned equipment

## Responsibility of the DCS Customer

The customer is responsible for taking custody of the chilled water from the point of delivery and using it efficiently. Some contracts require a specific return water temperature leaving the building, therefore, the HVAC systems must be designed accordingly. The customer is generally responsible for providing:

- Suitable space for ETS equipment for servicing and operation including, but not limited to:
  - Potable water connections for system filling and flushing
  - Floor drains for drainage of condensate and system water to sanitary sewer
  - Power for DCP controls and customer equipment
  - Removal of all hazardous material from site
- Isolation valves on customer's side of PHE
- Access to DCP personnel to inspect and maintain their equipment
- A properly sized and designed energy efficient HVAC system to deliver the highest return water temperature possible back to the DCP
- High quality materials and equipment to ensure reliability of operation
- Customer building chilled-water circulation pumps, expansion compensation, and controls
- Chemical treatment and makeup water for customer's side of system
  - Proper cleaning and flushing of all piping prior to energizing district cooling interconnection
- Proper commissioning and startup of all customer installed equipment and devices
- Proper maintenance and calibration of all customer-owned equipment

## Responsibility of the DCS Design Engineer

Finally, the system will not function properly without the close communication of the DCP design engineer and the customer's building engineer of record to ensure:

- Proper design of final pipe routing to the point of delivery and equipment selection and that the supply and return pipes are coordinated and not interchanged
  - Confirm pressure rating of all devices and equipment based on building height
- Determination of peak cooling load to use for sizing all piping and equipment

- High return water temperature from cooling equipment by offering guidance to building engineer of record on cooling coil and valve selection
- Correct installation of instrumentation and controls
- Adequate space is provided for servicing and maintaining all equipment
- All equipment is appropriately valved for isolation
- All equipment has appropriate structural supports
- Proper sequences of operation of equipment for interconnection control

## ECONOMIC ANALYSIS AND USER RATES

This section covers economic topics in more detail and assists consultants and building owners in comparing the costs of self-generated (in-building) central cooling plants to that of a district cooling contract or offer since a proper analysis will include more inputs than just energy and construction costs. The evaluation is extremely site specific, and while not complex, it will have many input parameters, some that are economically quantifiable and others that are qualitative and a monetary value cannot be assigned, yet those parameters do add/subtract value.

For the parameters that are quantifiable, the most common method of evaluation is a net-present value or life-cycle cost (LCC) analysis that encompasses all the major costs and charges associated with each option and includes the time value of money over the life of the project or life of the contract. The following discussion is intended to assist the evaluator in understanding the big picture and identifying all the costs to prepare a fair comparison. Typically, a district cooling contract is for a minimum 20 year period and the contract includes all conceivable charges including charges that the evaluator may not be aware of. The duration of the contract enables the district cooling provider to recoup their costs for the expenses incurred in connecting the building or reserving the capacity at the district cooling plant for the building's load.

Ideally a complete economic comparison between the in-building (self-generated) and district cooling options would include 8760 h thermal energy load data from an existing building or the results from a computer energy model looking at the annual energy usage. An 8760 h analysis is desirous especially for cooling applications utilizing electrically driven equipment, since many times, the reduction in electrical demand charges by connecting to district cooling will result in tremendous savings to the building owner. These savings are best captured by utilizing the exact utility electric rate structure on an annual basis and the information shared with all parties. This is especially true when the building electric rate utilizes real time pricing (RTP). The information in Table 2.3 summarizes the key parameters that are input into a detailed economic comparison calculation.

Since there are benefits to the building owner that cannot be assigned a monetary value, the comparison between the two alternatives should be a value-based decision. In other words, the analysis should not only include the quantitative variables itemized above, but also the qualitative variables and benefits that have intrinsic value for the proposition. For example, some qualitative variables that add value to a building connecting to a district cooling system would be:

- Reuse of the space vacated by the cooling equipment since some of the mechanical and electrical space can be rented out for uses other than storage. Uses such as office space, or a clean roof area that could be used for more sustainable purposes such as a roof garden, pool, etc.
- No plumes from cooling towers or boiler stacks
- Increased thermal cooling source reliability
- More stability in energy costs

**Table 2.3** Summary of Economic Analysis Factors

| | |
|---|---|
| **Capital Costs** | |
| Construction costs of the building plant vs. energy transfer station equipment | Includes the materials and labor for chillers, boilers, piping, pumps, heat exchangers, valving, instrumentation, controls, cost of electric service, cost of additional structures due to equipment weight on roof of building, etc. |
| Value of increased mechanical and electrical space that would house the plant equipment | Includes value of penthouse, basement, roof, and vertical chases for flues, condenser-water piping, etc. |
| Value of any equipment screening | Many times municipalities require screening for any equipment mounted on grade or on the roof |
| Cost of financing | Amount of project that is financed at the loan interest rate of the duration of the loan |
| Construction permits and fees | Typically a percentage of construction |
| Life of major equipment overhauls and replacement costs | This could include the replacement or overhauls of chillers, cooling towers, boilers, etc., over the life of the analysis/contract duration. If the district-energy contract is for 20 years and a piece of equipment must be replaced or overhauled (i.e., cooling tower replaced after 15 years or chiller-condenser-water tube replacement) this cost must be accounted for |
| Contract vs. installed capacity | The district-energy capacity will most likely be less than the planned in-building installed capacity, as dictated by the consultant due to many reasons, but mostly over sizing and diversity. Typically, the estimated peak loads can be reduced to 70% |
| Cost of redundant equipment for emergency or standby capacity | Similar to above, *N*+1 redundancy requirements would be accommodated and added to the first cost |
| **Energy and Utility Costs** | |
| Electric rate | From usage of each option from energy model or other estimate |
| Natural gas rate | From usage of each option from energy model or other estimate |
| Water and sewer charges for steam, chilled, and condenser-water systems | From usage of each option from energy model or other estimate. Water is increasingly becoming an important resource, hence makeup water and equipment blowdown/sewer discharge amount are estimated. It is not uncommon for this utility to have a different escalation rate |
| **Operations and Maintenance Costs** | |
| Labor and benefits of operations staff assigned to central plant activities | This would include any staff that is assigned to the duties of maintaining and operating the central plant including supervisors and overtime due to unplanned outages |
| Replacement or refilling of refrigerants | If refrigerant is scheduled for phasing out, chillers must be retrofitted to accept new refrigerant plus any topping off of refrigerants (or replacement) |
| Spare parts and supplies | Chiller and boiler and auxiliary equipment require replacement of parts for normal maintenance procedures including gears, oil, tubes, etc. |
| Cost of chemical treatment for steam, chilled, condenser, and hot-water systems | Includes scale and corrosion inhibitors, biocides, oxygen scavengers, etc., and these costs could be considerable |
| Cost of contracted maintenance | Some owners outsource specific tasks to service companies such as chiller or boiler maintenance and overhauls |
| **Energy and Resource Usage** | |
| Peak heating and cooling thermal loads | Used to apply the energy demand rate and the sizing of the plant equipment (chillers, boilers, pumps, electrical service, water service, etc.) |
| Annual heating and cooling usage | Used to apply the energy consumption rate of the utilities to the equipment meeting the thermal loads |
| Annual water and sewer usage | Quantify makeup water usage and blowdown discharge pertinent to the cooling towers and boilers |
| **Other Costs** | |
| Architectural and engineering design services | Specifically for new or retrofit applications |
| Fees and licenses | Air and water permits, high-pressure steam operator licenses, city franchise fees for running piping in street, etc. |
| Insurance of equipment | Typically a percentage of construction costs |
| Water and sewer charges for steam, chilled, and condenser-water systems | From usage of each option from energy model or other estimate. Water is increasingly becoming an important resource, hence makeup water and equipment blowdown/sewer discharge amounts are estimated. It is not uncommon for this utility to have a different escalation rate |

- Other than a possible demand charge, a customer is only billed for the energy used (metered)
- Less green house gas emissions and a lower carbon footprint
- Not having any equipment in hot standby that is idling and using energy
- Freeing up maintenance staff to perform other duties other than central plant operations

Of course, the building owner would have to determine from the above list if any or all of the parameters would be pertinent or valuable to his building(s).

Refer to Chapter 37 of ASHRAE (2015) for a more detailed explanation of preparing a life-cycle cost calculation.

**Example 2.1:** A building owner is evaluating two different methods of providing chilled water for cooling an office building: purchasing chilled water from a local district cooling provider or installing a conventional chiller plant. The building load is estimated to be 2400 tons with an annual cooling load of 6,264,000 ton-hours. The contract is for 25 years, the discount rate is 5.5%, and all costs except water/sewer will be escalated at 3.5% with water/sewer escalated at 10% per year (based on historical data from municipality).

For Alternative 1 (Purchase District Cooling), the district cooling provider charges are $285 capacity charge (dollars/ton applied to annual peak load) and $0.13 consumption charge (dollars/ton-h) for a total annual energy cost of $1,498,320. The interconnection charge (heat exchanger, piping, instrumentation, etc.) to the district cooling entity is $289,500. The building owner has decided to pay for this cost by financing it over the life of the contract in lieu of a lump sum basis.

For Alternative 2 (Self-Generated from In-Building Chiller Plant), the on-site chilled-water plant (chillers, pumps, piping, and cooling tower, etc.) is estimated to cost $8,981,000, with an expected life of 25 years and 90% of the cost will be financed at 5.5% interest rate. The owner has instructed his engineer to increase the chiller sizes (three at 900 tons) to accommodate for any future growth, add a little redundancy, and compensate for the aging of equipment. The cost estimate reflects these instructions. An escrow account for major chiller plant overhaul ($400/ton) was established and is expensed annually. Annual costs for preventative maintenance ($6.00/ton for electrical chillers) were obtained from the local chiller vendor; there is one and a half full time equivalent personnel (supervisor and operator) assigned to the plant with an annual salary of $99,000 (includes benefits burden of 40%) each, and water and sewer charges are at $4/1000 gallons for water and $4/1000 gallons for blowdown to sewer and chemical treatment ($0.0025/ton-h). The cost of insurance per year is based on 0.75% of the total construction costs.

The chiller plant (chillers, cooling towers, pumps, etc., but not including the distribution pumps) uses 4,389,950 kWh annually at a blended electrical rate of $0.11/kWh. Through an energy analysis, it has been determined that a district energy connection will reduce the electrical demand dramatically with a new blended rate to $0.0875/kWh. If all costs are to be escalated to keep pace with inflation, which option has the lowest life-cycle cost and the lowest net present value?

**Solution.** Table 2.4 provides the annual utility consumption for the two alternatives, while Table 2.5contains the maintenance and utility-cost comparison for each alternative, and Table 2.6 provides a comparison of the capital costs. The entire LCC analysis is summarized in Table 2.7 and Table 2.8. For the values provided, Alternative 1 has a 25-year present value (PV) life cycle cost of $25,575,051 and Alternative 2 has a 25-year PV life cycle cost of $26,171,587. While these two values are very similar and could be considered equal, if PV is the only basis for the decision, Alternative 1 is preferable because it has the lower life-cycle cost of $596,536 and saves close to $9,000,000 in initial installation costs.

Figure 2.2 summarizes the major cost components for the on-site generation of chilled water graphically. It is important to note how much the total costs are composed of water and sewer costs and that the energy costs only make up one-third of the life-cycle costs. Table 2.6 summarizes the construction costs for each option.

**Table 2.4** Annual Utility Consumption Summary for the Alternatives

| Utility | Alternative 1 | Alternative 2 |
|---|---|---|
| Electrical (kWh) | 0 | 4,389,947 |
| Water (1000 gallons [3785 L]) | 0 | 16,181 |
| Sewer (1000 gallons [3785 L]) | 0 | 3,773 |

**Table 2.5** Estimates of Annual Maintenance and Utility Costs

| Cost Category | Alternative 1 | Alternative 2 |
|---|---|---|
| Chiller O&M | $0 | $32,460 |
| Chiller Replacement Escrow | $0 | $43,200 |
| Operator Salaries | $0 | $99,000 |
| Estimated Water & Sewer | $0 | $78,816 |
| Estimated Chemicals | $0 | $15,660 |
| Insurance | $0 | $67,358 |
| Total O&M, Except Electricity | $0 | $336,494 |
| Electric Utility Cost | ($182,316) | $438,995 |

**Table 2.6** Capital Cost Estimate for the Two Alternatives

| Description of Cost Category | Alternative 1 | Alternative 2 |
|---|---|---|
| Electrical Equipment | $18,000 | $520,000 |
| Chilled-Water Side Equipment and Piping | $140,000 | $1,650,000 |
| Condenser-Water Side Equipment and Piping | $0 | $2,075,000 |
| Plumbing (floor drains, makeup water piping, etc.) | $5,000 | $131,000 |
| Miscellaneous Mechanical (HVAC, lighting, for equipment rooms) | $5,000 | $120,000 |
| General Construction (walls, floors, equipment pads, cooling-tower support, etc.) | $15,000 | $1,146,000 |
| Controls (BMS, instrumentation, startup, commissioning) | $8,000 | $325,000 |
| **Total Construction Costs** | $191,000 | $5,967,000 |
| Contractors Fee (7%) | $13,000 | $420,000 |
| Construction Contingency (10%) | $19,000 | $595,000 |
| **Subtotal** | $223,000 | $6,965,000 |
| Contractors General Conditions (10%) | $22,300 | $696,500 |
| **Subtotal** | $245,300 | $7,662,000 |
| Sales Tax (5.6%) | $14,000 | $429,000 |
| **Subtotal** | $259,300 | $8,091,000 |
| Consultant Design Fees (9.0%) | $25,000 | $728,000 |
| Permits and Fees (2%) | $5,000 | $162,000 |
| **Total** | $289,500 | $8,981,000 |

**Table 2.7** LCC for the Purchased Chilled-Water Alternative (Alternative 1)

| Alternative 1: Purchase Chilled Water from District Cooling Provider | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|
| | Year | | | | | | | | | |
| | **0** | **1** | **2** | **3** | **4** | **5** | **6** | **7** | **8** | **9** |
| Recurring Costs | $0 | ($37,298) | ($39,352) | ($41,381) | ($43,583) | ($45,869) | ($48,228) | ($50,670) | ($53,197) | ($55,820) |
| Chilled-Water Cost | $0 | $1,586,876 | $1,634,449 | $1,683,448 | $1,733,917 | $1,786,043 | $1,839,587 | $1,894,736 | $1,951,539 | $2,010,207 |
| Net Annual Cash Flow | $0 | $1,586,876 | $1,634,449 | $1,683,448 | $1,733,917 | $1,786,043 | $1,839,587 | $1,894,736 | $1,951,539 | $2,010,207 |
| | **10** | **11** | **12** | **13** | **14** | **15** | **...** | **23** | **24** | **25** |
| Recurring Cost | ($58,527) | ($61,329) | ($64,228) | ($67,238) | ($70,345) | ($73,560) | — | ($99,468) | ($103,702) | ($108,071) |
| Chilled-Water Cost | $2,070,471 | $2,132,542 | $2,196,474 | $2,262,506 | $2,330,334 | $2,400,195 | — | $2,951,997 | $3,040,495 | $3,131,393 |
| Net Annual Cash Flow | $2,070,471 | $2,132,542 | $2,196,474 | $2,262,506 | $2,330,334 | $2,400,195 | — | $2,951,997 | $3,040,495 | $3,131,393 |
| 25-year Net Present Value $25,575,051; Sum of 25-year Cash Flows $52,991,396 | | | | | | | | | | |

***Source:*** *2020 ASHRAE Handbook—HVAC Systems and Equipment*

**Table 2.8** LCC for the On-Site Generated Chilled-Water Alternative (Alternative 2)

| Alternative 2: Design and Install On-Site 2700 ton (8440 kW) Chilled-Water System | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|
| | Year | | | | | | | | | |
| | 0 | 1 | 2 | 3 | 4 | 5 | 6 | 7 | 8 | 9 |
| Recurring Cost | $0 | $923,570 | $934,797 | $946,416 | $958,442 | $970,924 | $983,807 | $997,141 | $1,010,941 | $1,025,264 |
| Energy Cost | $0 | $517,264 | $535,356 | $554,080 | $573,459 | $593,572 | $614,333 | $635,820 | $658,058 | $681,138 |
| Water Cost | $0 | $1,535,306 | $1,573,015 | $1,612,550 | $1,654,028 | $1,697,695 | $1,743,444 | $1,791,539 | $1,842,133 | $1,895,547 |
| Sewer Cost | $0 | $923,570 | $934,797 | $946,416 | $958,442 | $970,924 | $983,807 | $997,141 | $1,010,941 | $1,025,264 |
| Net Annual Cash Flow | $0 | $517,264 | $535,356 | $554,080 | $573,459 | $593,572 | $614,333 | $635,820 | $658,058 | $681,138 |
| | 10 | 11 | 12 | 13 | 14 | 15 | ... | 23 | 24 | 25 |
| Recurring Cost | $1,040,047 | $1,055,348 | $1,071,184 | $1,087,620 | $1,104,585 | $1,122,143 | — | $1,263,626 | $1,286,746 | $1,310,609 |
| Energy Cost | $704,961 | $729,618 | $755,137 | $781,622 | $808,959 | $837,253 | — | $1,065,245 | $1,102,502 | $1,140,956 |
| Water Cost | $184,634 | $203,084 | $223,377 | $245,763 | $270,322 | $297,335 | — | $579,459 | $637,364 | $700,871 |
| Sewer Cost | $22,029 | $22,800 | $23,597 | $24,425 | $25,279 | $26,163 | — | $33,288 | $34,452 | $35,654 |
| Net Annual Cash Flow | $184,634 | $203,084 | $223,377 | $245,763 | $270,322 | $297,335 | — | $579,459 | $637,364 | $700,871 |
| 25-year Net Present Value $26,171,587; Sum of 25-year Cash Flows $53,616,410 | | | | | | | | | | |

***Source:*** *2020 ASHRAE Handbook—HVAC Systems and Equipment*

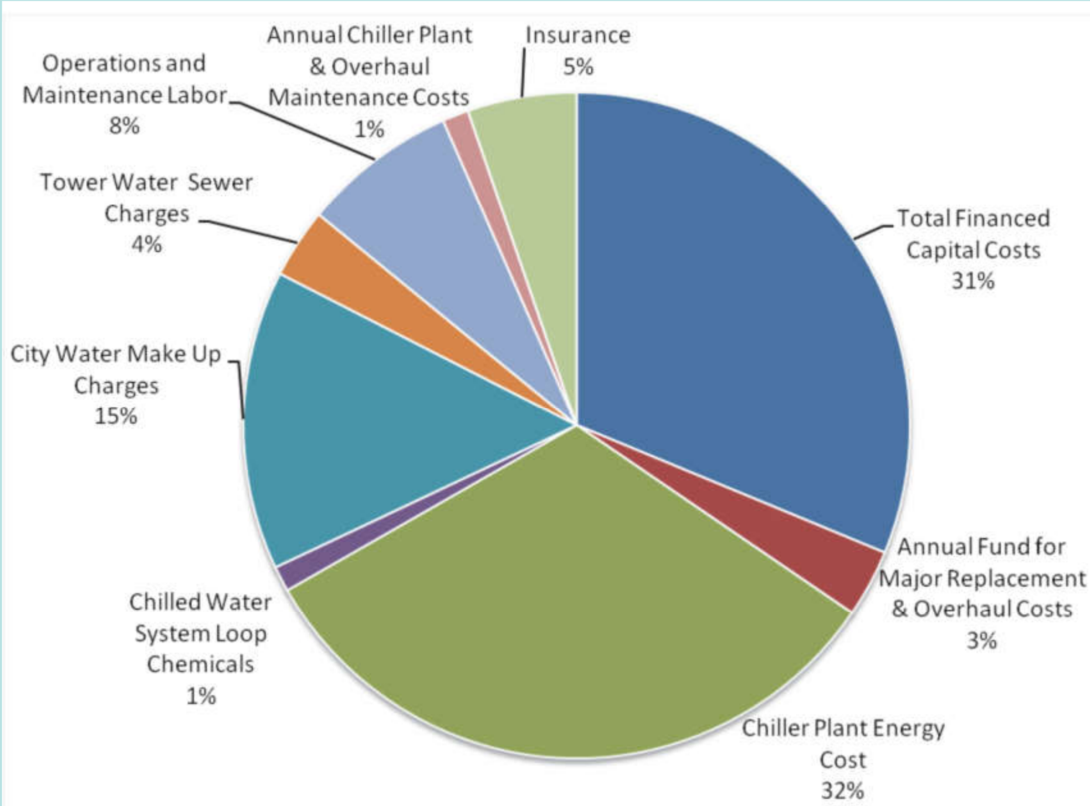



**Figure 2.2** Cost breakdown for on-site generation of chilled water.

Hence, in this case study example, district cooling is not only competitive in pricing with self-generation on a life cycle basis, but has the lower PV over 25 years.

## CONCLUSION

Master planning is as much art as science. It is only as good as the knowledge, creativity, and interest of the individuals that completed the tasks outlined above and, how capable they are of establishing strong client/engineer relationships that lead to honest and open communication. The owner must be free to challenge the engineer, and concurrently the engineer must be straightforward in discussing sensitive issues that challenge the owner. The most effective plans are built on communication, credibility, integrity, and trust of the parties involved.

Forty years of CHW system master planning in systems ranging from 300 to 30,000 tons supports the notion that master plans can have long lives and serve clients well for years and even decades. To be sure of their success, master plans need updating as the database is modified by the growth of the owner's requirements and facilities, new technology or system concepts should be developed, and there are delays in implementing the prioritized plan. If there has been an understanding of engineering fundamentals well supported by the data, system flow and control diagrams developed, system hydraulic models constructed in the plan, the master plan updates are straight forward and relatively low in cost.

It is recommended at the early planning phase in a project, an experienced designer be consulted if those undertaking the master plan and design do not have wide experience with DCSs. Early in the design process, wide and varied experience will normally provide large dividends.

## REFERENCES

ASHRAE. 2009. *Pocket Guide for Air-Conditioning, Heating, Ventilation and Refrigeration*, 7th Ed. Atlanta: ASHRAE.

ASHRAE. 2013. ASHRAE owning and operating cost database. www.ashrae.org/database.

ASHRAE. 2015. *ASHRAE Handbook—HVAC Applications*. Atlanta: ASHRAE.

ASHRAE. 2016. *ASHRAE Handbook—HVAC Systems and Equipment*. Atlanta: ASHRAE.

Alvarado, J.L., B. Jones, C. Marsh, D. Kessler, C. Sohn, C. Feickert, G. Phetteplace, E. Crowley, R. Franks, and T. Carlson. 2008. Thermal performance of microencapsulated phase change material slurry. ERDC TR-08-4. Washington, DC: US Army Corps of Engineers, Engineer Research and Development Center.

Bahnfleth, D.R. 2004. A utility master planning pyramid for university, hospital, and corporate campuses. *HPAC Heating, Piping, Air Conditioning Engineering* 76(5):76–81.

Bøhm, B. 1986. On the optimal temperature level in new district heating networks. *Fernwärme International* 15(5):301–06.

Bøhm, B. 1988. *Energy-economy of Danish district heating systems: A technical and economic analysis*. Lyngby, Denmark: Laboratory of Heating and Air Conditioning, Technical University of Denmark.

Electrowatt-Ekono Oy and FVB District Energy. 2002. Optimization of cool thermal storage and distribution. 2002:S5. Sittard, Netherlands: Netherlands agency for energy and environment (NOVEM), operating agent for International Energy Agency (IEA) District Heating and Cooling Project.

Hansen, T. 2002. *Behavior of ice slurries in thermal storage systems*. ASHRAE Research Project RP-1166 final report. Atlanta: ASHRAE.

Hyman, L.B. 2010. What building owners and designers need to consider for utility master planning for multiple building sites. *Proceedings of ASHRAE Annual Meeting*, June 23–27, Albuquerque, NM.

Koskelainen, L. 1980. Optimal dimensioning of district heating networks. *Fernwärme International* 9(4):84–90.

Meckler, M. 1997. Applying cost effective combined cycle/TES cogeneration to district heating and cooling. *Proceedings of the 1997 Annual Conference of the International District Energy Association* (IDEA), Westborough, MA.

Meckler, M. and L. Hyman. 2010. *Sustainable on-site CHP systems, design, construction, and operations*. New York: McGraw Hill.

Mornhed, G., D. Ballou, D. Murphy, and J-Y. Divoux. 1992. District cooling with gas turbine driven ammonia chillers. *Proceedings of the 1992 Annual Conference of the International District Energy Association* (IDEA), Westborough, MA.

Olama, A. 2017. *District Cooling, Theory and Practice*, CRC Press, Taylor & Francis Group, Boca Raton, FL.

Orlando, J. 1996. *Cogeneration design guide*. Atlanta: ASHRAE.

Phetteplace, G. 1989. Simulation of district heating systems for piping design. *Proceedings of the International Symposium on District Heat Simulation*, April 13–16. Reykjavik, Iceland.

Phetteplace, G. 1994. Optimal design of piping systems for district heating. Ph.D. dissertation, Stanford University, Stanford, CA.

Stewart, W.E. and C.L. Dona. 1987. Water flow rate limitations. *ASHRAE Transactions* 93(2):811–25.

Sweetser, R., G. Foley, and J. Freihaut. 2015. *Combined heat and power design guide*. Atlanta: ASHRAE.

UNEP. 2015. District energy in cities, unlocking the potential for energy efficiency and renewable energy. Paris: United Nations Environmental Programme.

Weinspach, P.M. 1996. Advanced energy transmission fluids for district heating and cooling. 1996:N2. Sittard, Netherlands: Netherlands Agency for Energy and Environment (NOVEM), Operating Agent for International Energy Agency (IEA) District Heating and Cooling.

# 3 Central Plant

District cooling systems (DCSs) have been introduced in several regions of the world. Most notably, the United States, the Middle East, and the Arabian Gulf area are currently involved in large district cooling plants (DCPs) to serve new development in a centralized approach rather than individual building solutions. It should also be noted that DCSs are gaining popularity in Europe as well; Werner (2017) estimates that there are 150 DCSs operating in Europe. (See Chapter 1 for more information on the history and current status of DCSs worldwide.) The centralized approach aims at optimizing energy, minimizing maintenance costs, and reducing peak power demands, compared to a decentralized approach with each building having its own cooling plant. Several approaches have been made by designers to achieve the intended goals; the DCS design approaches will be affected by several parameters, including temperature difference, land configurations, supply temperature, energy availability and cost, water availability and cost, etc.

This chapter will discuss district central plant components, chiller types, different refrigerants, pump types, chiller arrangements, pumping arrangements, plant-specific design issues, and cooling tower water as adopted in district networks.

## PLANT COMPONENTS AND ALTERNATIVE ARRANGEMENTS

A DCS consists of a chilled-water (CHW) production plant, hereinafter defined as a district cooling plant (DCP), distribution piping system, and consumer interconnection often called energy transfer stations (ETSs) if applicable.

The DCP(s) is the heart of the system and contains all major plant components with the distribution system being the main delivery arteries. The successful performance of DCSs depends on how such components are selected and how they interact with one another. Designers should have the proper background regarding plant components and their characteristics, energy and water consumption, efficiencies, operation and maintenance requirements, and life expectancy to achieve the most reliable and efficient design. More importantly, however, is that each of the components is integrated into the DCS to achieve the total system efficiency. For no matter how efficiently the plant is designed, it is only as efficient as its weakest link.

The DCP(s) major plant components are:

- Chillers
- Heat rejection component (cooling towers/radiators)
- Pumps

- Air separators and air-elimination devices
- Expansion tanks
- Chilled/condenser-water filtration system
- Chemical treatment
- Controls (See Chapter 7)

Proper attention should be paid to the plant component's environmental impact. Not only must the plant's impact on its surroundings be considered, but also the plant's environment must be safe and in an environmentally friendly condition for the operators and the surrounding area.

## TEMPERATURE DESIGN BASIS FOR THE CENTRAL PLANT

The temperature difference ($\Delta T$) between CHW supply and return plays an important role in the plants as well as the pumping system's design. The $\Delta T$, as this temperature differential is often called, is largely determined by the consumer's equipment, as is discussed in Chapter 5 on the end user interface or ETS. While the design of the consumer's equipment is discussed in detail in Chapter 5, some key points warrant repeating here because of their impact on efficient system design.

To minimize pumping power consumption, flow rate should be reduced to the minimum possible. The flow-rate reduction is a function of the system temperature differences and the load. The higher the system $\Delta T$, the lower the circulated flow rate for a given load. To maximize $\Delta T$ and thus plant efficiency, the return temperature should be as high as possible, and the supply temperature should be as low as possible.

On the other hand, designing for too high of a $\Delta T$ (high return water temperature) may impact end user cooling coil performance and impact the coil's ability to maintain humidification requirements due to less flow than expected. Furthermore, lower flow rates correspond to deeper cooling coil rows creating additional air pressure drops. Deeper coils are also more difficult to clean. Therefore, selecting coils that are a maximum of eight rows and preferably six rows deep will help keep coil pressure drops in acceptable ranges and ensure they will still be cleanable.

Figure 3.1 shows the reduction in the ability to dehumidify if the flow is less than 40% of design. It is currently possible with several manufacturers to select chillers with a

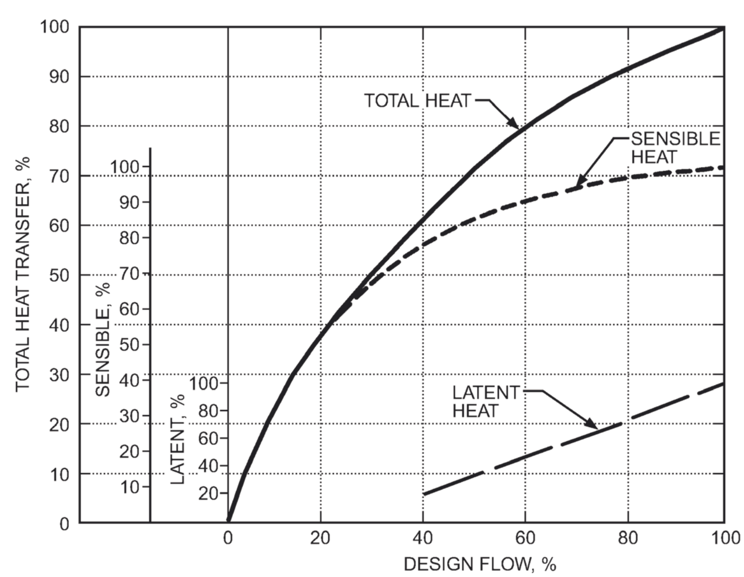



**Figure 3.1** Chilled-water coil capacity (Chapter 13, ASHRAE 2016d).

low supply temperature. However, it is recommended that supply temperature should not go below 34°F–35°F (1.1°C–1.7°C) to avoid freezing in the evaporator, especially in variable-flow systems. Lower supply temperatures, if required, may be achieved via circulation of a glycol solution; however, this is normally cost prohibitive due to the initial and operating costs of glycol solution.

When stratified CHW thermal energy storage (TES) is used, care should be taken to maintain the chilled-water temperature above 39.2°F (4°C) to allow proper temperature stratification within the TES tank due to water density differences. Water is most dense at 39.2°F (4°C); therefore a TES tank charging temperature lower than 39°F (3.9°C) without low temperature additives will result in mixing within the tank, loss of tank stratification, and may disturb the system supply temperature and the storage concept due to the water density differentials. Refer to Chapter 6 for additional information on TES.

Due to these considerations, it is normal to find that the majority of CHW DCPs are designed for a 40°F (4.4°C) supply water temperature and a 56°F (13.3°C) or higher return water temperature. Some plants may be designed for temperatures lower than 40°F (4.4°C) in order to serve high-rise towers using multiple cascaded heat exchangers in series. Some plants are designed for a higher return temperature (up to 60°F [15.5°C] or higher) to increase the $\Delta T$, but there is a physical limit, and there must be a great deal of additional coordination with the design of the customer's building's HVAC system to ensure the system operates per design intent.

The designer should establish the design return temperature based on what the customer's leaving water temperature is expected to be. Selecting a design temperature too high compared to what the system will achieve (low $\Delta T$ syndrome) results in piping and pumping that is undersized to serve those conditions. The system should be designed with adequate flexibility to operate in all operating conditions including low return water temperatures.

## CHILLER BASICS

### Chiller Types

Chiller types used in DCSs vary from one location to another depending on different parameters, including water availability, power availability, maximum power demand that can be offered by the utility, steam availability, gas availability, fuel oil availability, distribution temperature required, plant location with respect to development, and applied environment impact regulations including pollution and noise control, etc.

The chillers may be classified according to:

- Heat rejection source
- Driving energy source
- Supply temperature required
- Capacity

There are two methods of rejecting chiller-condenser heat: air cooled or water cooled. Air-cooled chillers are typically of the packaged type where the controls, compressors, evaporator, and air-cooled condenser are all on the same skid. The capacity of such chillers may go up as high as 450 tons (1600 kW), and they have been implemented in several central plants by installing multiple units in chiller farms, either on-grade or on building rooftops. The unit transfers absorbed building heat indirectly to the ambient. If properly maintained, the life span of such equipment is typically around 15 years, depending on ambient temperature and annual run time (e.g., the hotter the climate, the more run time hours at severe duty conditions, and hence a shorter life). Care should be taken when selecting an air-cooled chiller so as not to use just the design dry-bulb temperature as the ambient temperature. Consideration should be given to if the unit(s) are located on-grade or on a rooftop and how the characteristics of these areas add to the ambient temperature. This is especially true in a chiller farm because the discharge of one chiller may affect the inlet of another, thus derating it. The manufacturer's recommendations should be followed in selecting chillers and in making sure there are adequate clearances between adjacent chillers and structures.

Water-cooled chillers use water as the media to remove the heat from the chiller condenser. The heat carried by the water circuit is either rejected through bodies of water (rivers, lakes, and oceans), cooling towers (evaporative cooling), or radiators. The limiting factor of the capacity of a cooling tower is the ambient wet-bulb temperature (WB). Since the WB is lower than the coincident dry-bulb temperature (DB), using a cooling tower with a water-cooled chiller is more efficient at removing chiller heat due to lower condensing temperature when compared to air-cooled ones. Therefore, water-cooled chiller plants are typically a great deal more efficient than either air-cooled or water-cooled radiator plants and require less heat rejection area. The water-cooled chillers in the Arabian Gulf consume ~0.7 kW/ton (0.2 kW/kW) versus ~1.7 kW/ton (0.5 kW/kW) for the air cooled. The heat rejection area excluding surrounding free spaces around an area ~0.04 $m^2$/ton versus ~0.07 $m^2$/ton for air cooled. Hence, water-cooled plants are used in the majority of large DCPs.

Depending on the refrigerant selected, a single water-cooled chiller may be as large as 8500 tons (30,000 kW) per unit with some manufacturers. However, the most cost-effective size in the market tends to be around 2500 tons (8800 kW) to 3000 tons (10,550 kW) as the larger sizes are of the field erected industrial type and are more expensive on a unit capacity basis than a packaged unit or multiple packaged units piped together. The expected life span of industrial units is over 40 years.

The driving energy source, or chiller prime driver, can be electricity, steam, hot water, or natural gas driven. There are also two basic types of chillers—vapor compression and

**Figure 3.2** 4500-ton, field-erected electric- (4160V/60 Hz) driven chiller (HFC-134a).
*Photo Courtesy of Burns & McDonnell Engineering Company, Inc*

thermodynamic. Vapor-compression chillers use refrigerants and the reverse Rankine cycle and can have prime drivers of electric motors, steam turbines, or gas engines, which can be constant-speed or variable-speed driven.

Vapor-compression chillers have multiple compressor options, including reciprocating, scroll, rotary screw, or centrifugal. This chapter will limit the compressor discussion to centrifugal compressors because the typical large capacity chiller required in a DCP excludes other compressor types. (Refer to Chapter 38 of the 2016 edition of *ASHRAE Handbook—HVAC Systems and Equipment* for additional information.)

Thermodynamic or heat-driven chillers are typically of the absorption cycle that typically use a lithium bromide solution as the absorbent and water as the refrigerant. The firing methods or prime driver for absorption chillers may be indirect-fired using low to high temperature water and low to high pressure steam or direct-fired with a gas burner or the exhaust from a boiler or combined heat and power generator (gas turbine generator or reciprocating generator).

Absorption chillers may also be single effect (one stage), double effect (two stage), or triple effect. The processes are similar, but the multiple effect absorbers have additional solution generators that increase the cycle efficiency. (Refer to Chapter 18 of the 2018 edition of *ASHRAE Handbook—Refrigeration* for more details.)

Each of these technologies is explained in more detail in this chapter.

## Chiller Performance Limitations

The basic vapor-compression refrigeration cycle is shown in Figure 3.3. The coefficient of performance (COP) is a useful tool to characterize the equipment's performance and its efficiency and is defined for a chiller as the heat in divided by the work in:

$$\text{COP} = \frac{Heat_{in}}{Work_{in}} \tag{3.1}$$

where

COP = the coefficient of performance for the refrigeration cycle, dimensionless
$Heat_{in}$ = heat input in the evaporator, i.e., the cooling load, Btu/h (W)
$Work_{in}$ = work input to the refrigeration cycle, Btu/h (W)

For absorption systems, Equation 3.1 is modified where $Work_{in}$ is understood to include the heat input needed to drive the cycle.

The Carnot COP is the maximum COP that may be achieved by an idealized refrigeration cycle operating between an isothermal heat load and an isothermal heat sink. This

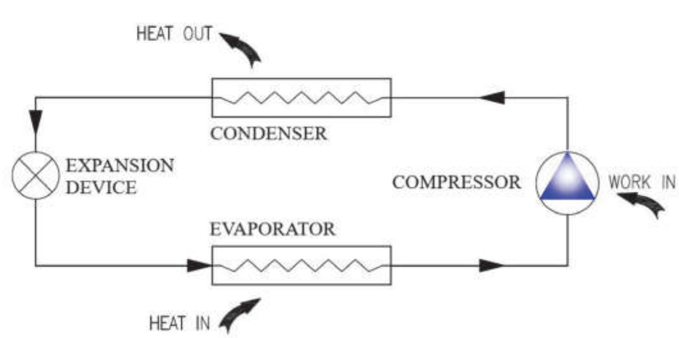



**Figure 3.3** Vapor-compression refrigeration cycle.

limitation is imposed by the Second Law of Thermodynamics. The Carnot COP can be expressed only in terms of the temperatures of the heat load (evaporator) and the heat sink (condenser):

$$COP_{Carnot} = \frac{T_{evap}}{T_{cond} - T_{evap}} \tag{3.2}$$

where

$COP_{Carnot}$ = the Carnot COP, dimensionless
$T_{evap}$ = evaporator (i.e., heat source) temperature, °R (K)
$T_{cond}$ = condenser (i.e., heat sink) temperature, °R (K)

Note that Equation 3.2 requires the use of absolute temperatures. Using Equation 3.2, it is a simple exercise to plot the $COP_{Carnot}$ as a function of the condensing and evaporating temperatures (see Figure 3.4). These are the maximums that are theoretically possible; real plants with all their attendant nonideal processes and inefficiencies will have significantly lower COPs as discussed below.

The work performed by the compressor is defined by lift. Lift (or head pressure) is defined as the difference between the condenser refrigerant pressure and the evaporator refrigerant pressure. Lift can also be approximated by using the difference between the entering condenser temperature and the leaving chilled-water temperature.

By examining Figure 3.4, it should come as no surprise that increases in condensing temperature or reductions in evaporator temperature result in lower $COP_{Carnot}$ values;

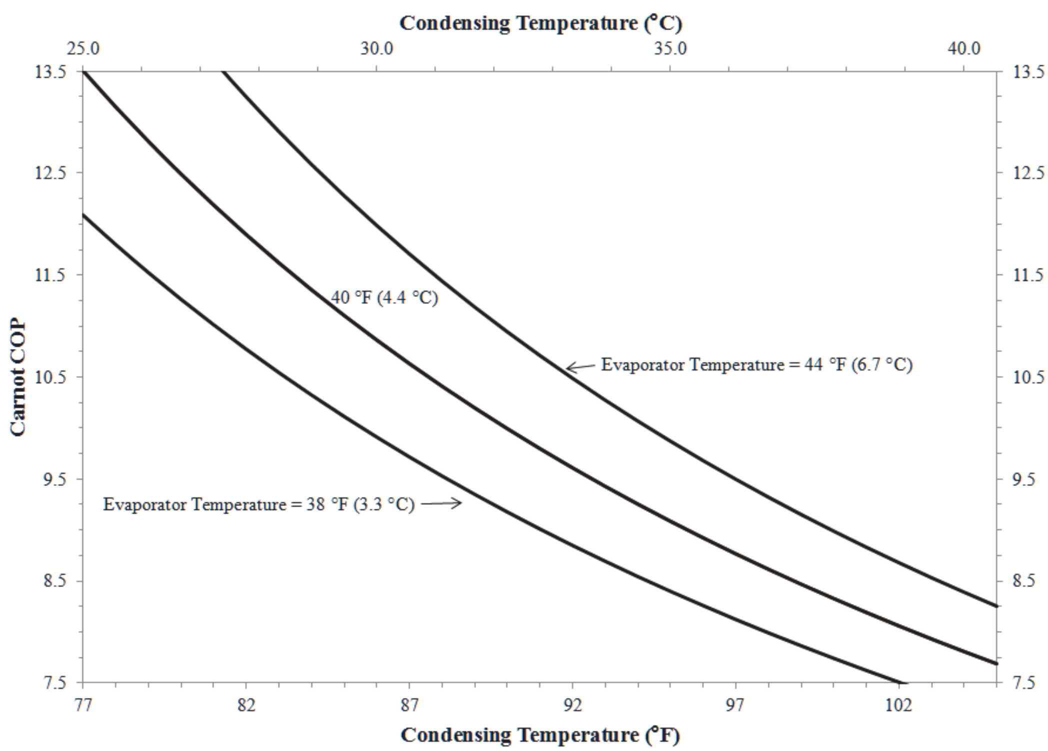



**Figure 3.4** Theoretical maximum (Carnot) COP as a function of condensing and evaporator temperatures.

either of these changes results in increased lift within the refrigeration cycle. Unfortunately for space cooling, there is little flexibility in either the condensing or evaporator temperatures as these are dictated by prevailing weather conditions and indoor comfort parameters. Approach temperatures within the condenser, evaporator, and other heat-transfer devices within the system (e.g., indoor cooling coils, cooling towers) will only increase the temperature difference or lift of the refrigeration cycle and thus reduce its $COP_{Carnot}$. In addition, the performance of a real refrigeration cycle is impacted by the refrigerant used and its relative cycle efficiency versus ideal. Furthermore, the efficiency of the prime mover (e.g., electric motor, engine, turbine, etc.) must also be factored into determination of the effective COP. When these parameters are taken into account, the COP of an electric vapor-compression chiller typically varies from about 4.7 to 7.0. The chiller has a corresponding kW/ton range of 0.75 to 0.50 kW/ton (0.21 to 0.14 $kW_e/kW_t$) and depends on the system-entering condensing temperature and leaving-evaporator temperature (i.e., compressor lift).

Table 3.1 shows the approximate range for COP of chillers types normally used in DCSs.

The system COP may be improved if heat-recovery equipment is used. Figure 3.5 illustrates a system with heat recovery. The system COP will be increased and the COP equation will be as follows:

$$\text{COP} = \frac{Heat_{out} + Heat_{in}}{Work_{in}} \tag{3.3}$$

where

COP = the coefficient of performance for the refrigeration cycle, dimensionless

$Heat_{in}$ = heat input in the evaporator, i.e., the cooling load, Btu/h (W)

$Heat_{out}$ = heat recovered (e.g., for hot-water heating), Btu/h (W)

$Work_{in}$ = work input to the refrigeration cycle, Btu/h (W)

In DCSs, the chiller is not the only component that consumes energy, but the entire integrated components of the plant including primary pumps, secondary pumps, condenser-water pumps, cooling towers, etc. In the Middle East, clients are targeting plants with an overall plant efficiency rating of 0.95 kW/ton (0.27 $kW_e/kW_t$), which includes the chillers, cooling towers fans, and chilled water and condenser water pumps. However, in the United States where conditions are not as extreme, modern DCS plants can typically achieve 0.8 kW/ton (0.23 0.27 $kW_e/kW_t$) or less. The differences can be explained by the higher distribution system pressure drops and therefore more pump energy due to the larger scale of the network to be pumped as well as the higher ambient temperature impact on chiller and cooling tower efficiencies experienced in the Middle East.

**Table 3.1** Chiller Technology Comparison Table

| | Vapor Compression Chillers | | | Absorption Chillers | |
|---|---|---|---|---|---|
| Type | Reciprocating | Screw | Centrifugal | Single-Stage | Two-Stage |
| Prime Driver | Electric Motor | Electric Motor | Electric Motor | Hot water, 65°C <temp, <80°C | Steam or fire, temperature > 170°C |
| Refrigerants Used | HFC, HFO, HCFC, $NH_3$, etc. | HFC, HFO, HCFC, $NH_3$, etc. | HFC, HFO, HCFC, $NH_3$, etc. | $H_2O$ with LiBr, $NH_3$ with $H_2O$ | $H_2O$ with LiBr, $NH_3$ with $H_2O$ |
| COP | 4–6 | 4–6 | >7.0 | 0.6–0.75 | 1.2 |

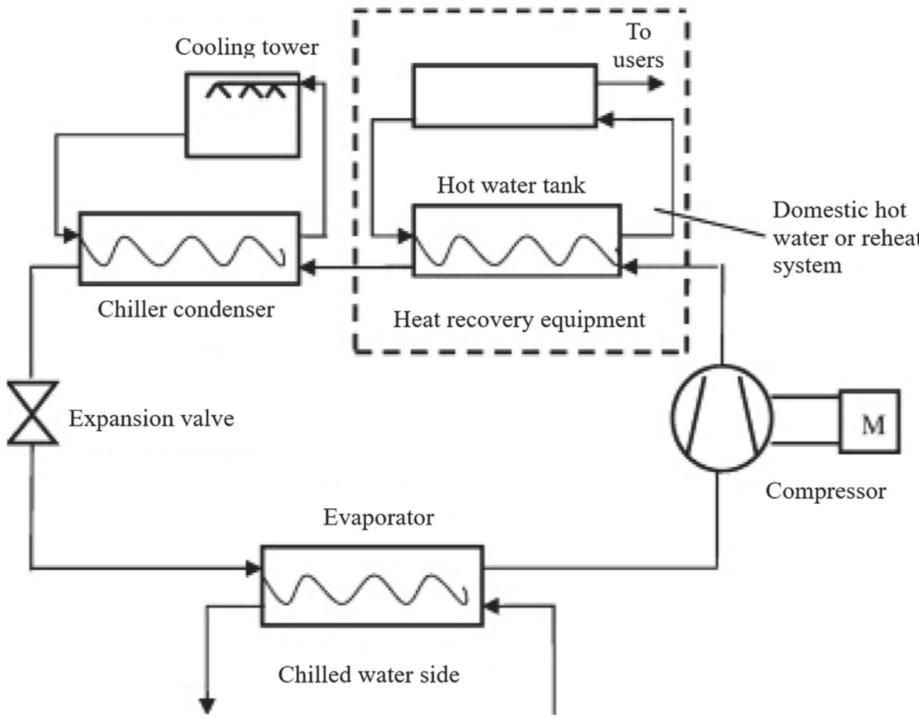



**Figure 3.5** Chiller with heat recovery.

Chillers are available in low, medium, and high voltage. Not all manufacturers can provide high-voltage chillers. In the Middle East, low voltage at 400 V for large plants is preferred versus medium voltage (3.3 and 6.6 kV) or high voltage (11 and 13.8 kV) at 50 Hz. In North America, low voltage is nominally 460 V, medium voltage is 5 kV (4.16 kV), and high voltage chillers are 15 kV (13.2 or 13.8 kV) at 60 Hz. While high-voltage motors will provide some economies to the project (less transformation and smaller wire size), the designer should be aware that high-voltage motors are expensive, special-order motors and take a long time to manufacture as well as repair. This fact is critical to any reliability concerns if a motor should fail. Furthermore, variable-frequency drives (VFD) for medium- and high-voltage chillers are more expensive and have a larger foot print than low voltage chillers.

One disadvantage of a low-voltage chiller is that they typically peak out at around 1400 tons for 50 Hz and 1750 tons at 60 Hz, whereas the higher voltage machines start around 400 tons at 50 Hz and 500 tons at 60 Hz and get larger. That is why it is common to see low-voltage chillers in tandem (series) pairs creating a larger capacity "chiller." An advantage of low-voltage units is that they have unit-mounted starters and VFDs, whereas higher voltage chillers require remote units impacting space and cost, while low-voltage motors are lower cost, readily available, and easier to repair.

The COP of any vapor-compression or absorption machine and the distribution-pumping energy depends on the temperature difference on which the evaporator and condenser operates. Increasing the evaporator temperature and reducing the condenser temperature will improve the refrigeration machine COP but also increase required pumping energy due to the higher flows created. The evaporator temperature depends on the load, chiller design, and distribution network length, whereas the condensing temperature will depend on prevailing weather, power and water cost, and any other available heat rejection sink parameters.

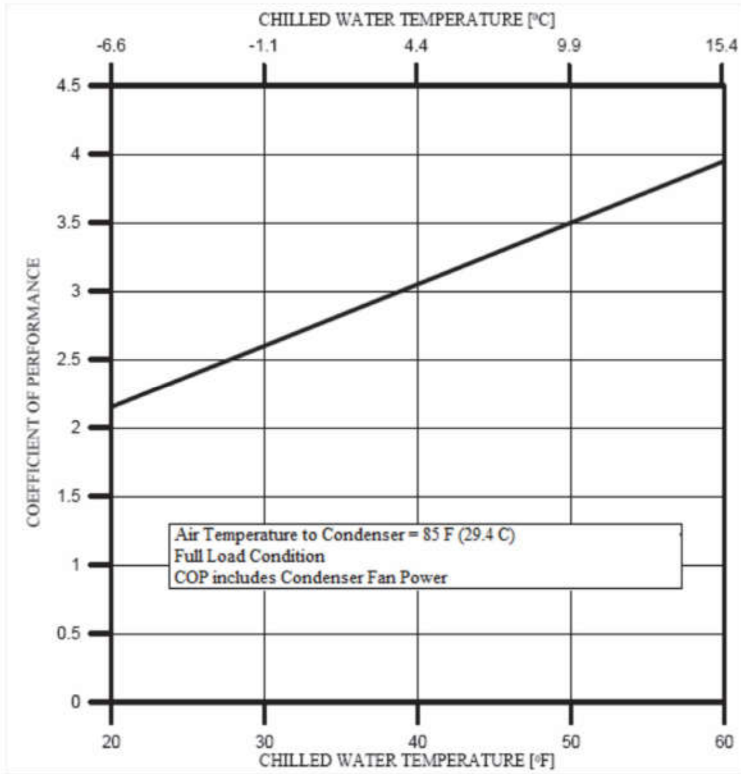



**Figure 3.6** Impact of chilled-water temperature on COP.

According to chiller manufacturers, each 1°F would save around 2% of chiller input energy (1°C would save around 4%). Figure 3.6 indicates the impact on the COP of centrifugal chillers when entering-condenser temperature is 85°F (29.4°C).

## Vapor-Compression Refrigerants Selection and Phase-Out Plans

The Montreal Protocol was written in 1987 to address the depletion of the ozone layer, banning the use and production of chlorofluorocarbon (CFC) refrigerants by January 1, 1996, as well as to establish a phase-out plan for hydrochlorofluorocarbons (HCFCs), which were defined as transitional refrigerants. This also affected the production of HCFC-123, whose production has been continually reduced since 2010 and will entirely be phased out of use in new equipment by January 1, 2020, and completely phased out of production by 2030. Similarly, in 1997 the Kyoto Protocol extended the phase out of refrigerants to address and reduce the production of greenhouse gases (GHG).

Furthermore, in October 2016 the Kigali Amendment to the Montreal Protocol was issued and soon after ratified by the United Nations Environment Programme (UNEP). The amendment extended the phase outs to hydrofluorocarbons (HFCs), including HFC-134a. Soon after in December 2016, the US Environmental Protection Agency (EPA) issued the latest Significant New Alternatives Policy (SNAP), which addressed the phase-out date of high GWP refrigerants including HFC-134a for January 2024 for all new equipment manufactured. SNAP was established under Section 612 of the Clean Air Act to identify and evaluate substitutes for ozone-depleting substances.

**Table 3.2** Characteristics of Commonly Used Vapor-Compression Refrigerants

| | Medium-Pressure Refrigerants | | | | Low-Pressure Refrigerants | | |
|---|---|---|---|---|---|---|---|
| Refrigerant | R-134a | R-513A | R-1234yf | R-1234ze(E) | R-123 | R-1233zd(E) | R-514A |
| Type | HFC | HFO Blend | HFO | HFO | HCFC | HCFO | HFO Blend |
| Flammability | 1 | 1 | 2L | 2L | 1 | 1 | 1 |
| Toxicity | A | A | A | A | B | A | B |
| Theoretical Fluid Efficiency | 8.5 COP | 8.3 COP | 8.2 COP | 8.5 COP | 9.4 COP | 8.85 COP | 8.91 COP |
| Capacity Change Compared to Base | Base | Similar | 5% Loss | 25% Loss | Base | 35% Gain | ~5% Loss |
| GWP | 1300 | 573 | <1 | <1 | 76 | <1 | 2 |
| ODP | 0 | 0 | 0 | 0 | 0.012 | ~0 | 0 |
| Atmospheric Life | 4900 days | 2200 days | 16 days | 11 days | 475 days | 26 days | 22 days |

*Source: Various sources including Trane (2017)*

Historically, the refrigerants of choice for much of the district cooling industry have been either HCFC-123 or HFC-134a. However, while currently still available and viable, both refrigerants are in the process of being phased out. With the pending departure of the "go-to" options, a new generation of refrigerants has emerged that further reduces the ozone depletion potential (ODP) and GWP compared to the previous options and banned CFCs.

The new generation low-GWP refrigerants required a new classification within ASHRAE Standard 34 of Class 2L for lower burning velocity Class 2 (slightly flammable) (ASHRAE 2016b). The new refrigerants are hydrofluoroolefins (HFO) or HFO blends, and HFO Blend 513A appears to be the current replacement for HFC-134a for medium-pressure refrigerants and HFO-514A the re-placement for HCFC-123.

It should be noted that HFO-514A is a drop-in replacement for HCFC-123 in a new chiller as long as it is properly designed for the future retrofit. However, in the Middle East, with the high lift conditions, HFO-1233zd(E) is the preferred refrigerant. Similarly, because HFO-513A is a blend containing HFC-134a, it also is a drop-in replacement for HFC-134a but with a minor loss in efficiency and is the preferred medium pressure replacement. Although HFO-1234yf has lower GWP values, it is still in the early stages of research and will require a compressor redesign prior to wide-scale implementation.

DCS designers should check local refrigerant suitability during the initial design phase of the project with vendors and local code authorities having jurisdiction (AHJ). Table 3.2 summarizes the fundamental properties of several viable refrigerants as well as their capacity loss or gain compared to the current base refrigerant.

DCS designers should also be aware that low pressure refrigerant chillers have an integral purge unit that purges noncondensables (mostly air) from the refrigerant circuit. The unit comes standard with the chiller but is something that must be maintained periodically since if the refrigerant accumulates noncondensables, then the condensing pressure increases and the chiller efficiency decreases. Moisture can also be an issue because it allows the formation of acids within the chiller that can damage the components such as motor windings and bearings.

Electrical-driven, water-cooled, centrifugal-type chillers (Figure 3.7) are globally quite common in the industry and are becoming the chiller of choice for many owners and engineers due to their high efficiencies, reliable operation, and lower unit pricing. Several technologies and prime drivers are discussed in the following sections.

**Figure 3.7** Electric centrifugal chiller.
*Courtesy of Carrier Corporation*

## Electrical-Driven, Water-Cooled Centrifugal Chillers

Electrical-driven, water-cooled centrifugal chillers use the principles of a vapor-compression cycle. These chillers are able of producing CHW temperatures below 35°F (1.7°C) and can tolerate entering condenser-water temperatures above 95°F (35°C). In smaller size ranges, the equipment is packaged and in larger sizes (typically industrial installations) the equipment must be field erected on site due to shipping limitations.

For electrical connections and spatial layouts, designers should be aware that above low voltages, starters cannot be unit mounted and must be remote mounted. A chiller may have a single compressor or dual compressors that extend the capacity range of the model. Furthermore, a VFD can be added to the unit to vary the compressor speed and save electrical energy.

While the use of higher voltage motors reduces the size and cost of the electrical system installation, there are disadvantages because of the long lead time for motors for the initial installation as well as any future maintenance repairs because rewinding high voltage motors is a highly specialized and time-intensive effort.

## Engine-Driven Chillers

Engine-driven chillers (Figure 3.8) use the same centrifugal chiller described in the previous section with the difference being that instead of using an electric motor to drive the compressor an internal combustion engine is used. There are two ways of using engines in chillers as prime drivers. One method uses the engine as a generator of electricity (similar to emergency power applications) that serves the compressor's electric motor. This is typical for a chiller that has a hermetic compressor that uses low-pressure refrigerants and cannot be direct coupled from the prime driver shaft to the compressor shaft. The other method has the engine directly coupled to the compressor, where the compressor is open type and is piped to the evaporator and condenser shells of the chiller. The engine can run on fuel oil (diesel) or natural gas (spark ignition). System designers must take into account noise and vibration as well as the combustion and ventilation air requirements due to the engine's needs and heat loss to the machine room.

Typical uses are where natural gas is available but there is not adequate electrical infrastructure to support a chiller. They are also used to reduce the electrical demand on the plant where on peak and off peak electrical rates are used. A directly engine-driven chiller is inherently variable-speed driven.

## Absorption Chillers

Absorption chillers are available in two types: lithium bromide-water (with water as the refrigerant and lithium bromide as the carrier) and water-ammonia (with water as the

**Figure 3.8** Engine-driven chiller.
*Courtesy of Tecogen*

carrier and ammonia as refrigerant). Lithium bromide (LiBr) chillers use a nontoxic salt solution as the absorbent and water as the refrigerant. An LiBr/$H_2O$ absorption cycle is based on two principles: first, LiBr solution, a salt, has a high affinity for water vapor; and second, water boiling in a flash process cools itself at a relatively low temperature when subjected to low absolute pressure (high vacuum), which is typically 0.25 in. Hg (6.4 mm Hg) absolute. This low pressure is equivalent to the refrigerant (water) at a saturated temperature of around 40°F (4.4°C) and allows CHW supply to be produced above 42°F (6°C). The water-ammonia machine essentially has the same cycle as the LiBr-water chiller, with additional equipment to deal with the ammonia. The ammonia machines are typically utilized in industrial processes where lower-chilled water temperatures are required.

The absorption cycle is similar to the vapor-compression cycle, except that the compressors are replaced by pumps. The vapor from the evaporator is converted into a solution using the absorber. The absorption cycle is shown in Figure 3.9.

Chilled water is circulated through the tubes and the heat is rejected to the refrigerant in the shell side as it is vaporized. The vaporized refrigerant is absorbed into the LiBr solution in the absorber section and is pumped from there to the generator section where the diluted solution is heated to drive off the previously absorbed refrigerant. The refrigerant (water) is then condensed and returned to the evaporator.

The heat that is generated in the absorber section, as a result of the heat of condensation and heat of dilution between the LiBr and the $H_2O$, has to be removed to maintain the proper solution temperature. This heat combined with the heat that is generated in the condenser is rejected to the condenser water circuit and then to the atmosphere by means of cooling towers.

For direct-fired absorption chillers, the source of heat in the generator section is typically fuel oil or natural gas. A further energy source is the use of boiler exhaust or combustion gases from CHP equipment.

The use of absorption chillers has certain limitation in terms of the chilled and condenser-water temperatures:

- The lowest CHW supply temperature for stable operation is approximately 42°F (5.6°C). This is simply a function of the refrigerant and the machine's construction. In fact, there are few manufacturers that will guarantee a leaving-water temperature of 40°F (4.5°C) or lower, which may be required for district cooling application.
- On the condenser side, absorption chillers are usually rated at nominal tonnage for operation with an entering condenser-water temperature of 89.6°F (32°C).

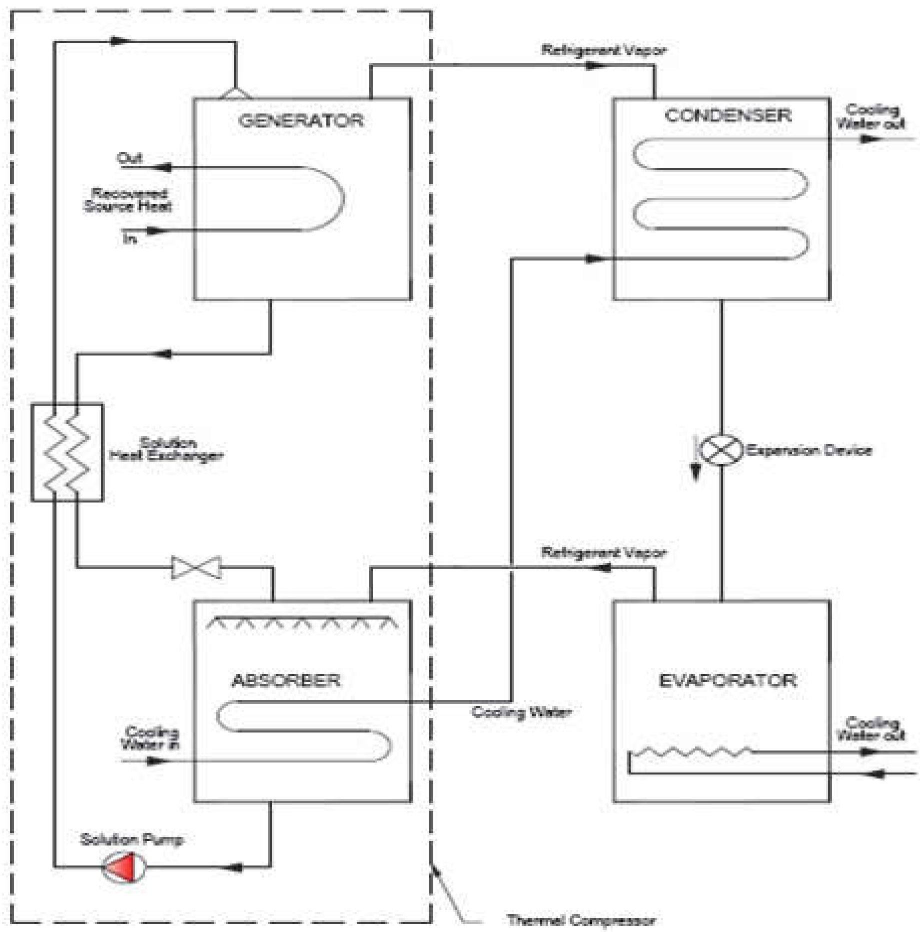



**Figure 3.9** Absorption cycle, single effect.

> The performance curves are derated up to 95°F (35°C) and the chiller is not rated beyond this point. At 95°F (35°C) the chiller's performance is approximately 10% less for every 1.8°F (1°C) above 89.6°F (32°C). For the Arabian Gulf coast and similar regions, the design wet-bulb temperature is approximately 86°F (30°C), which means that the cooling tower approach would have to be 3.6°F (2°C) to achieve an 89.6°F (32ºC) condenser-water supply temperature. This is not only impractical to achieve, but also extremely expensive.

The absorption chiller’s rejected heat is higher than electrically operated chillers. Therefore, gas-fired absorption chillers must have cooling towers sized for almost 40% more capacity compared to electrically driven chillers.

The indirect-fired chiller functions much the same as the direct-fired chiller, however it uses steam or hot water as its source of heat for the generation process.

Applications of absorption chillers must consider the following unique issues to ensure long chiller life span and to maintain its performance:

- Crystallization occurs when LiBr, as a function of temperature and concentration, begins to change from a liquid to a solid. When this happens, the piping within the machine can become plugged. A buildup of noncondensable gases can cause the LiBr concentration equilibrium to shift and hence cause crystalli-


**Figure 3.10a** Typical direct-fired absorption chiller.
*Courtesy of Kawasaki*


**Figure 3.10b** Typical steam-fired absorption chiller.
*Courtesy of Kawasaki*

zation. Crystallization is also caused when cooling tower water is supplied too cold or the entering condenser-water temperature decreases rapidly. Absorption chillers usually cannot tolerate entering condenser-water temperatures lower than 75°F (24°C), which means that a bypass/blending line must be installed to ensure that the condenser-water supply temperature does not fall below this temperature.

- Noncondensable gases must be continuously purged to allow the machine to function as intended. Similar to a low-pressure vapor compression chiller, a buildup of noncondensable gases within the chiller will result in a loss of vacuum, which will drastically affect the chiller's performance.
- The shell side of the chiller must remain airtight and under a continuous vacuum. This is required to first allow the refrigerant to vaporize, and second, to prevent and retard corrosion on the inside of the machine. LiBr is a salt, and it has the characteristic of being extremely corrosive to steel, particularly in the presence of oxygen.
- Direct-fired absorption chillers may be considered as boilers and require trained personnel to operate, dependent on the steam operating temperature and local codes.
- Ammonia chillers for low-temperature applications use inhibitors to prevent corrosion. These inhibitors require special handling.
- Ammonia has a flammability range of 16%–25% by volume of air. It is also very toxic and if released to the atmosphere in large quantities may affect the local community.

Local chiller service organizations should be consulted to determine appropriate maintenance costs since they are higher than vapor-compression chillers; see Chapter 8 for a generalized discussion. One significant consideration in the use of absorption chillers is their higher heat rejection rates per unit of net cooling, which require significantly larger heat rejection equipment capacity (e.g., condenser-water piping and pumping, and cooling tower), and flows and thus increased costs that must be accounted for.

One of the absorption chiller features is the linear reduction of energy consumption at part-load operation, as shown in Figure 3.11.

As stated earlier, the refrigerant in LiBr CHW absorption chillers is water, and hence does not pose any environmental problems. Because LiBr is essentially a nontoxic salt

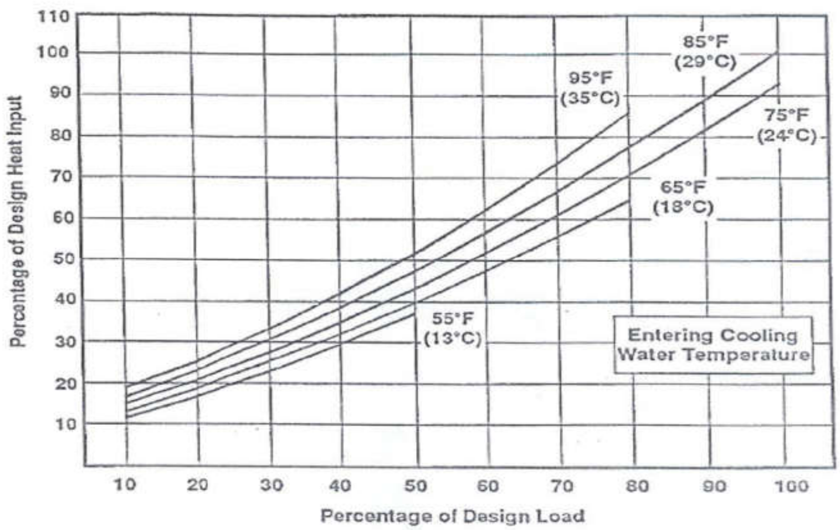



**Figure 3.11** Absorption chiller part-load capacity chart, single-effect.

solution, its environment impact is negligible. Corrosion inhibitors are typically used in conjunction with the LiBr to retard the natural corrosion between the LiBr and internal metal surfaces. These inhibitors can be toxic and must therefore be handled carefully and disposed of in accordance with environmental laws.

The demand for absorption chillers is increasing as these machines are used throughout the world where waste heat is available from power generation or other processes. Experience with these types of machines is that they require constant attendance by factory trained and qualified technicians or else they become prone to failure, such as crystallization.

Absorption chillers have parasitic loads such as solution pumps, purge pumps, and refrigerant pumps that must be considered if an energy analysis is being performed. These parasitic loads are dependent on chiller size but are typically 0.28% of chiller output for chillers above 1000 tons (3500 kW); 0.43% between 500 and 900 tons (1750–3200 kW) and 0.71% for chillers under 300 tons (1050 kW). As an example, for a 300 ton chiller the ancillary electric load would be approximately 300 × 0.71% = 2.1 tons or 7.5 kW, for a 1250 ton chiller the ancillary electric load would be approximately 1250 × 0.28% = 3.5 tons or 12.3 kW. For chiller capacity in kW, a 2000 kW capacity chiller would have a parasitic load of 2000 × 0.43% = 8.6 kW. The designer should consult with the chiller manufacturer for information pertaining to a specific chiller selection.

## CHILLER CONFIGURATION

### Selecting Chiller Quantity and Size

The size and number of chillers in a plant are impacted by several parameters, including but not limited to the type of chiller (screw, centrifugal, absorption, etc.), commercially available chiller capacities, plant-maximum load, plant-minimum load, and plant-construction phasing. Table 3.3 provides a summary of chiller characteristics for consideration in establishing plant chiller configuration.

System load estimates and calculations can be analyzed on an annual basis to assist in selecting the actual size of the chillers to use. Several factors affect the selection of the size and quantity of chillers for a DCP, including available space, energy efficiency, redundancy requirements, and available budget.

While the following discussion is specific to a single-building plant, the same logic and process is scalable for larger district plants. The example we will use is a hospital located in Miami, Florida. Through load analysis software, the peak load has been calculated to be 500 tons. The designer may select one chiller at 100% of the peak load; two chillers at 50% of the peak load for equal sized chillers, or some other percentage for unequal sized chillers. Because most buildings and DCSs operate close to their peak cooling load only a few hours a year, having fewer chillers has fewer stages of chilled-water production to meet the annual load profile.

Chillers must be selected to satisfy both peak and minimum cooling loads and all partial loads in between and operate efficiently at all conditions. Operating close to peak capacity typically equates to peak operating efficiency. Hence, a single chiller would have to operate over the full load range from 0% to 100%. While selecting a single chiller for 100% of the cooling load saves space and cost, it may not be able to turndown effectively to meet the minimum cooling loads or operate efficiently, necessitating multiple chillers to meet the load demands.

Despite chiller manufacturer claims of operating down to 10% of peak load, a centrifugal chiller typically can only turn down to 20% of its peak capacity and operate stably, depending on entering water conditions. Furthermore, centrifugal chillers are not recommended to operate below 40% of their design load unless provided with variable-speed drives; otherwise, chillers will operate at low efficiencies and have higher specific energy consumption (i.e., higher kW/ton). In variable-flow systems, the turndown is impacted by the water velocity within the evaporator tubes, which is typically between 3 to 12 fps (0.9 to 3.66 m/s) with the low end starting the laminar flow condition. Typically, chillers are

**Table 3.3** Summary Table of Chiller Characteristics

| Chiller Type | Typical Efficiency | Size Range, ton | Equipment Cost, $/ton | Maintenance Cost*, Estimated Annual Maintenance Costs |
|---|---|---|---|---|
| Electric Centrifugal (Standard Single Compressor) | 0.61–0.7 kW/ton (COP 4.7 to 5.4) | 500 to +1500 | 200 to 275 | $3,700 |
| Electric Centrifugal (Standard Dual Compressor) | 0.61–0.7 kW/ton (COP 4.7 to 5.4) | 1500 to +4,000 | 250 to 350 | $4,400–$5,000 |
| Electric Centrifugal (Single Compressor Industrial—Field Erected) | 0.61–0.7 kW/ton (COP 4.7 to 5.4) | 2,500 to +5,500 | 650 to 800 | $4,800–$5,500 |
| Engine-Driven Centrifugal | (COP 1.5–1.9) | 100 to +3000 | 450 to 650 | $4,400+ engine maintenance |
| Steam-Driven Centrifugal | — | 100 to +4000 | — | $5,500+ depending on size |
| Direct-Fired Absorption Chiller (Double Effect) | (COP 0.85–1.20) | <100 to >3250 | 400 to 2000 | $4,800–$5,500 |
| HW Absorption Chiller (Single Effect) | (COP 0.55–0.70) | <60 to >3250 | 450 to 1000 | $4,800–$5,500 |
| Steam Absorption Chiller (Single Effect) | (COP 0.60–0.75) | <60 to >3250 | 450 to 800 | $4,800–$5,500 |

** Maintenance costs courtesy of Johnson Controls, Inc./York International (Grunewald 2013). Typical annual activities include changing oil filter, oil filter analysis, and motor checks. Costs do not include cleaning tubes, eddy current testing, or complete oil or refrigerant replacement. For approximate pricing:*

1. *Cleaning evaporator or condenser tubes as required use $1,200.*
2. *Eddy current testing for under 500-ton chiller use $1,700 and $2,500 for larger chillers.*
3. *Complete oil replacement contact vendor's service department.*
4. *For all absorbers a Bromide test is conducted twice per year. Costs do not include chemicals.*

selected in the 6 to 7 fps (1.8 to 2.1 m/s) range, therefore, the flow can only turn down to about 50% or so of design flow. Selecting chillers with higher tube velocities (and therefore pressure drop) results in higher turndown capabilities. The design should consider the trade-offs of higher pressure drop and greater turndown if the plant will operate many hours below 50% capacity of one chiller.

While minimal load concerns are not a common issue experienced in the Middle East or other regions of the world with similar climates because of the size of the DCPs and the quantity of chillers, an annual load duration curve can be of use in other locations in selecting unequal sized chillers to match all load conditions.

Also, chillers operating at low part load and high condensing temperatures are more prone to surge. Surge is when the refrigerant flows backwards through the compressor momentarily until pressure can be built up adequately to flow the refrigerant forward again. This typically occurs at part load when the lift conditions are still high. While most chiller control panels have software to prevent this condition, surge can still occur and ultimately cause damage to the compressor shaft thrust bearings. Therefore, knowing the load profile and load duration curve can eliminate this by sizing a chiller accordingly to its expected operating range.

A method of falsely loading the compressor to operate stably at lower lift conditions is to implement hot gas bypass. The process takes hot gases from the compressor discharge and reintroduces it into the refrigerant cycle to artificially load the compressor. Hot gas bypass is discouraged by ASHRAE Standard 90.1 (ASHRAE 2016c). Absorption chillers on the other hand have a more linear relationship with capacity and input energy and may reduce down to 10%–15% of peak load as long as the entering water conditions are within design parameters.

Returning to the hospital example, Figure 3.12 represents the annual cooling load duration curve for the sample prototype hospital. The load duration curve sorts the hourly cooling load from highest to lowest to determine the number of hours that occur above or below at a specific load. Load duration curves can be an important tool to use in determining the size and quantity of chillers based on hours of operating at more efficient operating range. Selecting a chiller at 33% of peak load results in a size that will operate over 7100 hours per year or about 81% of the time. This keeps the chiller operating closer to its peak design efficiency for more hours during the year as well as being able to turndown lower better to meet the minimum load of 62 tons.

It should be noted that 7100 hours of operation per year is typically on the extreme side for a building located in the United States, which typically sees less than 4000 full-load hours per year; however, that is dependent on where the chilled-water plants are located in the world and the type of buildings being serviced.

Once the load profile is determined, an energy analysis can be conducted comparing different chiller sizes as summarized in the following example.

The example proves that unequal size chillers work for this specific example and should be considered in all plant configurations to meet all load conditions. Furthermore, when selecting unequal sizes of chillers in a plant, more care should be directed toward a pumping scheme where pumps on the primary side will have different flows and heads (due to different chiller characteristics) and consequently differences in pump and system curves. These pressure and flow differences can be overcome either by

- selecting the chillers for equal pressure drops,
- using individual primary pumps, or

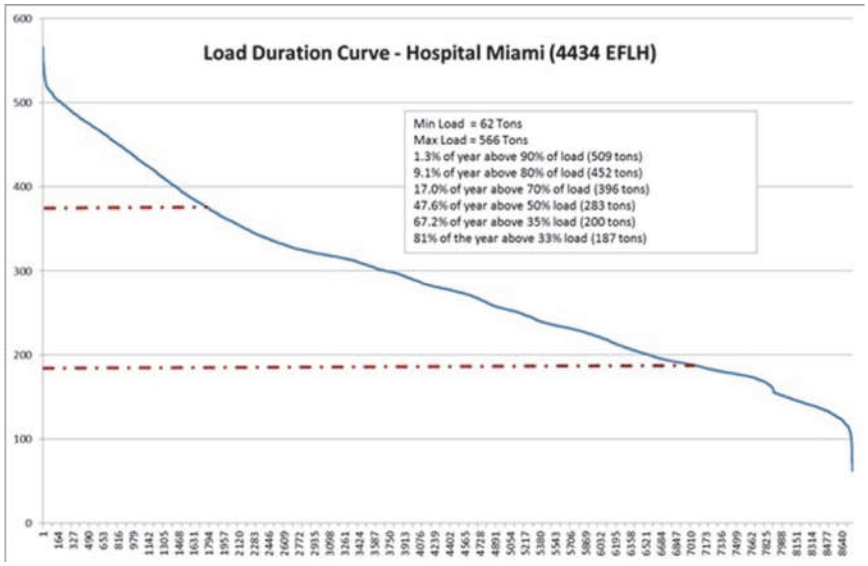



**Figure 3.12** Miami, Florida hospital cooling load duration curve.

**Example 3.1:** Using the previously mentioned prototype hospital, and assuming an average electrical consumption cost of $0.1022/kWh and a monthly peak electric demand cost of $12.5/kW, calculate the annual energy costs of two plant configurations: two chillers at 50% each (283 tons) and one chiller at 66% (374 tons) and the other at 33% (187 tons) to determine the most energy efficient configuration.

**Solution:** Using a chiller plant analyzer software tool, the three different plant configurations were modeled for energy usage and cost. All chillers were selected to exceed ASHRAE 90.1-2013 energy efficiency requirements at 0.60 kW/ton. The comparison is summarized in Table 3.4, which shows a 185,481 kWh and $18,906 in annual savings from the equal sized chiller option. These annual savings should be compared on a life-cycle basis to determine if the installation actually would payback in a reasonable time frame.

**Table 3.4** Comparison of Chiller Energy Usage and Cost

| Option | Chiller Quantity and Capacity | Annualized Chiller Performance, kW/ton | Annual Electrical Use, kWh | Total Plant Annual, Electrical Cost |
|---|---|---|---|---|
| 1 | Two chillers at 283 tons | 0.470 | 1,191,993 | $150,865 |
| 2 | One chiller at 187 tons and one chiller at 374 tons | 0.393 | 1,006,512 | $131,959 |

- actively modulating valves using a flow or pressure signal at each chiller to equalize the pressure drop through each chiller.

## Level of Redundancy Required

It is typically recommended to have one additional unit of production of each chiller type and size to act in a standby capacity in case one unit of production fails. This is known as redundancy, and in this particular case, is notated as $N$+1 with $N$ being the largest chiller size. Furthermore, the size of a chiller is dictated by the peak and part-load profile of the plant, so if the selected unit's size cannot turn down adequately to meet the part-load conditions, either variable-speed-driven (VS) chillers or even smaller sized chillers are selected to meet this operating condition.

The topic of redundancy should be a well thought out, cognizant decision by the district cooling provider on how they will address unforeseen equipment outages. Equipment will eventually fail, and typically this occurs when it is the most inconvenient time during peak demands. Because of the large diversity of customer loads experienced in large district plants, redundant or standby units may not be provided. Many times, a plant is not fully built out but grows with the load demands over several years. In that case, there typically is a redundant unit above the estimated peak demand to compensate for any loss of chilled-water production until the plant is fully subscribed on capacity. While a redundant chiller may not be present, there are typically redundant pumps or cooling towers to increase the level of reliability of the plant. Redundancy and reliability are key features of district energy plants; therefore, maintaining redundant units is preferred, but not always implemented. If the nature of the load makes thermal energy storage (TES) feasible it can provide a degree of redundancy. See Chapter 6 for more information on TES system sizing and design.

District energy plants are typically staffed with full-time, highly qualified, trained staff 24 hours per day. Furthermore, they implement proper preventive maintenance practices, minimizing the risk of lost capacity and downtime because of this heightened level of reliability. Refer to Chapter 8 for additional information on maintenance practices.

Similarly, the nominal oversizing of units of production, such as cooling tower cells, will assist in overcoming any operation issues resulting from a tower failure or a tower undergoing maintenance. Under normal operating conditions, this oversizing improves chiller efficiency by providing lower entering condenser-water temperatures.

## CHILLER STAGING

Staging constant-speed (CS) chillers (i.e., no VFDs) in a chiller plant that uses supply temperature reset, works best if the leaving-water temperature is reset as low as possible to ensure chillers are fully loaded before starting the next chiller. This will reduce operating chiller efficiency for that period of time, but the total plant energy remains less than starting another chiller and its auxiliaries. Chillers must be staged off conservatively to avoid staging back on too quickly, causing excessive wear and tear on chillers and starters, so the manufacturer typically adds an anti-recycle timer in the control sequence to limit the number of starts and stops in a certain time frame.

For plants with variable-speed (VS) chillers, the most efficient plant operation has to be analyzed based on chiller make and model necessary to meet a certain load criterion since it might be more energy efficient to operate three chillers at low load and low speed (33%) than one chiller at full load (100%). This is true because most VS chillers operate more efficiently at part loads above 30%, compared to a CS chiller at 100% loading.

Properly sequencing chillers in a DCP plays a vital role in reducing energy costs and improving plant COP. The most efficient chiller sequence takes advantage of the type of chiller employed in a plant and the way chillers unload or modulate to meet the load. The sequence of operating single-stage chillers varies from those with multistage or modulating type compressors. Sequencing will also vary when a hybrid plant uses both absorption and centrifugal chillers.

Single-stage chillers are sequenced on and off depending on the cooling load, which is usually in response to return-water temperature or flow in the secondary (distribution loop) compared to the primary (chiller loop) side, depending on the pumping configuration. Because the majority of DCPs have identically sized chillers, it does not matter which one operates first or second, except for maintaining the same equivalent operating hours; however, if a plant consists of different types or sizes, there are advantages to starting the smaller and more efficient chillers first. Where multistage chillers are used, unloading takes place by disabling a chiller stage.

In applications where chiller types are different, such as centrifugal and screw chillers, the chillers should be switched off at the minimum acceptable load condition that the chiller manufacturer recommends. Where thermal storage is applied, it is recommended to operate chillers at their best efficiency point and divert the excess load to storage to optimize their operation and efficiency of the plant. This is more of a trickle charge than a well-defined charge mode. When several centrifugal/screw chillers in a plant are operated, it is normally advisable to start with the smaller size chillers first. If the chillers are the same size, always attempt to have all chillers share the load and operate with the same loading percentage instead of different loadings. It is also a good idea to load each chiller as close as possible to the best efficiency point; hence the control system should use the compressor map of each chiller to determine what that operating condition is.

Hybrid plants with both absorption and vapor-compression chillers should have the absorption chiller(s) upstream of the vapor-compression chiller in order to optimize the absorber's efficiency and provide stable operating conditions. When sequencing chillers, care should be taken to enable and operate the absorption chillers in advance of demand, due to the lengthy delay before they reach capacity. For plants with CS chillers, only an adequate number of chillers to serve the load should be energized. Running more chillers than is required to meet the load usu-ally means the chillers are not operating at their best efficiency point (70%–80% loaded) and are wasting energy since each chiller in operation means energizing its auxiliary equipment: primary pumps, condenser pumps, cooling tower fans, etc. This is a symptom of low $\Delta T$ syndrome, which should be addressed at the cooling load location (i.e., cooling coil and control valve).

## CHILLER ARRANGEMENTS AND PUMPING CONFIGURATIONS

A DCS consists of three main components: the production (plant), the transport (distribution), and the utilization (consumer interconnection or ETS). Each of these components consumes energy, and designers should design each at its point of maximum efficiency and cost-effectiveness. However, it is not possible to optimize the design of one component without consideration of the other two components and achieve optimal overall performance for the system. Depending on the selected configuration, when these components are combined, the total energy consumed and cost-effectiveness of the sum of all three components might not necessarily be the best possible. To achieve the optimal result, the components must be arranged together properly and operated in a manner that takes advantages of that arrangement. In some instances, the cooling production is selected at its best efficiency and the distribution pumping is selected at its best efficiency point, but when combined the total system efficiency is less than optimal. This section will highlight the different chiller arrangements and pumping schemes that impact the overall plant and system efficiency.

### Chiller Arrangements

With respect to the chilled-water systems, chillers may be arranged either in parallel or in-series as shown in Figures 3.13 to 3.21. The chillers can be either centrifugal, absorption, or a combination of both.

## Circulating Fundamentals

There are essential fundamentals for the circuiting of chillers that should be followed to maintain the design intent and system maximum efficiency. Some of these are:

- Circuit piping so that energy consumption of chillers is not increased.
- Arrange piping so that all chillers receive the same return water temperature.
- Size piping so that part-load operation will not lead to flow increase beyond the flow range acceptable to the chiller when some number of pumps in the system have been deenergized and consequently the friction loss reduced through the system.
- Ensure that the minimum required flow through the evaporators is achieved.

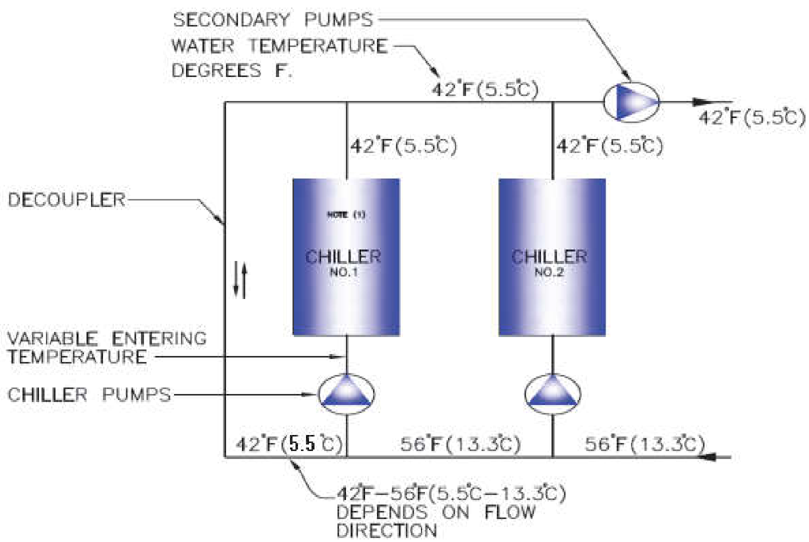



**Figure 3.13** Decoupler location that effects loading.

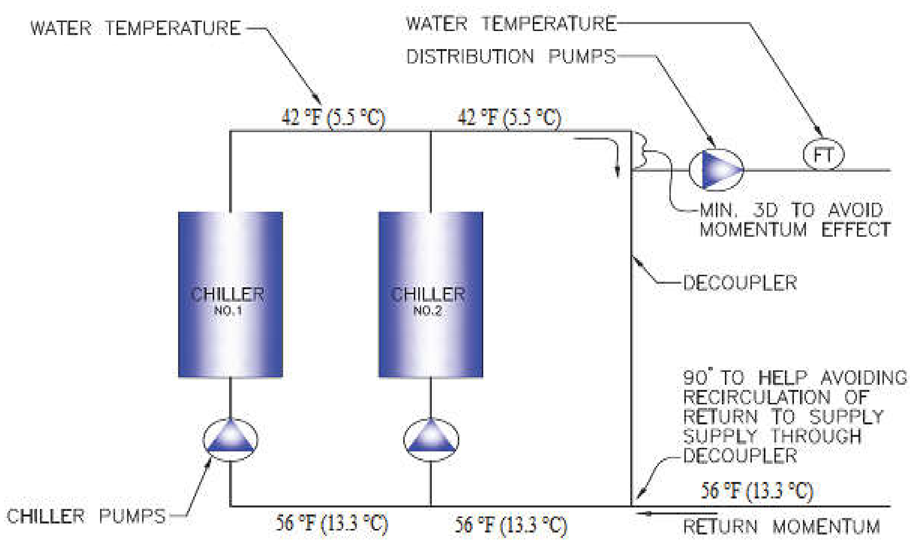



**Figure 3.14** Traditional hydraulic decoupler location.

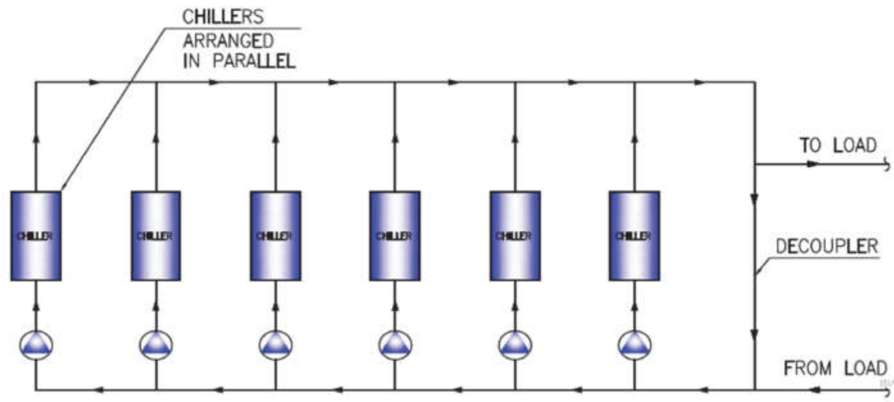



**Figure 3.15** Parallel chillers with individual pumping for each chiller.

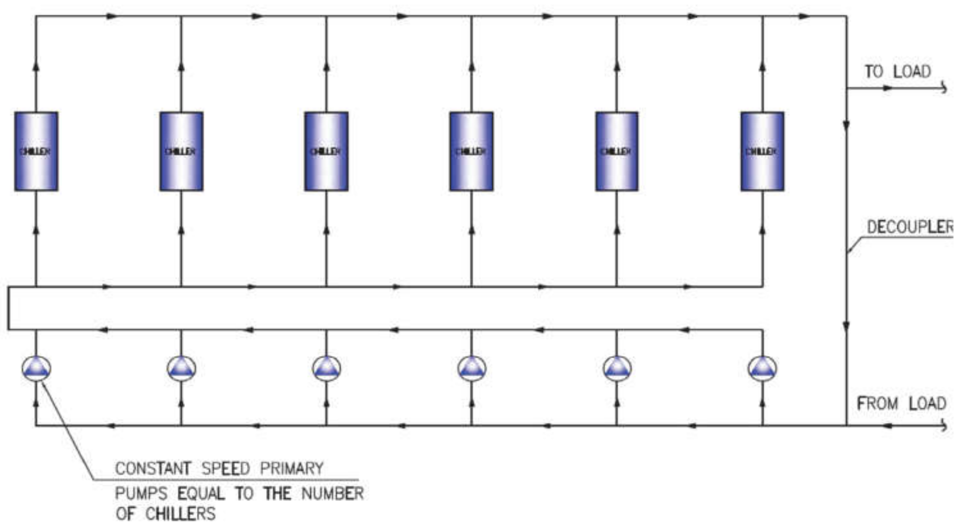



**Figure 3.16** Parallel chiller arrangement with headered constant-speed pumping.

When secondary pump flow is less than primary flow, chiller-entering water temperature will vary according to flow variation and will not be constant. As indicated in Figure 3.13, colder primary supply water will blend with the warmer return water on chiller number one to create a lower mixed entering temperature that will result in a different entering condition than chiller number two, thereby unequally loading the two chillers even though the flow volume will be the same since each chiller is individually pumped since chiller number one is "preferentially loaded."

### Parallel Chillers Arrangement

The parallel chiller arrangement has all chillers arranged in-parallel and pumps may be individual per chiller or use a headered or manifold arrangement.

The advantages of the parallel-chiller arrangement over the series-chiller arrangement are:

- In dedicated pumping, failure of one pump has less impact on plant total capacity compared to the in-series module arrangement.

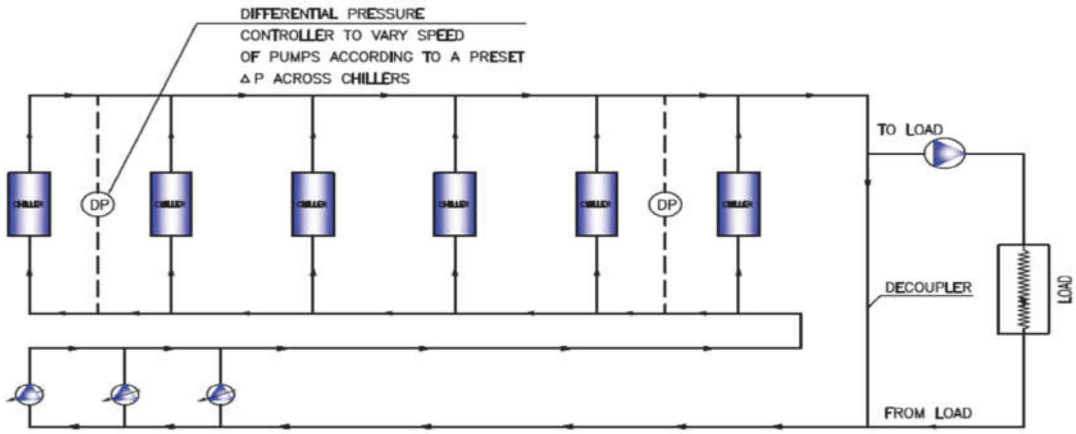



**Figure 3.17** Parallel chiller arrangement with headered variable-speed primary pumping.

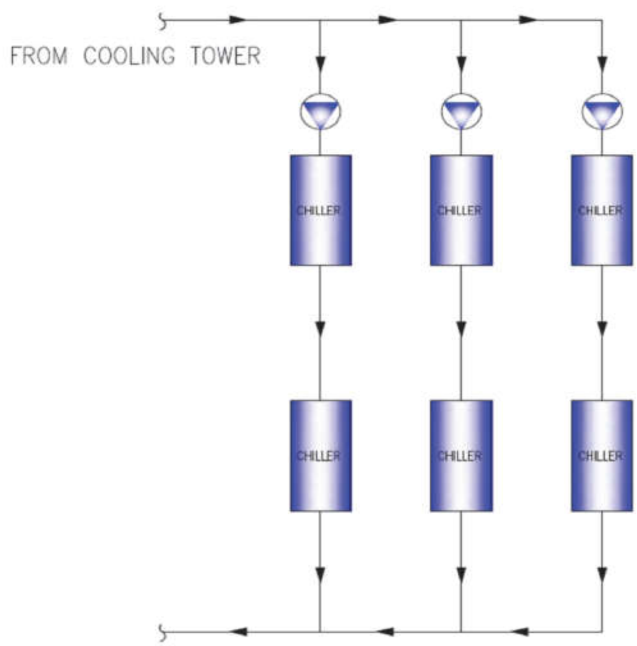



**Figure 3.18** Condenser-water flow for in-series chillers with individual pumping per module.

- Chiller piping and accessories are smaller and easier to handle during construction.
- Where primary/secondary pumping is used, the circulated water through the decoupler is smaller compared to the series arrangement case, thus better plant efficiency may be achieved.

### Series Chillers Arrangement

To improve chiller performance, chillers may be arranged in-series such that two chillers form a single module. The CHW return will enter the upstream chiller in the module to be cooled prior to entering the downstream chiller. The condenser water will start with the downstream chiller then continue to the upstream chiller for series counter-

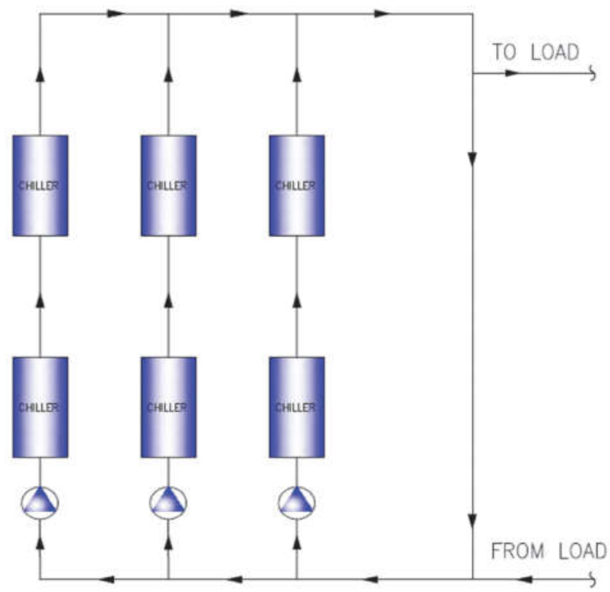



**Figure 3.19** Chilled-water flow with in-series chillers with individual pumping per chiller module.

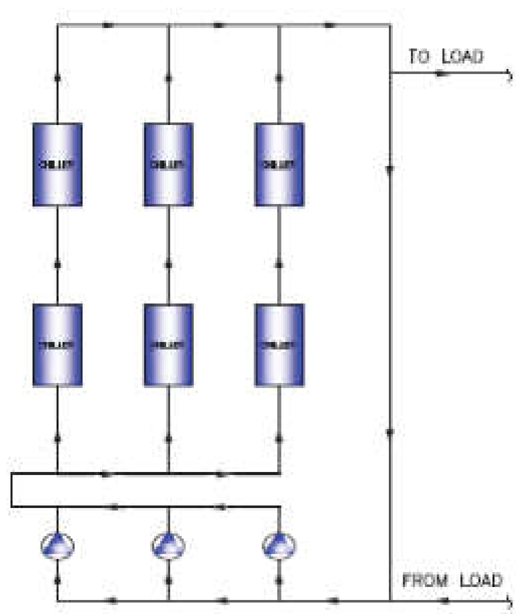



**Figure 3.20** Series chillers with headered variable-speed pumping.

flow configuration or alternatively each chiller will have its own condenser circuit (refer to Figures 3.18 to 3.20).

The advantage of the series arrangement over the parallel arrangement is the increased overall chiller-module efficiency, especially where large CHW $\Delta T$s (greater than 16°F or 8.9°C) are used because the compressor lift is split between two units. Typically, the temperature differential and capacity are not an even 50/50 split but are selected so that the compressor lift for each chiller is similar.

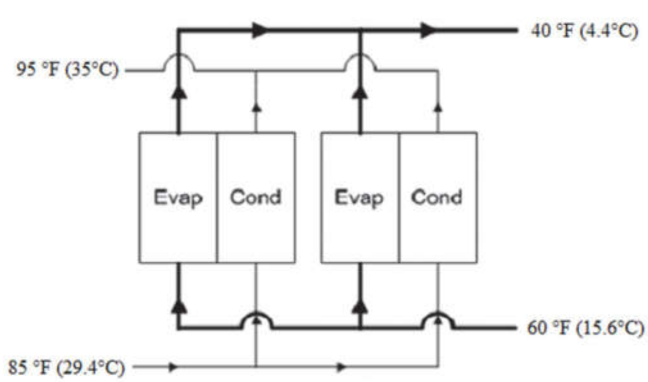



**Figure 3.21** For parallel flow, each chiller operates at maximum system lift.

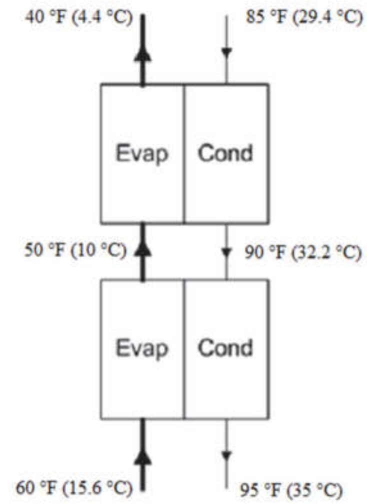



**Figure 3.22** With series-counterflow arrangement, each chiller has lower lift than when in a parallel arrangement.

The disadvantages of series chilled-water flow are that other plant components such as primary pumps and condenser pumps might have higher power consumption since the combined chiller pressure requirements at the combined series chillers flow must be considered. Thus, total plant energy consumption, taking into consideration the chiller-loading percentages according to load profile and the energy waste due to pumping energy, must be evaluated for any efficiency gains. Due to higher pump head in the in-series chiller arrangement, chillers should be a single pass type to mitigate the higher pump head.

Figures 3.21 and 3.22 show the difference in ideal chiller performance due to arrangement, assuming a close approach between saturated refrigerant temperature in the evaporator and condenser. In examining Figures 3.21 and 3.22, note the following:

- For the parallel flow arrangement, refrigeration temperature lift is
    - 95°F – 40°F = 55°F (35°C – 4.4°C = 30.6°C)
- For the series-counterflow arrangement the lift
    - Upstream chiller: 95°F – 50°F = 45°F (35°C – 10°C = 25°C)
    - Downstream chiller: 90°F – 40°F = 50°F (32.2°C – 4.4°C = 27.8°C)

Tables 3.5a through 3.7b illustrate the difference in efficiencies between parallel and series-counterflow chiller arrangements for low-pressure and medium-pressure refrigerants for a 6000 ton chiller “module.”

Thus, in the series-counterflow arrangement, the chillers will be subjected to lower average lift and thus can achieve higher performance. Chiller manufacturers normally claim that series-counterflow arrangement will reduce chiller energy consumption by approximately 5%, and the information in Tables 3.5 through 3.7, indicates a savings over 8%.

Series chillers can handle large CHW $\Delta T$s more efficiently than parallel chillers. Series chillers are more suited to the variable primary pumping concept since there is no flow through the decoupler other than at low-flow conditions, and there is no excess primary flow diluting the temperature entering the chillers. It is recommended that series

**Table 3.5a** 6000-Ton Chiller Efficiency Comparison Table by Configuration using HFO-1233zd(E) at 50 Hz (I-P)[1]

| | Parallel (Evap/Cond) | | | Series-Counterflow (Evap/Cond) | | |
|---|---|---|---|---|---|---|
| Chiller | Capacity, Tons | Temperature Evaporator Condenser, °F | Peak Efficiency, kW/ton | Capacity, Tons | Temperature Evaporator Condenser, °F | Peak Efficiency, kW/ton |
| CH-1 (Downstream) | **3000** | 40–58/93–103 | 0.6099 | 2820 | 40–48.4/93–98 | 0.5821 |
| CH-2 (Upstream) | **3000** | 40–58/93–103 | 0.6099 | 3180 | 48.4–58/98–103 | 0.5459 |
| **Average** | **3000** | **40–58/93–103** | **0.6099** | **3000** | **40–58/93–103** | **0.5629** |

[1] *Chiller selections using HCFC-123 were not available in this size range, therefore, only HFO-1233zd(E) selections are indicated.*

**Table 3.5b** 6000-Ton Chiller Efficiency Comparison Table by Configuration using HFO-1233zd(E) at 50 Hz (SI)[1]

| | Parallel Evaporator / Parallel Condenser | | | Series-Counterflow (Evap/Cond) | | |
|---|---|---|---|---|---|---|
| Chiller | Capacity, kW | Temperature Evaporator Condenser, °C | Peak COP | Capacity, kW | Temperature Evaporator Condenser, °C | Peak COP |
| CH-1 (Downstream) | 10551 | 4.4–14.4/33.9–39.4 | 5.77 | 9918 | 4.4–9.1/ 33.9-36.7 | 6.04 |
| CH-2 (Upstream) | 10551 | 4.4–14.4/33.9–39.4 | 5.77 | 11184 | 9.1–14.4/ 36.7–39.4 | 6.44 |
| **Average** | **10551** | **4.4–14.4/33.9–39.4** | **5.77** | **10551** | **4.4–14.4/ 33.9–39.4** | **6.25** |

[1] *Chiller selections using HCFC-123 were not available in this size range, therefore, only HFO-1233zd(E) selections are indicated.*

**Table 3.6a** 6000-Ton Chiller Efficiency Comparison Table by Configuration using HFC-134a at 50 Hz (I-P)

| | Parallel (Evap/Cond) | | | Series-Counterflow (Evap/Cond) | | |
|---|---|---|---|---|---|---|
| Chiller | Capacity, Tons | Temperature Evaporator Condenser, °F | Peak Efficiency, kW/ton | Capacity, Tons | Temperature Evaporator Condenser, °F | Peak Efficiency, kW/ton |
| CH-1 (Downstream) | **3000** | 40–58/93–103 | 0.6562 | 2640 | 40–47.9/93–97.4 | 0.6051 |
| CH-2 (Upstream) | **3000** | 40–58/93–103 | 0.6562 | 3360 | 48.4–58/97.4–103 | 0.5769 |
| **Average** | **3000** | **40–58/93–103** | **0.6562** | **3000** | **40–58/93–103** | **0.5927** |

**Table 3.6b** 6000-Ton Chiller Efficiency Comparison Table by Configuration using HFC-134a at 50 Hz (SI)

| | Parallel (Evap/Cond) | | | Series-Counterflow (Evap/Cond) | | |
|---|---|---|---|---|---|---|
| Chiller | Capacity, (Tons) | Temperature Evaporator Condenser, °C | Peak COP | Capacity, Tons | Temperature Evaporator Condenser, °C | Peak COP |
| CH-1 (Downstream) | 10551 | 4.4–14.4/33.9–39.4 | 5.36 | 9285 | 4.4–8.8/33.9–36.3 | 5.81 |
| CH-2 (Upstream) | 10551 | 4.4–14.4/33.9–39.4 | 5.36 | 11817 | 8.8–14.4/36.3–39.4 | 6.1 |
| **Average** | **10551** | **4.4–14.4/33.9–39.4** | **5.36** | **10551** | **4.4–14.4/33.9–39.4** | **5.93** |

**Table 3.7a** 6000-Ton Chiller Efficiency Comparison Table by Configuration using HFO-513A at 50 Hz (I-P)

| | Parallel (Evap/Cond) | | | Series-Counterflow (Evap/Cond) | | |
|---|---|---|---|---|---|---|
| **Chiller** | **Capacity, Tons** | **Temperature Evaporator Condenser, °F** | **Peak Efficiency, kW/ton** | **Capacity, Tons** | **Temperature Evaporator Condenser, °F** | **Peak Efficiency, kW/ton** |
| CH-1 (Downstream) | **3000** | 40–58/93–103 | 0.6725 | 2640 | 40–47.9/93–97.4 | 0.6192 |
| CH-2 (Upstream) | **3000** | 40–58/93–103 | 0.6725 | 3360 | 47.9–58/97.4–103 | 0.6015 |
| **Average** | **3000** | **40–58/93–103** | **0.6725** | **3000** | **40–58/93–103** | **0.6093** |

**Table 3.7b** 6000-Ton Chiller Efficiency Comparison Table by Configuration using HFO-513A at 50 Hz (SI)

| | Parallel Evaporator/Parallel Condenser | | | Series-Counterflow | | |
|---|---|---|---|---|---|---|
| **Chiller** | **Capacity, kW** | **Temperature Evaporator Condenser, °C** | **Peak COP** | **Capacity, kW** | **Temperature Evaporator Condenser, °C** | **Peak COP** |
| CH-1 (Downstream) | 10551 | 4.4-14.4/33.9-39.4 | 5.23 | 9285 | 4.4-8.8/33.9-36.3 | 5.68 |
| CH-2 (Upstream) | 10551 | 4.4-14.4/33.9-39.4 | 5.23 | 11817 | 8.8-14.4/36.3-39.4 | 5.85 |
| **Average** | **10551** | **4.4-14.4/33.9-39.4** | **5.23** | **10551** | **4.4-14.4/33.9-39.4** | **5.77** |

chillers be applied for plants with capacities exceeding 15,000 tons (52,750 kW) to reduce the impact of high-circulated water flow through the decoupler when primary and secondary flows are mismatched and also to reduce the impact of losing a dedicated pump on the plant total capacity.

## Absorption Plus Centrifugal Chillers

Where absorption chillers are used with centrifugal chillers to form a series-chiller arrangement, the absorption chillers should be on the upstream side while the centrifugal chillers are installed on the downstream. This piggyback module arrangement offers the following advantages:

- The centrifugal chillers will be able to reduce CHW supply below 42°F (5.6°C), which is typically about the lower limit for absorption chillers.
- Absorption chillers operating at higher entering and leaving-water temperatures increases the chiller COP.
- The absorption chiller will handle part-load return temperature with no energy penalty as the chiller performance is linear with return temperature, unlike centrifugal chillers that lose efficiency and exhibit higher specific energy consumption with lower return temperature.
- Absorption-chiller evaporator flow can typically be reduced by 10% of peak flow.
- When the centrifugal chillers flow is not matched with the absorption chiller flow due to size restrictions or when a variable primary pumping concept is used, the arrangement of the centrifugal-absorption chillers may be separated as shown in Figure 3.23.

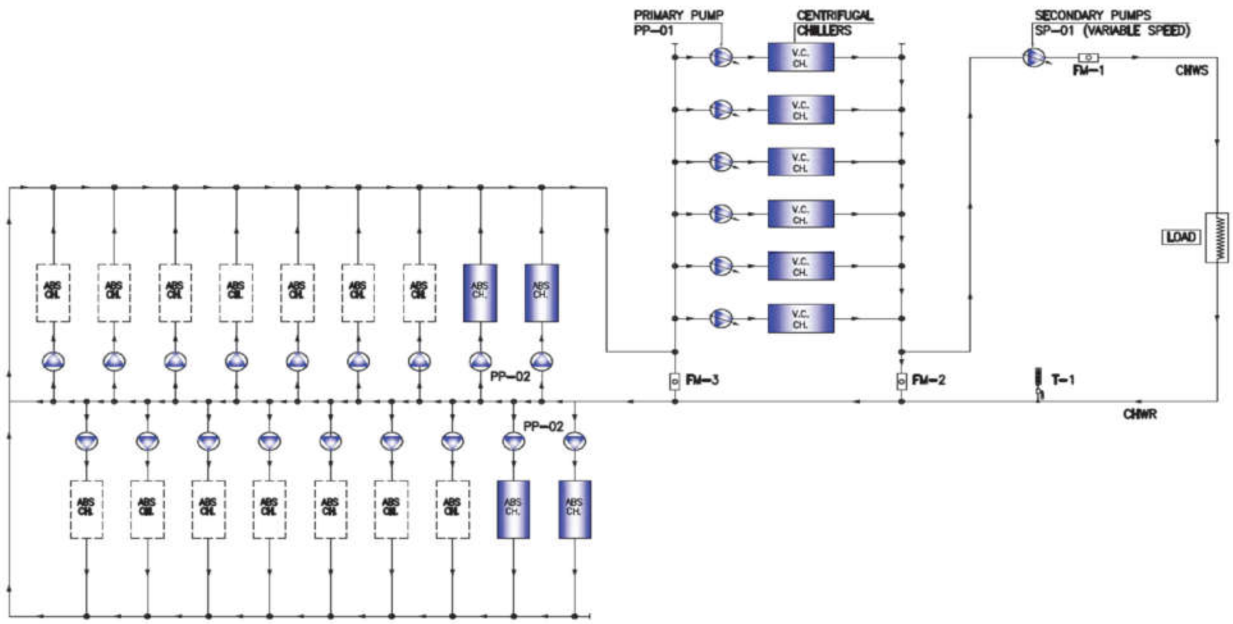



**Figure 3.23** Arrangement of absorption and centrifugal chillers in series or piggyback configuration.

# PUMPING SCHEMES

## Plant Pumping

Understanding different pumping schemes is vital in the design of DCPs. The pumps and their installation and operation on district cooling CHW systems are of utmost importance in achieving efficiency and cost-effectiveness.

The designer must recognize that the pumping arrangement and the type of pumps used will positively or negatively impact the chiller performance, and they should install pumps in a way that will not adversely affect the chiller energy consumption and impair efficiency. All else being equal, a reduction in water flow through the chiller evaporator will lead to capacity loss and a flow increase will potentially lead to increased CHW supply temperature.

Also, a reduction in water flow through the condenser will lead to chiller efficiency reduction in addition to capacity reduction, while an increase of condenser-water flow will reduce chiller-specific energy consumption up to the point of cooling tower capacity. For series chillers, the pumping energy will increase as flow is increased due to higher pressure drops through the chiller.

The designer should consider the impact to the overall chiller plant energy performance from various chilled- and condenser-water temperature differences. Table 3.8 summarizes the impact of two different chilled-water and two different condenser-water temperature ranges on the plant power consumption by looking at each component's power consumption. As the table shows, options with the widest temperature differentials (lowest leaving chilled water and the highest leaving condenser water) offered the best plant efficiency gains even though these options used more chiller energy. Designers should be cautioned to conduct their own detailed analyses because a great number of assumptions were used in developing Table 3.8 regarding temperatures, pipe sizes, and pressure drops, which may not correspond to your specific system.

**Table 3.8** Impact on Plant Efficiency by varying Chilled and Condenser Water Temperatures for a 6000-Ton Chiller Module using HFO-1233zd(E) (Weiman 2017)

| Config. | Chilled-Water Temp, °F (°C) | Cond.-Water Temp, °F (°C) | Chiller Power, kW | Chilled-Water Pump Power, kW | Cond.-Water Pump Power, kW | Cooling Tower Power, kW | Total Plant Power, kW | Plant, kW/ton | Plant COP | Total Power Savings, % |
|---|---|---|---|---|---|---|---|---|---|---|
| Parallel | 40–58 (4.4–14.4) | 93–103 (33.9–39.4) | 14638 | 2412.7 | 2581.8 | 847.5 | 20480.0 | 0.8533 | 4.12 | — |
| Parallel | 40–58 (4.4–14.4) | 91–105 (32.8–40.6) | 14854 | 2412.7 | 1092.1 | 847.5 | 19206.3 | 0.8003 | 4.39 | 6.2 |
| Parallel | 38–60 (3.3–15.6) | 93–103 (33.9–39.4) | 15031 | 1324.3 | 2581.8 | 847.5 | 19784.6 | 0.8244 | 4.27 | 3.4 |
| Parallel | 38–60 (3.3–15.6) | 91–105 (32.8–40.6) | 15252 | 1324.3 | 1092.1 | 847.5 | 18515.9 | 0.7715 | 4.56 | 9.6 |
| Series Counterflow | 40–58 (4.4–14.4) | 93–103 (33.9–39.4) | 13512 | 2882.4 | 3456.2 | 847.5 | 20698.1 | 0.8624 | 4.08 | (1.1) |
| Series Counterflow | 40–58 (4.4–14.4) | 91–105 (32.8–40.6) | 13556 | 2882.4 | 1465.5 | 847.5 | 18751.4 | 0.7813 | 4.5 | 8.4 |
| Series Counterflow | 38–60 (3.3–15.6) | 93–103 (33.9–39.4) | 13788 | 1592.1 | 3456.2 | 847.5 | 19683.8 | 0.7354 | 4.29 | 3.9 |
| Series Counterflow | 38–60 (3.3–15.6) | 91–105 (32.8–40.6) | 13744 | 1592.1 | 1465.5 | 847.5 | 17649.1 | 0.8631 | 4.78 | 13.8 |

## Pressure Gradient in CHW Distribution Systems

Pressure gradient diagrams are an excellent tool for checking pump energy use in a DCS. Figure 3.24 shows a simple flow diagram for a sample distribution system at full load. The static pressure, or fill pressure, of the system is shown in Figure 3.25 as point A; in this case it is 40 ft (12 m). The pump head in this figure is 160 ft (48.5 m) of $H_2O$. The pressure drop through the network is 120 ft (36.4 m) total, while the pressure drop across the critical consumer's cooling coil, control valve, and piping is 20 ft (6 m) in total. Note that typical buildings at different locations in the network will have differing pressure drops as shown in the pressure gradient diagrams of Figures 3.24 and 3.25.

## Part-Load Condition

At part load, the friction loss in the distribution network is considerably reduced due to less flow in the distribution system and consequently lower velocity and pressure losses. Figure 3.26 illustrates what the pressure gradient diagram will look like at part load.

It should be noted that for constant-volume pumping and variable-flow pumping, until the $\Delta P$ sensor ramps down the pump speed when the system is at part load, the pressure across the consumer interconnection will be increased. While the cooling-coil pressure drop and static-balancing valve pressure loss will be reduced due to reduced flow rate across them, the control valve pressure drop will be increased. This may result in having the control valve subjected to a pressure drop beyond its control range, and thus may lead to system malfunction. See Chapter 5 for a discussion on methods to deal with this issue.

If the pump is not variable volume, the pump operating point will ride the pump curve and lead to pump discharge pressure reaching close to its shutoff pressure (Figures 3.27 and 3.28).

It is advisable for such systems to provide some sort of a pressure control valve to maintain constant differential pressure across the zone control valve. Where VFDs are used to reduce pump speed based on network differential pressure, $\Delta P$ control valves are

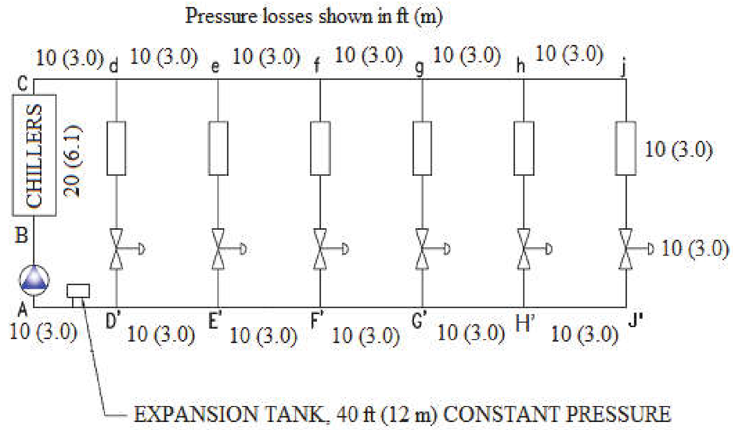



**Figure 3.24** Distribution system diagram.

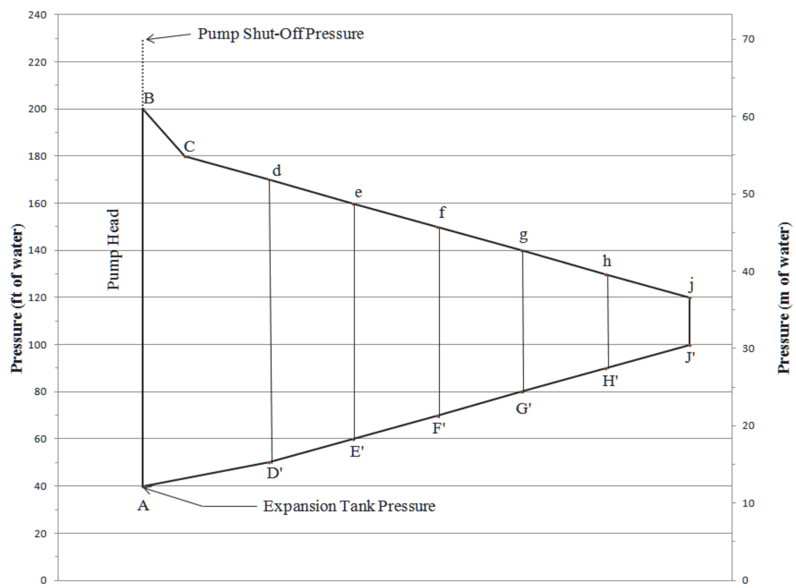



**Figure 3.25** Pressure gradient diagram for a distribution system at full load.

still required to account for $\Delta P$ variations in ETS rooms. The temperature control valve on the district CHW return line can also be used to perform this duty to accommodate the pressure fluctuations, but this requires an industrial style control valve (refer to Chapter 5 for more information).

Alternatively, one may design the network on a reverse-return concept. Figures 3.29 and 3.30 describe the utilization of this system to mitigate the high $\Delta P$ impact on the control valve at part load.

The reverse return in a network offers almost equal and constant differential pressure across ETS rooms at peak and part loads. However, this concept requires additional pip-

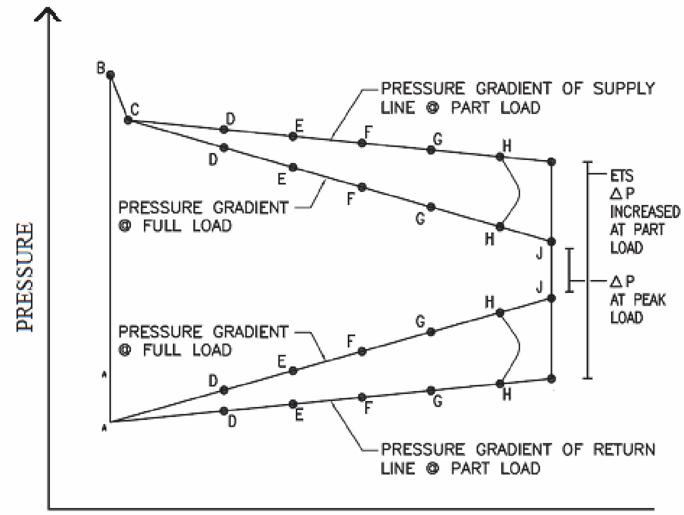



**Figure 3.26** Direct return pressure gradient at part-load operating. (Note: pump is not riding the curve.)

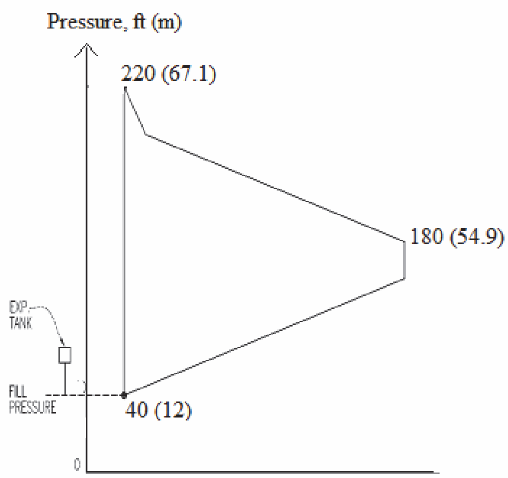



**Figure 3.27** New pressure gradient with pump riding the curve-constant-flow system.

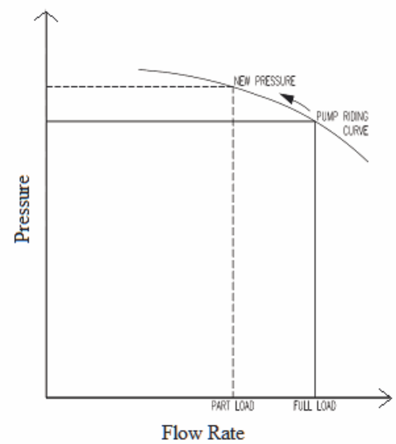



**Figure 3.28** Pump riding the curve at part load.

ing in most instances, although it may be readily adopted when the network forms a loop and is fed from a single plant; Figure 3.31 illustrates such an arrangement.

## Distribution Network Pumping-System Configurations

The distribution system circulates chilled water from the plant to the end user's heat exchangers and then returns the water back to plant. To achieve the objectives of the CHW distribution system, the following pumping schemes are used:

- CS primary
- VS primary
- Primary-secondary
- Primary-secondary-tertiary
- Primary-distributed secondary

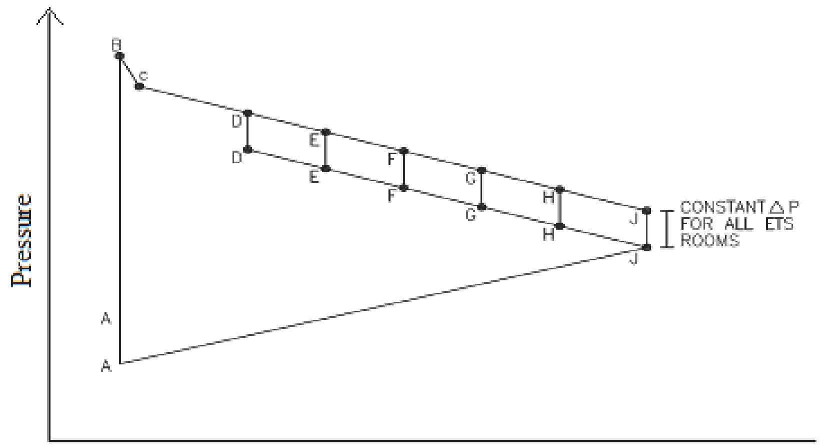



**Figure 3.29** Pressure gradient at full load in a reverse-return system.

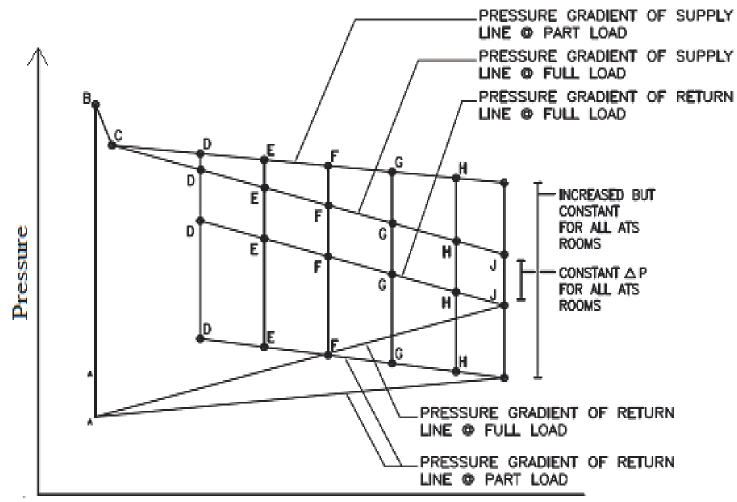



**Figure 3.30** Pressure gradient at part load in a reverse-return system.
(Note: pump is not riding the curve.)

The pumping arrangement and pressure gradient diagram of each system are presented in Figures 3.32 to 3.36.

The layout and pressure gradient figures at peak and part loads (Figures 3.32 to 3.36), illustrate the advantages and parameters that should be taken into consideration while designing a network. The pressure gradient demonstrates that the primary-distributed secondary consumes less pumping energy than other pumping configurations. It also indicates that the pressure within the plant is reduced, the secondary pump surge is avoided, and the network pressure is reduced as ETS room pressure is dealt with by ETS pumps. However, there are costs (installation and energy use) associated with having the pumps in each ETS room. In distributed secondary pumping, the speed of the in-building secondary pumps will vary to maintain the required differential pressure across the building heat exchange, thus maintaining the control valve authority at its peak during partial load conditions. This system requires attention to the pressure loss in the network at the peak condition and is suitable when the system peak loads are well defined from the start of the project.

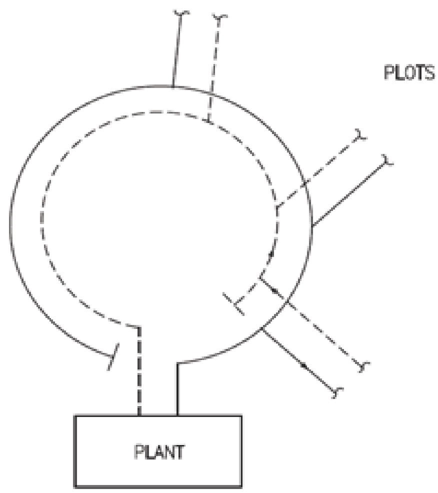



**Figure 3.31** Reverse return implemented in a loop-distribution system.

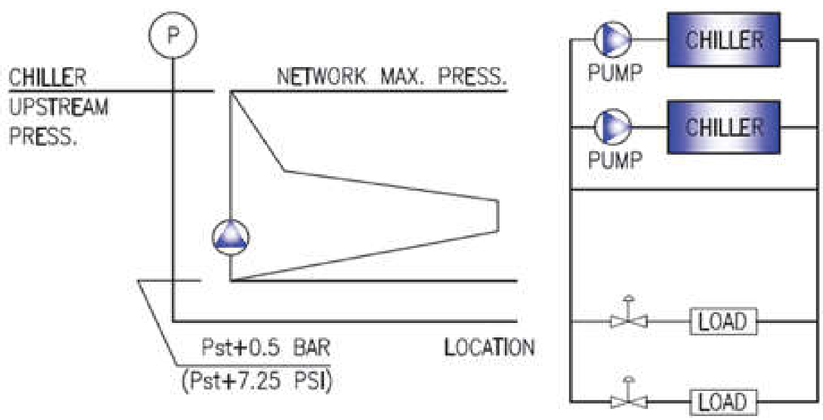



**Figure 3.32** Primary pumping system.

The friction loss through the network, and accordingly the CHW velocity, should be maintained at low values (i.e., larger piping) to avoid the need to pressurize the expansion tank at high levels to maintain the required net-positive suction head (NPSH) for the pumps, particularly those at the end of the network. One of the features of this system is the high pressure of the network return line compared to the supply. Precautions should be taken to avoid the situation where a user exceeds the return pressure at that location and thus impacts other users. Typically, the distribution system pipe sizes in this system are oversized compared to other systems to keep the distribution system's pressure drops down. The primary-distributed secondary is very attractive when the development loads are well known and when the network length is not so extensive. The disadvantage is, if a building is located somewhere within the distribution system where it was not planned, its pumps will negatively impact all the other existing pumps in the system since their pressure drops are affected. To overcome this, distributed pumps might be oversized or operated at a higher frequency than required prior to the addition of the new load and pump.

The pressure gradient of the primary-secondary or the variable primary highlights the need to have a differential pressure sensor at each building's ETS. This is to mitigate the

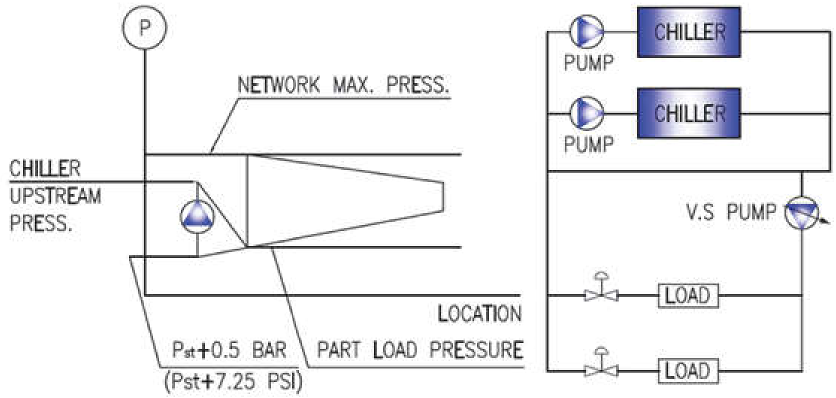



**Figure 3.33** Primary-secondary pumping system.

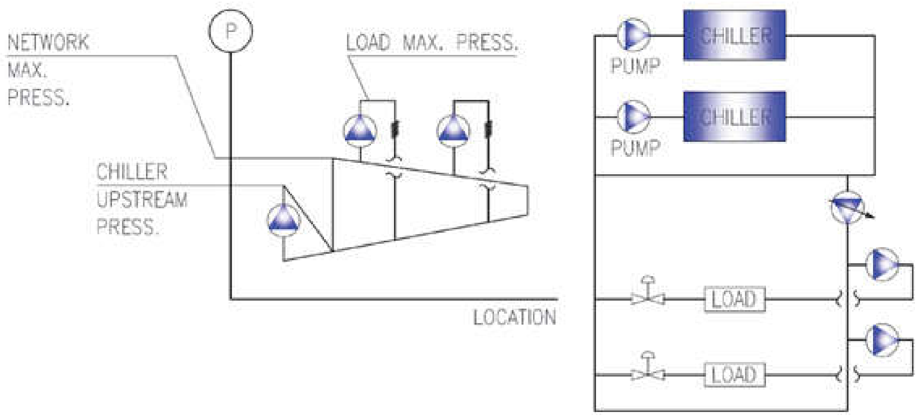



**Figure 3.34** Primary-secondary-tertiary pumping system.

impact of the higher differential pressure on the ETS control valves close to the plant at peak load and also those at the network end during part load conditions. The primary-secondary system could be used where network length is long and there is a possibility of pump cavitation in the case of primary distributed secondary approach. Also, this system has a flexibility of changing building loads from planned ones without affecting the systems in other buildings. Where supply pressure becomes high due to lengthy network, it may be desirable to introduce a cascaded pumping concept in the network.

Depending on system size, primary-secondary pumping systems may also provide a further benefit of keeping all distribution (secondary) pumps in the low voltage range and not the medium voltage range. Low-voltage pumps save installation costs and the VFDs are physically smaller. For primary-secondary pumping with constant-speed primary pumps, there always must be more flow in the primary loop than the secondary loop, otherwise the warmer return water will blend with the cooler supply water, increasing the chilled-water supply temperature. Having constant-speed primary pumps wastes energy in two ways: 1) the loops are not balanced and must have more primary flow than secondary flow, therefore wasting primary pump energy, and 2) in order to have more primary loop flow, additional equipment may be required to be energized (chiller, primary chilled-water pump, condenser water pump, and cooling tower) just to meet the flow require-

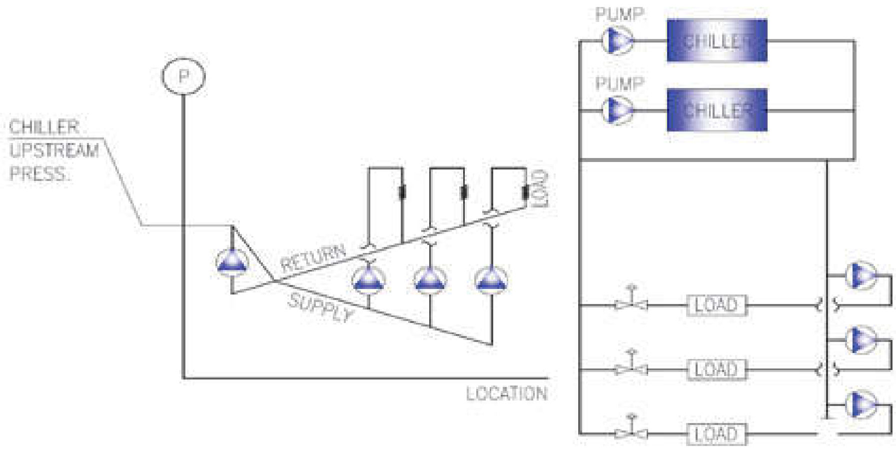



**Figure 3.35** Primary with distributed secondary pumping system.

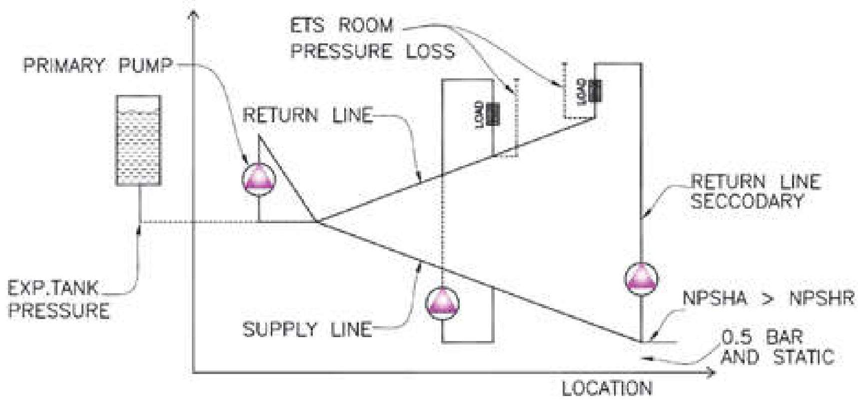



**Figure 3.36** Primary with distributed secondary pumping.

ments. Varying the flow of the primary loop so it exceeds secondary loop flow by a nominal amount reduces the overall plant energy consumed. Both variable primary flow pumping and variable primary-variable secondary pumping achieves this goal.

Variable primary pumping will require a bypass valve around the chillers that will only open when the system flow is lower than the chiller minimum flow. For most district systems in the Middle East, this valve only opens upon the start of the second or third chiller in sequence since the loads may not reduce to a level below the minimum flow of one chiller.

Variable primary-variable secondary pumping uses a traditional decoupler to handle loop flow imbalances. Both loops must have flow meters. Only a nominal amount of additional primary flow is required to maintain leaving chilled-water temperature. Typically, the secondary loop pump speed is controlled to satisfy remote differential pressure sensors and the primary loop pumps ramp up and down to match the secondary pump flow plus a small additional amount of incremental flow. The actual value of the incremental flow is contingent upon the accuracy of the flowmeters and the quality of the controls and instrumentation. The higher the quality of instrumentation, the smaller the incremental flow can be.

Regardless of the pumping configuration, the chilled water or condenser water flow rate of change through a variable-flow chiller should be confirmed with the chiller manufacturer. Typically, this value is below 30% of design flow per minute.

## CHW Primary Pumping Configuration

CHW pumping for the primary circuit within plants may be classified as follows:

- Individual CS pumps directly connected to the chillers (Figure 3.37)
- Individual CS pumps connected to a common system header (Figure 3.38)
- VS pumps connected to a common system header (Figure 3.39)
- VS pumps directly connected to the chillers (Similar to Figure 3.37)

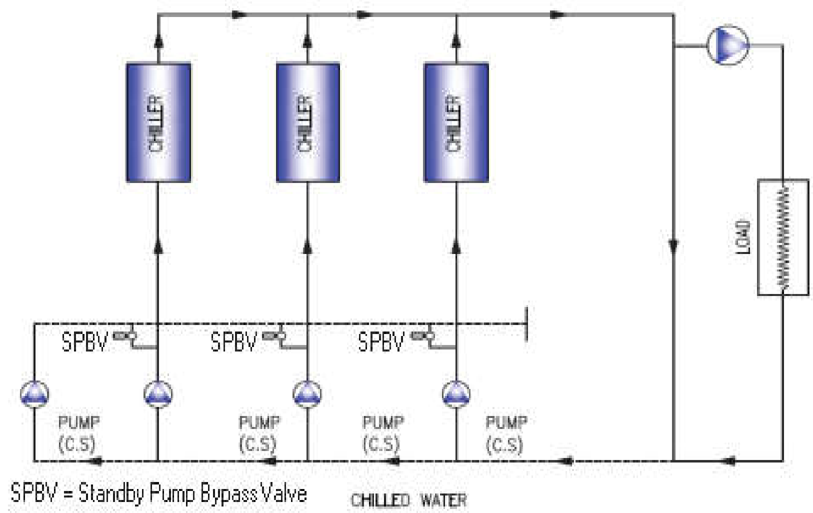



**Figure 3.37** Primary-secondary pumping with individual CS pumping directly connected to each chiller.

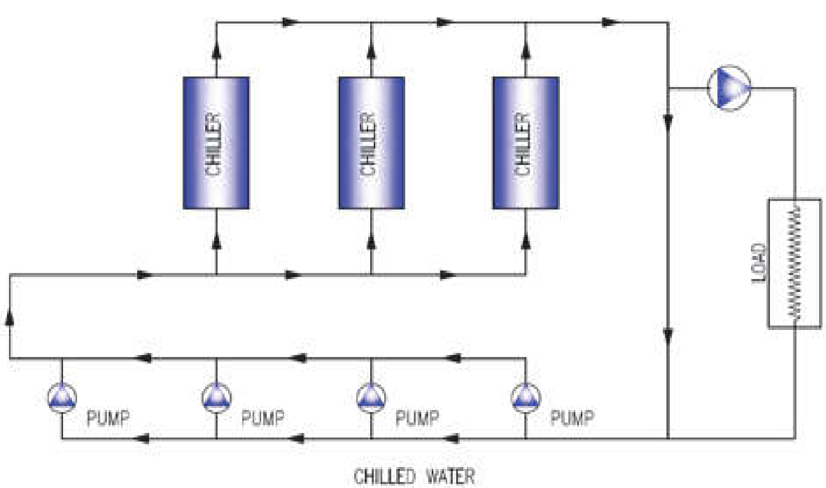



**Figure 3.38** Individual module CS pumping and a common header.

The following are common features of CS or VS individual pumps directly connected to chillers:

**CS Pumps**

*Advantages*

- Have no associated energy losses or increased maintenance due to the use of a VFD
- Water flow will increase at part load due to less friction in the header, thus improving chiller-specific performance

*Disadvantages*

- Energy consumption is only reduced by pump riding its curve as pressure differential reduces

**VS Pumps**

*Advantages*

- Saves energy by reducing flow at part load
- Reduces pump start in-rush current

*Disadvantages*

- VFD is not 100% efficient, therefore there are drive losses and heat is rejected into the room
- Drive cabinet takes up more space

The following are common features of CS or VS primary pumps connected to a common header:

*Advantages*

- Any pump may serve as a standby pump for the others
- No need for interconnecting bypass header for standby pump(s)
- Lesser number of pumps and associated accessories

*Disadvantages*

- Failure of a pump will affect a chilling module unless a costly standby pump with a bypass arrangement, including additional header and valving, is installed
- Pumps should be adjacent to chillers to avoid extensive runs of piping
- Less flexibility if chillers have different evaporator pressure drops

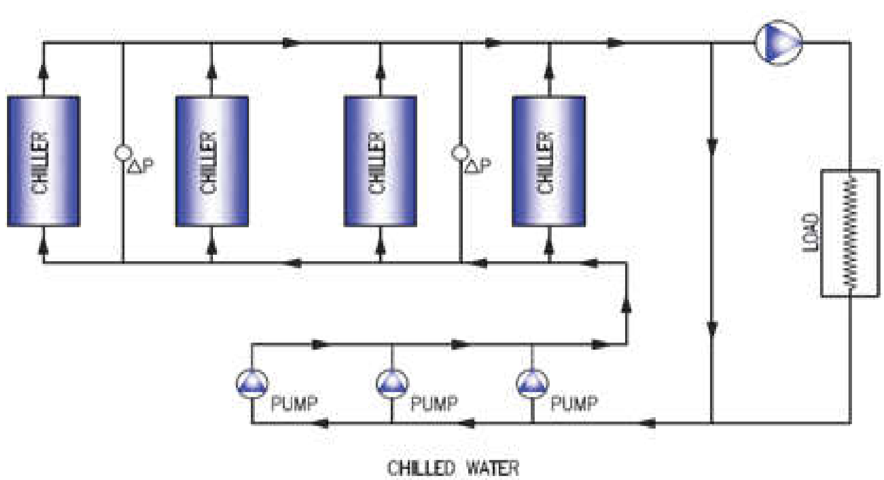



**Figure 3.39** VS pumps connected to common system header.

Regardless of the pumping configuration, having variable flow in both the distribution loop and chiller loop (either variable primary flow or variable primary-variable secondary) will ensure the chillers see the highest entering return water temperatures possible since there is minimal over pumping of the primary loop. This feature is a critical component in optimizing the chiller and system performance and should be the basis of design for all large district cooling plants.

## Plant Condenser Pumping Arrangement

Similar to chilled-water pumping configurations, condenser-water pumping within the plant may be classified as:

- Individual CS pumps directly connected to chillers (Figure 3.40)
- Individual VS pumps directly connected to chillers (Similar to Figure 3.40)
- Individual CS pumps connected to a common header system (Figure 3.41)
- VS pumps connected to a common header system (Figure 3.42)

The common features of CS individual pumps directly connected to chillers are:

- No VFD and associated energy loss and maintenance
- Pump starting current can be managed within acceptable limits

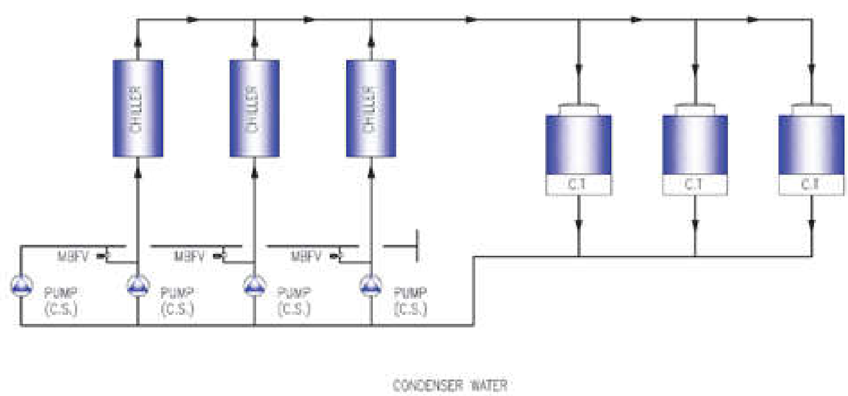



**Figure 3.40** Individual module CS condenser-water pumping. Pump is directly connected to the chiller.

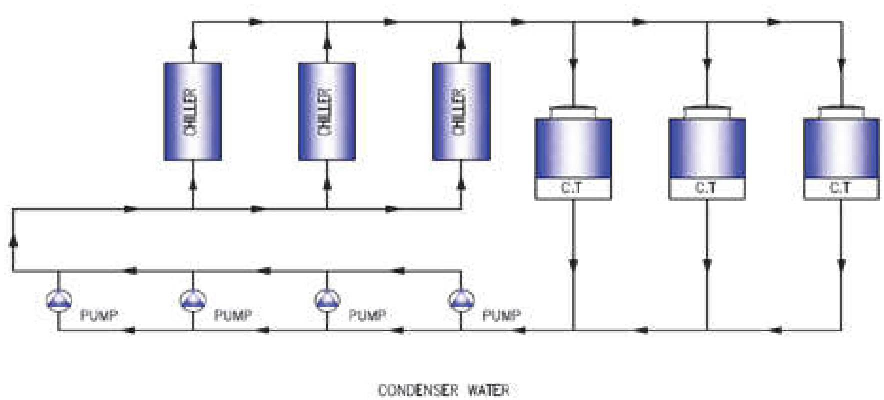



**Figure 3.41** Individual module CS pumping with pumps on a common header.

- Condenser-water flow will increase at part load due to less friction in the header, thus improving chiller performance
- Should a standby pump(s) be required, another header has to be provided and furnished with valves that will increase the cost and add congestion to plant piping.

The common features of VS individual pumps directly connected to chillers are:

- Opportunity for increased energy savings at part load conditions
- Pump starting current can be managed within acceptable limits

The common features of CS individual pumps connected to chillers via a common header are:

- Absence of VFD and associated energy loss and maintenance
- Pump starting current can be managed within acceptable limits
- Condenser-water flow will increase at part-load due to less friction in header, thus improving chiller performance
- Standby pump(s) can be any of the pumps
- No need for extra header for standby pump(s)

The features of VS individual pumps connected to chillers via a common header are:

- Constant flow through chillers regardless of varying load conditions; this is recommended when absorption type chillers are in use
- Lower number of pumps and relevant accessories
- Possible increase of condenser flow by varying frequency to improve chiller performance

## Condenser-Water Piping and Pumping for Unequal Numbers of Chillers and Cooling Towers

When the number of cooling tower modules are unequal to the number of chiller modules, a special pumping arrangement may be needed. For such systems the goals are:

- Maintain the design flow through the chiller condenser
- Maintain the cooling tower desired flow rates

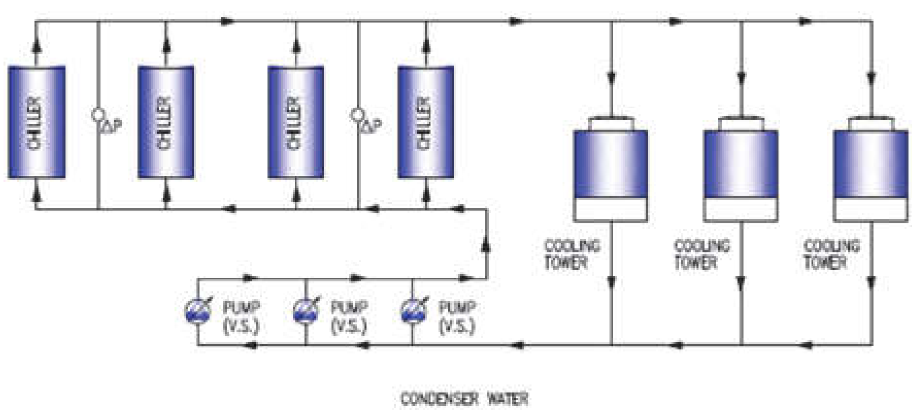



**Figure 3.42** VS pumps connected to common system header.

The flow to each chiller condenser can be controlled or trimmed via a monitoring signal from either a differential pressure sensor or through a flowmeter across the chiller cooling water headers (preferred method). A control valve or VFD on the pump can be modulated to vary the flow based on input from the monitoring signal in order to equalize pressure drops or flows. The $\Delta P$ sensor is normally used when the chillers are of the same model, capacity, and the chillers are headered. A flowmeter or pressure differential transmitter in conjunction with a modulating valve (Figure 3.43) can be used when pressure drop through chillers are different to dynamically balance the water flow through different chillers.

## Pumps

District plants require large flow rates and therefore large pumps. The pump head varies from low head for cooling systems (condensers) and primary CHW circuits to large head pumps for distribution. With such large flow-rate pumps, centrifugal pumps are the most suitable.

Based mounted horizontal split case are commonly preferred for most applications due to improved component access for maintenance purposes compared to vertical in-line pumps. Typically, higher pump efficiencies are achievable (above 80%) due to flow and head conditions.

These large pumps are typically double-suction design to minimize the end thrust due to water entering the impeller. Designers should select the pump such that the pump curve intersects with the system curve at all possible applicable design conditions. Larger pumps correspond to larger and heavier motors. A 400 HP (298.3 kW) motor, which is an average sized distribution pump, weighs over 3000 pounds (1364 kg), which is very cumbersome to remove and replace, especially if it is a vertical inline pump. Hence, designers should be cognizant of the weight of the motor and maintain a clear path and means of removal, including monorails and A-frame hoists, and consider base-mounted pumps as the design preference.

# HEAT REJECTION

CHW systems absorb heat from the end user, the distribution system, pumps/flow friction, and from chiller inefficiencies and must ultimately reject this heat to a sink. The heat sink could be the ambient air, sea, river, lake, or the groundwater. It should be noted that ground coupling (as is done with a ground-coupled or geothermal heat pump system) using a central plant chiller does not produce the same building energy performance as a unitary building system connected to a ground loop. The energy efficiency

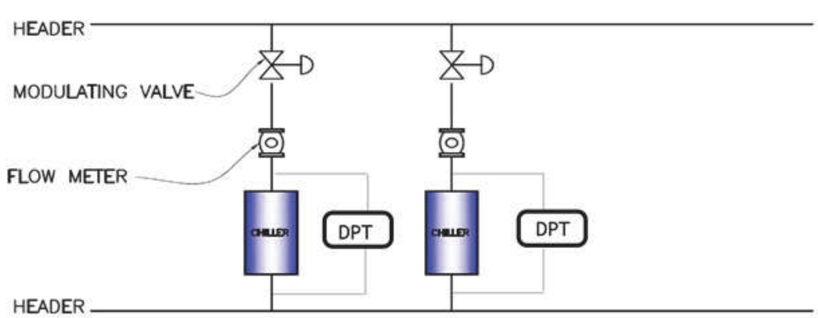



**Figure 3.43** Flowmeter and modulating valve to vary flow.

of a ground-coupled or geothermal heat pump systems is as much a function of the type of equipment used within the building as it is to the use of the ground as a heat source/sink. Ground-coupled systems also require a balance between heat rejection and heat extraction to achieve their best performance and economics. While the undisturbed ground temperature might look very attractive as a heat sink, as heat is rejected to the ground its temperature will rise due to the limitations imposed by heat transfer to the surrounding earth. Geothermal DCSs require careful design by those who fully understand and appreciate the limitations of heat transfer with the ground.

Many times the ambient air is used as the heat sink. Ambient air is used to cool the compressed hot gas either through a direct connection between hot gas tubes and ambient dry bulb and using an air-cooled condenser or water-cooled using evaporative cooling techniques that wets the tubes or fill material in evaporative condensers or cooling towers. Evaporative cooling efficiency is linked to the ambient wet bulb where air-cooled equipment is associated with the ambient dry-bulb. Another use of water-cooled chillers is to circulate condenser water from the chiller condenser to an air-cooled radiator. While the chiller may be more efficient by being water-cooled compared to air-cooled, the radiator's performance is related to ambient dry-bulb temperature and not wet-bulb temperature and this process will not be as efficient. However, it is used in many locations where there is a lack of makeup water for an evaporative cooling process. Since the ambient wet-bulb temperature is lower than the dry-bulb temperature, the former will lead to lower compressor lift and consequently less compression energy.

### Heat Rejection Equipment

Heat rejection equipment used in comfort cooling systems includes air-cooled condensers, open-cooling towers, closed-circuit cooling towers, and evaporative condensers. The energy consumption of heat rejection devices will impact overall cooling efficiency and thus these devices should be controlled to offer the greatest overall cooling performance.

Per ASHRAE Standard 90.1, each fan speed control powered by a motor larger than 7.5 hp (5.6 kW) should have the capability to operate that fan at two-thirds of full speed or less and shall have controls that automatically change the fan speed to control the leaving fluid temperature or condensing temperature pressure of the heat rejection device. A cooling tower, according to Standard 90.1, should be capable of turning down to 50% water flow. Standard 90.1 also dictates the minimum condenser water flow per cooling tower fan horsepower to establish component efficiencies.

Exceptions to Standard 90.1 are:

- Condenser fans serving multiple refrigerant circuits
- Condenser fans serving flooded condensers
- Installations located in climate zones 1 and 2

## CONDENSER WATER

Water is the media for removing heat from the condensers in water-cooled chillers. This condenser heat is from the heat of the building cooling loads transferred through the CHW distribution system, heat gains in the distribution system, and pump work, in addition compression heat in the case of electric or engine-driven vapor-compression chillers or the generator heat in the case of absorption chillers. Water is circulated through the chiller condenser, picking up the heat and then transferring the heat to cooling towers (open) or evaporative condensers (closed-circuit fluid coolers) or other heat sink (GeoExchange well, lake, river, ocean, etc.), where heat is rejected to the atmosphere or groundwater source.

Sediment will build up in the cooling tower basin since airborne particulate will be drawn into the tower intake and washed down into the basin. The airborne debris can also stick to the wetted fill, and the fill can become fouled and reduce heat transfer. This sediment that either is entrained into the condenser water or that settles in the basin normally contains dissolved solids that contain sulfates and chlorides. As the water is evaporated at the cooling towers or evaporative condensers, the concentration of the total dissolved solids (TDS) in the circulated water increases. High levels of TDS will coat and scale chiller condenser tubes, cooling tower fill and negatively affects chiller and cooling tower heat transfer efficiency; therefore, the TDS level must be controlled to always remain within the acceptable limits as dictated by the chiller and cooling tower manufacturers. A TDS level of 2000 ppm is normally acceptable to manufacturers, but there may be other parameters that dictate lower levels, such as the case of using treated sewage effluent, hence, the chiller and heat rejection manufacturers should be consulted as to the water quality limitations.

To avoid the TDS increasing beyond acceptable levels, part of the circulated water is drained off (i.e., blown down) and fresh or treated makeup water is added. Makeup water can be fresh water, treated sewage effluent (gray water), ground water (lakes and rivers) or salt water. Each water source will have its own specific design material and equipment requirements.

The amount of blowdown and makeup water to replace the drained and evaporated water depends on the evaporation rate of the cooling tower and the makeup water quality (TDS level). Generally speaking, makeup water rate $M$ equals evaporation $E$ plus blowdown $B$ plus drift $D$. All these components ($E$, $B$, and $D$) can be calculated individually and estimated from the following equation (SPX Cooling Technologies 2009):

$$B = \frac{E - [(C-1) \times D]}{(C-1)} \tag{3.4}$$

where:

$B$ = Rate of blowdown, gpm ($m^3/h$)
$E$ = Rate of evaporation, gpm ($m^3/h$)
$C$ = Cycles of Concentration (COC) or concentration ratio,
Final TDS (ppm)/Initial TDS (ppm)
$D$ = Rate of drift loss, gpm ($m^3/h$)

A common rule of thumb for rate of evaporation losses $E$ is 3 gpm per 100 tons (0.54 l/s per 1000 kW) of refrigeration. If a more precise rate of evaporation is desired, it can be estimated using the following equation (SPX Cooling Technologies 2009):

$$E(\text{gpm}) = 0.0008 \times \text{circulating flow rate (gpm)} \times \text{tower temperature range (°F)}$$

or for metric units:

$$E(\text{m}^3/\text{h}) = 0.0014 \times \text{circulating flow rate (m}^3/\text{h)} \times \text{tower temperature range (°C)}$$

Cycles of concentration (COC or $C$) or concentration ratio is more difficult to determine. COC refers to the ratio of impurities or TDS in the circulating water to the TDS in makeup water. Chemical treatment and tower vendors traditionally utilize calcium carbonate ($CaCO_3$) as the surrogate for TDS or conductivity measured in parts per million (ppm). Tower vendors also set maximum levels of water quality depending on materials of construction as shown in Table 3.9.

Operating at higher COC reduces makeup water requirements. Typical COC for fresh water is in the range of 3 to 7; sea water is 1 to 1.5; once through systems is 1.0 and treated sewage effluent is 2 up to 9 (this entirely dependent on the materials of construction of the cooling tower and chiller condenser water heat exchanger [tube bundle]). Typical hardness concentrations in cooling tower basins are between 50 and 250 ppm.

Assuming fresh water is used for makeup water with a TDS of 20 ppm and a basin TDS of 120 ppm, then the COC would be 120/20 or 6 cycles. Local chemical treatment companies can be consulted as to the normal ranges seen in the project region.

Drift refers to small droplets of water that are entrained in the tower discharge air. Drift losses are typically minimal compared to evaporation losses and are obtained from the cooling tower vendor. If the drift loss rate is not known, it may be estimated from (SPX Cooling Technologies 2009), or a typical value of 0.02% or less of circulated flow can be used:

$$D(\text{gpm}) = 0.0002 \times \text{circulating flow rate (gpm)}$$

or for metric units:

$$D(\text{m}^3/\text{h}) = 0.0002 \times \text{circulating flow rate (m}^3/\text{h)}$$

## COOLING TOWERS

Cooling towers are evaporative cooling devices that spray water across an airstream and are characterized by several design features. Cooling towers can be classified by the nature of the contact between the air and water—direct or indirect contact. Direct contact refers to a typical open cooling tower where the water to be cooled is in direct contact with the ambient air. Indirect contact typically refers to closed-circuit fluid coolers that have a heat exchanger between the air and water, like a radiator on an internal combustion engine in a car, but water is sprayed over the coil for its evaporative cooling effect.

**Table 3.9** Recommended Water Quality Levels for Various Materials of Construction for Treated Circulating Water

| Property of Water | Galvanized Steel | Type 304 Stainless Steel | Type 316 Stainless Steel and Fiberglass |
|---|---|---|---|
| pH | 6.5 to 9.0 | 6.5 to 9.2 | 6.5 to 9.5 |
| Total Suspended Solids | 25 ppm | 25 ppm | 25 ppm |
| Total Dissolved Solids | 1500 ppm | 2050 ppm | 2050 ppm |
| Conductivity | 2400 (microohms/cm) | 3300 (microohms/cm) | 4000 (microohms/cm) |
| Alkalinity at $CaCO_3$ | 500 ppm | 600 ppm | 600 ppm |
| Calcium Hardness as $CaCO_3$ | 50 to 600 ppm | 50 to 750 ppm | 50 to 750 ppm |
| Chlorides (CL) | 250 ppm | 300 ppm | 750 ppm |
| Sulfates | 250 ppm | 350 ppm | 750 ppm |
| Silica | 150 ppm | 150 ppm | 150 ppm |

*Data from Baltimore Aircoil Company Water Quality Guidelines*

Another classification characteristic is the flow path relationship between the air and water within the tower. The two most commonly used cooling towers in DCPs are cross-flow and counterflow. Cross-flow towers are configured to flow air horizontally across the downward path of the water and counterflow towers move air upward through the fill against (i.e., counter to) the flow of water. The fill acts as the heat transfer media in the cooling tower. Fill increases the surface area between the airflow and the waterflow for maximum contact. Increasing the contact time between water and air flows, increases the cooling tower's effectiveness and efficiency.

A further method of characterizing cooling towers is the air is circulated through the tower by either natural ventilation or mechanical draft using fans. Mechanical draft units are further classified as whether the airflow is forced draft at the air inlet of the tower or induced draft at the outlet of the tower.

Cooling tower fans also differ and may impact tower electric power consumption. Axial flow and centrifugal fans are commonly used in cooling towers; however, axial flow fans integrated with cross-flow or counterflow consume less power and are more commonly found in large DCSs.

Centrifugal fans are used where a large static pressure is required due to cooling tower location, particularly when indoor towers are required, or acoustic louvers or sound attenuators are needed, which typically are forced draft applications. Figures 3.44 and 3.45 schematically show the difference between the counter and cross-flow towers.

As shown, the cross-flow type cooling tower relies on water that flows by gravity to contact with air that flows crossing the water droplets. The fill is located around the tower perimeter with small fill width. Cross flow towers are typically packaged units and have a maximum capacity less than 1500 tons per cell. They can be arranged side by side in a single row but manufacturer spacing should be followed so the airflow isn't constricted.

This configuration minimizes pumping pressure as the water falls by gravity and fan pressure as the fill is not deep. Accordingly, the fan and condenser-water pumps consume less power.

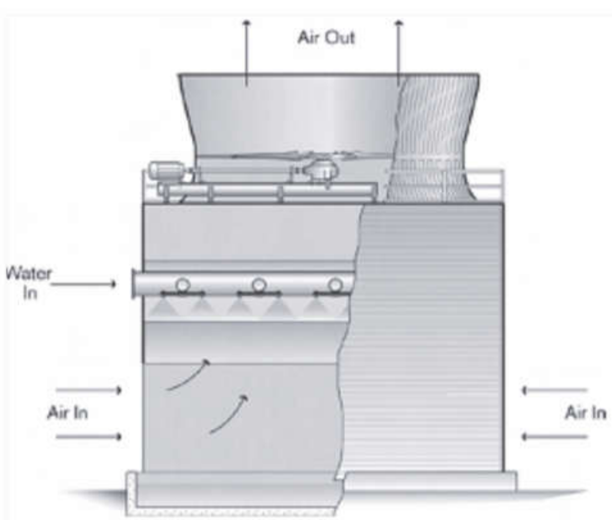



**Figure 3.44** Induced-draft counterflow tower.
*Courtesy of SPX Cooling Technologies*

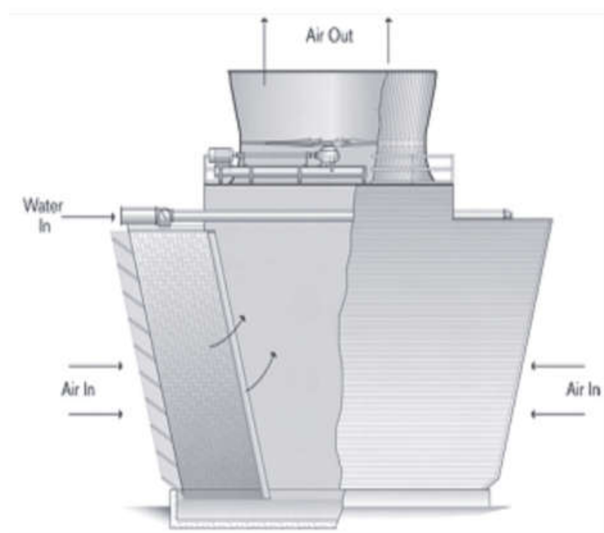



**Figure 3.45** Induced-draft cross-flow tower.
*Courtesy of SPX Cooling Technologies*

In the counterflow concept, the air moves vertically upward through the fill, counter to the downward fall of water thus the driest air encounters the coolest droplets of the falling water that improves the tower performance. To distribute warm condenser water over the fill, high-pressure spray nozzles are used. Counterflow towers are typically used in larger plants because their capacities can easily exceed 3500 tons per cell.

Because of the need for extended intake and discharge plenums, the use of high-pressure spray systems; and the typically high air pressure losses, some of the smaller capacity counterflow towers are typically taller; requiring more pump head; and utilizing more fan power than cross-flow towers. However as capacity increases, the installed fan horsepower should be less than cross-flow options.

In addition, it may be concluded that counterflow towers have a smaller foot print as compared to cross-flow towers and are well utilized where the plant foot print is of a limited area. Pump head is higher in counterflow cooling towers due to spray nozzle pressure drop and consequently condenser pump heads are higher compared to cross-flow cooling. As the intake louvers in cross-flow cooling towers takes up the tower side majority, less resistance to air flow is expected and consequently lower fan power.

Due to their higher thermal efficiency, counterflow towers are the best option in plants where height does not pose a limitation. Cross-flow towers are sub-classified by the number of intake sides that are served by each fan. Towers can have multiple air inlets where a fan induces air through either one, two, three, or even all four sides. The pros and cons of different types of cooling towers normally utilized in DCPs are summarized in Table 3.10.

## Tower Selection

Tower selection is affected by the thermal load to be rejected, inlet hot-water temperature, outlet cold-water temperature, and design wet-bulb temperature. Design wet-bulb values are typically derived from the Evaporation WB 0.4% value from ASHRAE (2017) and designer experience in the region. Furthermore, this value is typically increased by 2°F (1.1°C) by designers as a safety factor to account for fouling of the fill, deleterious aging affects, and climate resiliency.

Figure 3.46a graphically shows the relationship of range and approach as the heat load is applied to the tower for a typical moderate climate condition and Figure 3.46b shows the same relationship but for a more severe climate typical of the Middle East for example. The approach temperature is defined as the difference between the cooling tower leaving-water temperature and the ambient wet-bulb temperature and relates to the size and efficiency of the cooling tower. Unreasonably low approach temperature requirements will result in oversized cooling towers and have cost implications. Furthermore, typically cooling towers are not selected with an approach less than 5°F (2.8°C) due to the fact the Cooling Technology Institute (CTI) will not certify a tower's performance below this approach temperature.

Figure 3.47 indicates the increase in tower cost because of reduced approach, and it is recommended to select towers based on the ASHRAE maximum wet-bulb temperature after adding the circulation impact on wet bulb with 7°F (3.9°C).

Figure 3.48 shows the effect of the chosen approach on tower size at a fixed heat load, gpm, and wet-bulb temperature.

**Table 3.10** Comparison of Field-Erected Versus Packaged Cooling Towers

| Field-Erected Counterflow Cooling Towers<br>Advantages | Packaged Cross-flow Cooling Towers<br>Advantages |
|---|---|
| ***Service life***—Typically longer life since constructed of fiberglass or concrete with concrete, stainless steel, or fiberglass basin.<br>***Flexibility***—More flexible due to multiple variety of wet deck available to suit type of liquid being cooled. Each cooling tower is specially designed and constructed for the site and application. Available in both belt drive and gear drive. Can effectively wet the fill at variable-flow conditions down to 40% of peak load.<br>***Maintenance***—Easier to maintain since cold-water basin is open on all sides with no restrictions from wet deck.<br>***Space usage***—For many applications this style of cooling tower has the smallest footprint of any cooling tower design. Less space is needed because of this increased efficiency and lack of plenum space required for cross-flow cooling towers.<br>***Energy use***—Induced-draft counterflow cooling tower design offers lower fan power design compared to most other styles.<br>***Construction***—The wet deck (fill) is supported from structural supports underneath. This prevents sagging and creates a working platform on top of the fill for service. Upper casing is under negative pressure, reducing the risk of water leaks. Water distribution system is constructed of noncorrosive PVC piping and ABS nozzles.<br>***Design***—Entire working system is guarded from the sun's rays and helps reduce algae growth. The wet deck (fill) is encased on all four sides. The prevailing winds do not directly affect the fill. Air inlet louvers serve as screens to prevent debris from entering system. | ***Flexibility***—Available in both belt drive and gear drive.<br>***Size***—Usually shorter in height than counterflow towers.<br>***Costs***—Typically lower first cost than counterflow towers.<br>***Construction***—Water distribution system is constructed of noncorrosive PVC piping and ABS nozzles.<br>***Design***—Most sizes are usually FM rated and do not require fire sprinkling. Usually CTI certified for performance. Lower draft losses.<br>***Delivery***—Shorter manufacturing and installation lead times. Arrives fully assembled (larger units come in two pieces). |

| Field-Erected Counterflow Cooling Towers<br>Disadvantages | Packaged Cross-flow Cooling Towers<br>Disadvantages |
|---|---|
| ***Costs***—Typically higher initial cost vs. cross-flow.<br>***Size***—Typically taller than other styles and require handrails or piping at top of tower if motor is mounted on top of tower.<br>***Layout***—Requires airflow on all four sides for maximum performance.<br>***Delivery***—Longer to engineer and construct.<br>***Construction***—May have to be sprinkled, depending on materials of construction. | ***Service life***—Usually shorter life due to limited material selection.<br>***Maintenance***—Requires cleaning of hot-water basin on top of tower where applicable. Difficult to clean cold-water basin under wet deck (fill) because of limited access. May require handrail, safety cage, and service platform to meet local code requirements. Usually have shorter life than counterflow towers. However, some manufacturers offer internal platforms for servicing of motors, therefore, eliminating the need for handrails and ladders on the exterior of the unit.<br>***Flexibility***—Shorter height means more prone to recirculation effects and derating. Limited configurations available to fit layout. Care must be taken not to lay out more than two towers side by side or middle cells will be difficult to access, and outer cells may have to be shut down to service inner cells.<br>***Construction***—Wet deck (fill) is encased on two sides only. The prevailing winds directly affect the fill. If rods support the fill, wear may deteriorate the fill making it sag, which may affect performance. Open gravity hot-water basins require balancing valves to insure even flow and maximum performance. Most working components exposed to sun’s rays.<br>***Energy Use***— Higher installed power and operating costs. |

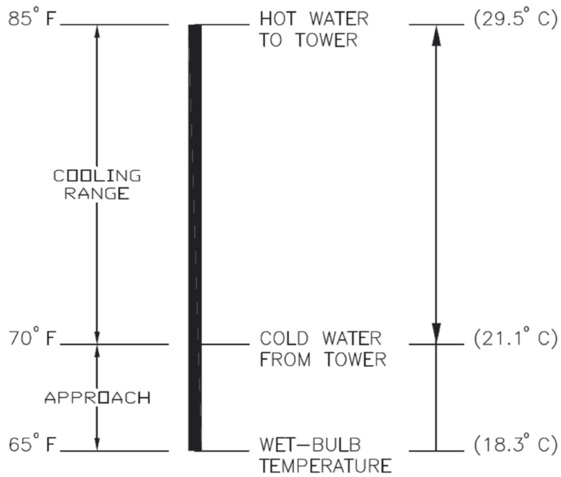



**Figure 3.46a** Diagram showing definition of cooling range and approach. Note: All temperatures used are illustrative only and subject to wide variation.
*Courtesy of SPX Cooling Technologies*

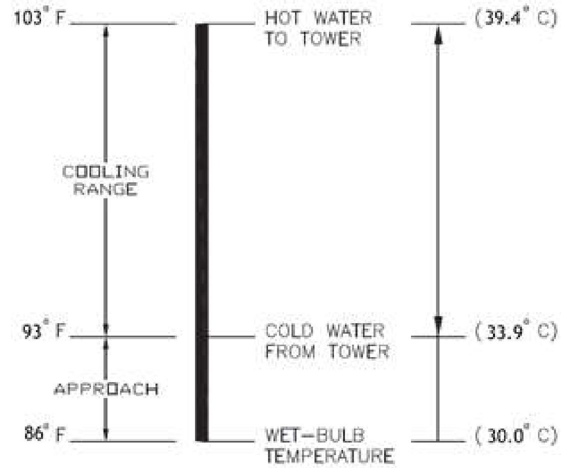



**Figure 3.46b** Diagram of cooling range and approach temperatures typical of Middle Eastern climate conditions.

The recirculation impact and increase of wet-bulb temperature can be calculated most accurately through computational fluid dynamics (CFD) modeling, typically carried out by the cooling tower supplier; however, it may be estimated from the following equations:

For dry coolers:

$$\text{Percent recirculation} = \frac{\text{intake temperature} - \text{ambient temperature}}{\text{discharge temperature} - \text{ambient temperature}} \times 100 \tag{3.5}$$

where the temperatures can be expressed in any consistent set of units.

For wet coolers:

$$\text{Percent recirculation} = \frac{\text{intake humidity} - \text{ambient humidity}}{\text{discharge humidity} - \text{ambient humidity}} \times 100 \tag{3.6}$$

where humidity is expressed in absolute units of moisture content, for example, grains of moisture per pound of air (grams of moisture per kilogram of air).

## Fan Speed Type

Cooling tower fans are CS, dual speed, or VS. The fans should be capable of moving large amounts of air efficiently, yet with minimum vibration. A vibration switch should be provided to switch the tower off when a potentially damaging vibration threshold is exceeded. The fan blade materials and construction must be capable of withstanding the

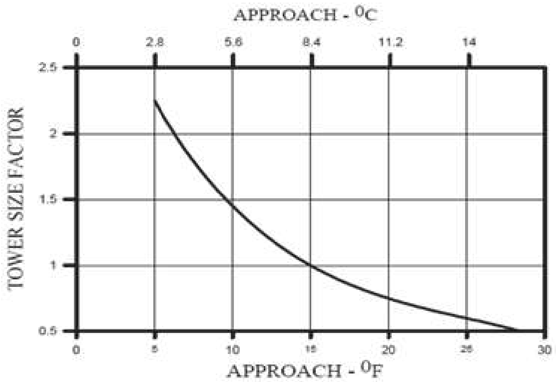



**Figure 3.47** Effect of chosen approach on tower size at fixed heat load, gpm, and wet-bulb temperature.
*Courtesy of SPX Cooling Technologies*

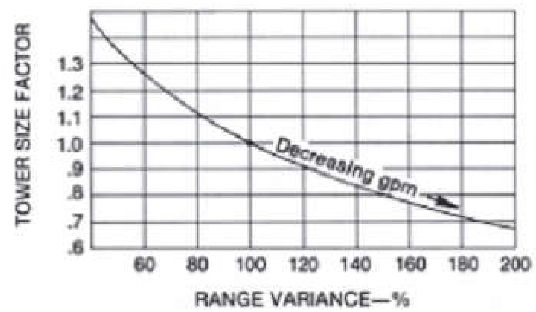



**Figure 3.48** Effect of varying range on tower size when heat load, wet-bulb temperature, and cold-water temperature are constant.
*Courtesy of SPX Cooling Technologies*

corrosive effect of the environment. Due to large delivered volume of air at low static pressure, cooling towers with propeller type axial fans are commonly used in DCPs. Fan noise in standard type fans is around 80 to 82 dBA; however, fans below 72 dBA are available in the market as special order as "low noise" or "whisper quiet" options (Figure 3.49). The noisiest exposures are the top and the air inlet sections. As a safety measure, a fan discharge hood should be provided with a fan guard when the hood is less than 6 ft (1.8 m) high.

Due to high inrush current at motor startup and consequent heat buildup in the motor windings, the number of start-stop or speed change cycles should be limited. As a rule, 30 s of acceleration time per hour should not be exceeded. For example, if a tower fan motor requires 15 s to achieve full speed, the number of starts per hour should be limited to two.

A cooling tower's performance and efficiency will impact heat rejection and consequently chiller efficiency. The best result is if the fans modulate their output to match the heat rejection requirement of the chiller. Staged fans can result in poorer control of condenser water temperature and lower chiller plant efficiency.

ASHRAE Standard 90.1 has a few provisions that impact cooling tower energy consumption including:

- Fans greater than 5 HP shall be cable to operate at two-thirds speed or less (i.e., two-speed motors or VFDs)
- Water flow limitations per fan horsepower as well as capability to turn down flow to 50% of design flow.

Electric motors used in conjunction with VFDs should be inverter-duty rated per Part 31 of NEMA MG.1-2011 or other applicable codes and standards.

## Draft Type

As previously noted, cooling towers are also classified by the draft type, either forced draft, in which the fan is in the ambient airstream entering the tower and the air is blowing through the cooling tower, or induced draft wherein a fan is located in the exiting airstream and air through the tower (Figures 3.50 and 3.51).

**Figure 3.49** Whisper quiet fan blades.
*Courtesy of Steve Tredinnick*

Forced-draft towers are characterized by high air-entrance velocities and low exit velocities. Accordingly, they are subjected to recirculation of moisture-laden discharge air and are therefore considered less efficient than induced-draft towers. Forced-draft fans typically use more power when compared to induced-draft units, but they are easier to sound attenuate due to the configuration of the fan inlet and may be used in sound-sensitive areas.

Most induced-draft towers have an air-discharge velocity of three to four times higher than their air-entrance velocity and therefore the potential for recirculation is reduced. The normal discharge velocity from an induced-draft tower is about 20 mph (32 km/h), whereas the plume velocity leaving a forced-draft tower is approximately 5 to 6 mph (8 to 9.5 km/h) (SPX Cooling Technologies 2009). The increase in the ratio between the plume velocity and wind velocity will affect the recirculation percentage. Figures 3.52 to 3.54 show how velocity and shape of plume will affect the recirculation percentage.

## Tower Location and Layout

Due to the possibility of recirculation, it is advised that towers are properly oriented such that plume will not be in the intake direction as presented in Figure 3.55. Figures 3.57 and 3.58 are guides for proper distance and orientation of cooling towers in district plants relative to the prevailing wind direction. Generally, the guides represent locating a large cooling tower on grade in an open area similar to a power plant, and since many DCPs are in urban areas and on the roof of the plant, the wind patterns are affected not only by prevailing winds but also adjacencies of taller structures and layout of the tower.

Prevailing wind direction can be obtained from a wind rose. Wind roses show the quantity of hours per year the wind blows from a specific direction and can also indicate average wind speeds. Wind roses are valuable in locating cooling towers away from outside air intakes as well as any structures that obstruct airflow or force re-entrainment of tower effluent and are typically available for major cities. Figure 3.58 shows a wind rose for Dubai, United Arab Emirates and indicates that the prevailing winds come from the west necessitate locating the tower to the east of any outside air intakes.

Packaged and field erected tower manufacturers also have guides as to minimum recommended distances between solid walls, screened or louvered walls as well as adjacent tower cells to mitigate recirculation or re-entrainment of moist air into the tower. Similarly, the top of the fan shroud should be as tall or taller than the tallest adjacent element

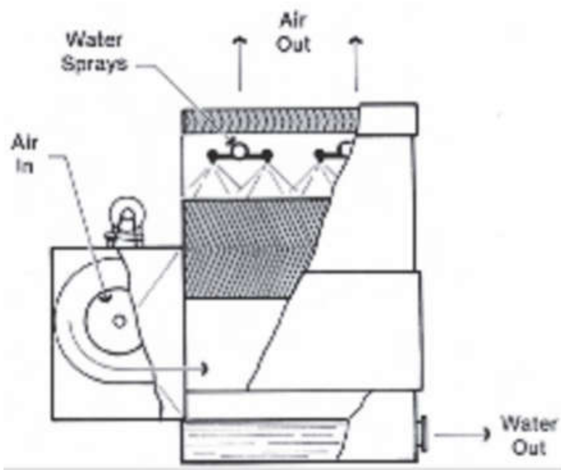



**Figure 3.50** Forced-draft, counterflow, blower-fan tower.
*Courtesy of SPX Cooling Technologies*


**Figure 3.51** Induced-draft, cross-flow, propeller-fan tower.
*Courtesy of SPX Cooling Technologies*

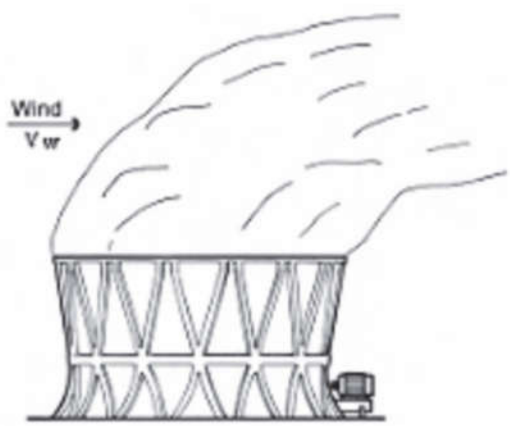



**Figure 3.52** Effect of wind velocity and discharge velocity on plume behavior. Note that the higher the discharge velocity, the better the plume velocity.
*Courtesy of SPX Cooling Technologies*


**Figure 3.53** Recirculation potential in forced-draft cooling tower.
*Courtesy of SPX Cooling Technologies*

(walls and screens). The manufacturer will normally provide package tower layout guidelines, typically limit the number of cells in a single row grouping to four and then desire a space between the next grouping of cells. If the layout guidelines cannot be met, then the manufacturer should be consulted to confirm the designed layout is not derated in any manner or the selections adjusted.

The designer should be sensitive to the adjacent areas where plume will migrate, not only for legionnaires bacteria and outside air intakes, but also the fact that due to the high content of TDS, the water is hard and will cause water spots that are difficult to remove. This is an issue with any adjacent glazing as well as automobile finishes. In these sensi-

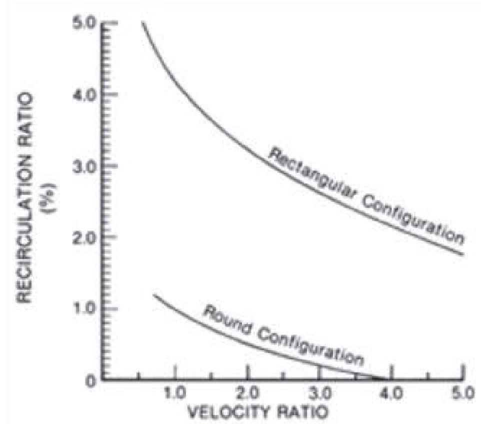



**Figure 3.54** Comparative recirculation potential of round and rectangular towers.
*Courtesy of SPX Cooling Technologies*

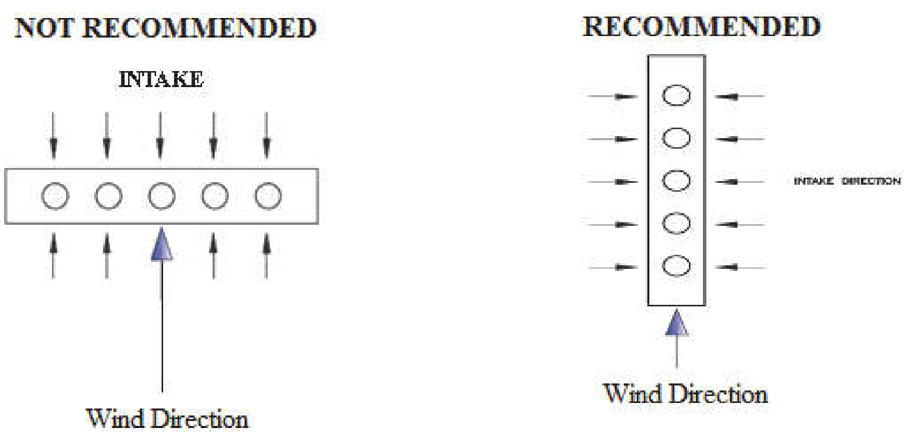



**Figure 3.55** Recommended tower cells orientation.

tive areas or where designs deviate from the tower manufacturer's recommendations or the designers prior experience, it may be advisable to conduct simulations via CFD.

## Tower Basin

Large cooling towers located on-grade normally have concrete basins, whereas roof-mounted units could be fabricated with concrete, stainless steel, or fiberglass provided by the cooling tower manufacturer or constructed on site. Concrete basins are constructed by the structural contractor and must have close coordination with the cooling tower manufacturer's data. Fiberglass basins are typically used with fiberglass towers.

The pressurized condenser water supply and the return pipes are headered in multiple cooling tower (CT) installations with a valve on each cell to open or close when a cell is utilized or isolated from the rest of the cells. With headered towers, the supply flow to each cooling-tower cell must be balanced at the tower using balancing valves to ensure that each active cell gets approximately the same flow since the total system flow is affected by how many cells and condenser water pumps are in operation. If suction is not the same from the cooling tower cells, an equalizing line sized for 20% to 30% of the flow should be provided and fitted with an isolating valve to close the cell during mainte-

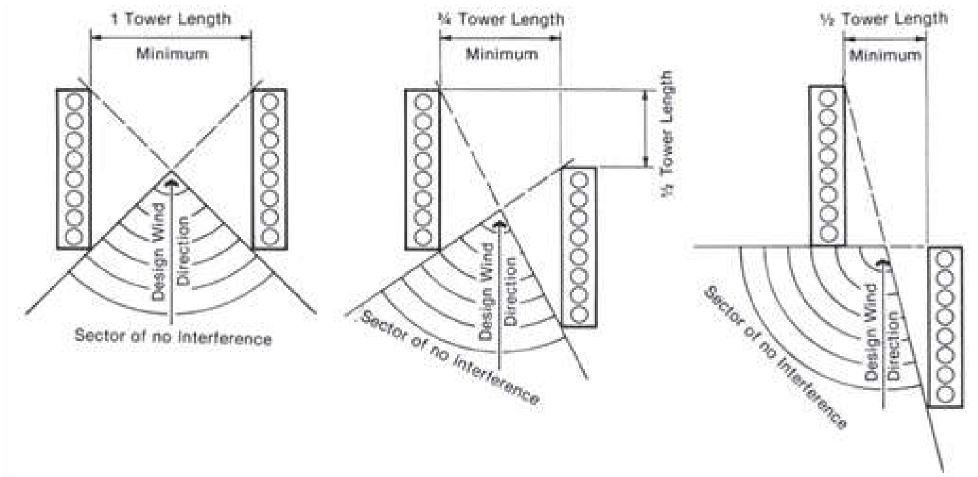



**Figure 3.56** Proper orientation of towers in a prevailing longitudinal wind. (Note: this requires relatively minimal tower size adjustment to compensate for recirculation and interference effects.)
*Courtesy of SPX Cooling Technologies*

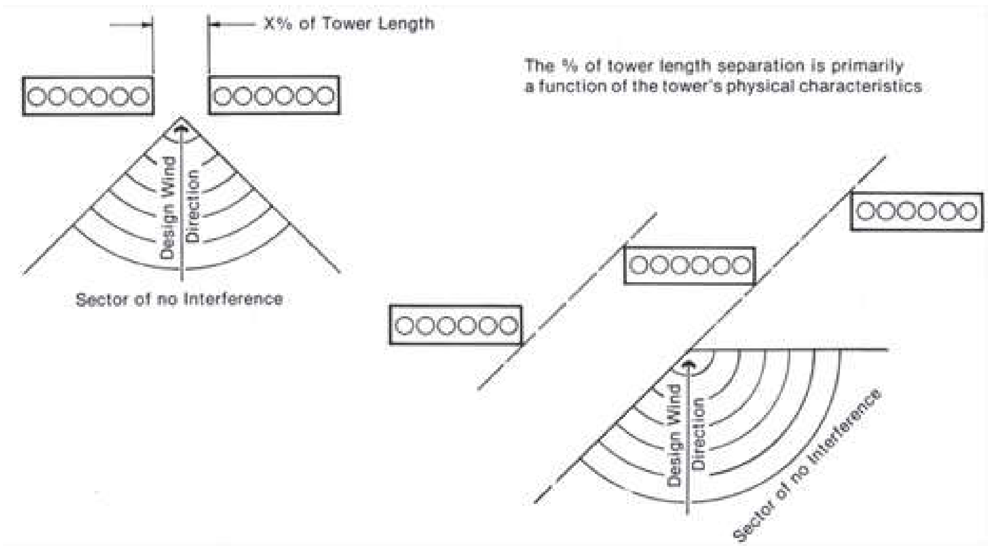



**Figure 3.57** Proper orientation of towers in a prevailing broadside wind. Note: this requires significantly greater tower size adjustment to compensate for recirculation and interference effects.
*Courtesy of SPX Cooling Technologies*

nance. The concrete basin of multiple cells should be subdivided into sections to form one basin for each tower for easier isolation of the tower during maintenance.

Where the basin sump pits are used, the sump entering water velocity should not exceed 3 ft/s (0.9 m/s). Velocities lower than this are recommended. The sump should be of sufficient depth to satisfy pump NPSH requirements.

The depth of water above suction outlets should be deep enough to avoid water vortex creation, which leads to air intrusion into the system. Anti-vortex plates can be provided by packaged tower manufacturers or the piping contractor for field erected towers.

The sump/basin capacity should accommodate the water volume that is in suspension water contained within piping above the sump as well as water in the fill will flow back into the sump when the tower is switched off (transient water in the system at shutdown).

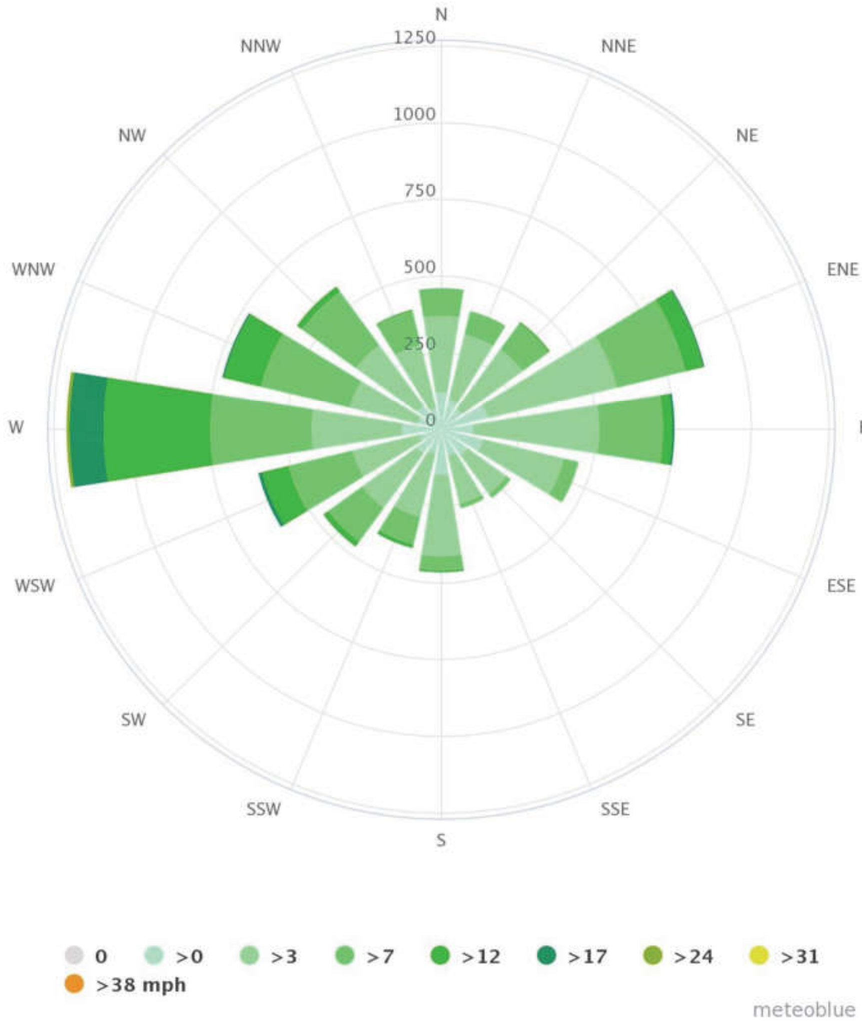



**Figure 3.58** Wind rose for Dubai.
*Obtained from https://www.meteoblue.com/en/weather/forecast/modelclimate/dubai_united-arab-emirates_292223*

The basin or sump depth is normally 24 in. (600 mm) in addition to the aforementioned requirements. More depth may be added to act like a water reservoir should makeup water interruption take place or to stabilize water temperatures under highly variable loads or to act as a reservoir to supply the plant fire protection system if AHJ accepts. Furthermore, if the sump is intended to "fill" the condenser water system upon startup or after a shutdown, additional capacity is required to satisfy several minutes pump runtime to compensate for the water circulated from the sump to the chiller condenser and back to the sump without pumping the sump "dry."

The basin could be concrete, glass fiber, treated galvanized steel, or stainless steel. Concrete basins and sumps having waterproofing additives are most common in large DCPs.

## Tower Fill Options

The tower fill acts as the heat exchanger between the ambient air and the condenser water and the most important component of the cooling tower. There are two basic classifications of fill, splash type and film type, and the applicability of their use is dependent upon what style of tower is used. For example, packaged cooling towers, whether counterflow or cross flow, all use polyvinyl chloride (PVC), polypropylene, or other polymer fills that are a film-type classification. PVC is a light weight, relatively low-cost product that is thermoformed into corrugated sheets and then constructed into stackable blocks that create a great deal of surface area for water and air contact and offer multiple air paths and channels with a typical honeycomb shape. There are different thicknesses available for use. Thicker PVC with a larger honeycomb design is advisable if the water is not clean and could foul a thinner and smaller opening. However, with thicker media also comes an energy penalty since it is not as efficient as rejecting heat as thinner media. Film-type fill can also be used in field-erected towers. Film thicknesses from 10 to 25 mils (0.25 to 0.63 mm) are available with 15 mil (0.38 mm) being a typical thickness. PVC is limited to 125°F (51.7°C) condenser-water temperatures; other fill materials, such as chlorinated polyvinyl chloride (CPVC), should be investigated if the temperature will be higher.

Splash fill is almost entirely used in field-erected cooling towers and there are several methods. Simplistically, the fill is layered within the tower and water is sprayed on it and splashes. With each progression of splashing, the water droplets decrease in size and can exchange heat more effectively with the air. Splash fill can be treated wood, fiberglass, steel, plastic, and ceramic.

The tower manufacturer should be consulted on their recommendation on fill type based on the material requirements to be resistive to fire and smoke spread, fireproof, quality of water, etc.

## Materials of Construction

While the internal components of cooling towers can be very similar from style to style, the exterior materials of construction can be varied. Prior to the advent of plastics, wood constructed cooling towers were extremely popular. While wood was readily available and relatively inexpensive, it had the disadvantages of being extremely heavy, less durable with decreased life due to rotting and decay, as well as being prone to algae formation and combustible. Galvanized steel is the most cost-effective material for the construction of packaged towers with G-235 being the heaviest galvanizing thickness available, and it offers reliable corrosion protection. Stainless steel, while being more expensive than galvanized steel, is extremely durable and is used where longer tower life is desired or the location of the tower is in a corrosive area such as coastal areas.

Fiber-reinforced plastic (FRP) has gained popularity because the material is lightweight but strong, chemically resistant, and fire retardant. Hybrid towers with stainless steel basins and fiberglass enclosures can also be fabricated by most packaged tower manufacturers. Field-erected cooling towers constructed out of concrete are exceedingly durable, having life expectancies similar to the building they serve, and they can be formed into many aesthetically pleasing shapes and finishes, but are extremely expensive and typically cost prohibitive.

## Water Sources

Water sources for the condenser water could be municipal domestic water, gray water, treated sewage effluent (TSE), ground/lake water, or seawater.

### Municipal Domestic

Municipal domestic or potable water is most favored by designers due to its availability, high level of cleanliness, requirement of less space for the treatment-process equipment, and lower TDS. However, in some regions, due to scarcity of water, coupled in some cases (i.e., the Arabian Gulf region) with the expansion of water-cooled district systems to serve the rapid development of new construction, governments have issued regulations to limit its use or they have increased the domestic water tariff rate. The TDS of municipal water is normally in the range between 300 to 450 TDS.

### Seawater

Where available, seawater is a potential heat-rejection medium. For example, as Arabian Gulf developers target locations for their properties, water fronts and islands are the developer's prime selection. Seawater at such locations is normally available for condenser water use. The seawater might be used in a once-through manner or as a makeup for evaporative seawater cooling towers. Due to the impact of temperature change on marine life and the small difference between suction and discharge temperature, the once-through system installation will be a costly choice. Consequently, evaporative seawater cooling has become the choice of several developers as well as district energy providers in regions close to seawater and where municipal domestic water is a precious resource and thus costly. The once-through system might be the only seawater option if cooling towers cannot physically be installed. The seawater temperature must be determined if a once-through system is the choice. For example, the seawater temperature in certain Arabian Gulf locations might reach as high as 100°F (38°C), which will impact plant chiller energy use.

Evaporative seawater faces some design challenges, including the presence of concentrated salts in drift and blowdown and on the size of towers. To avoid drift issues, the cooling towers should be designed with a maximum drift rate of 0.0005%. The arrangement and orientation of the cooling towers needs to be studied such that the plume direction shall not discharge towards sensitive items like electrical switchgear and other similar items. A CFD model should be carried out to demonstrate that the plume will not impact surrounding buildings and sensitive components based on the frequency of wind directions.

The blowdown water TDS concentration should not exceed 55,000 ppm, which is also considered the maximum limit acceptable by chillers and cooling tower manufacturers. The blowdown water may be diluted prior to discharge back into the intake source to make sure concentration will not exceed the values recommended by the environmental authorities.

Cooling tower thermal performance, and consequently its size, will also be impacted by the seawater properties. The vapor pressure will be lowered, which reduces the rate of evaporation. The water density will be increased, which increases the thermal performance; however, the specific heat will be reduced resulting in thermal performance reduction. The end result of the above water properties will lead to a cooling tower net efficiency reduction of approximately 7.5%.

The designer should also consider the appropriate system materials to use for all equipment that contacts the seawater. For example, the chiller tubes must be stainless steel or titanium, the cooling towers should be fiberglass with stainless steel hardware, and the condenser pumps should be duplex stainless steel. The condenser and makeup water pipes should be nonferrous and should be suitable for seawater. High-density polyethylene and glass reinforced plastic are suitable materials for condenser water pipes in seawater applications.

Elaborate intake and discharge chambers must be provided and evaluated using lifecycle cost analysis methodology to demonstrate the system feasibility as compared to other traditional alternatives.

### Treated Sewage Effluent

Treated sewage effluent (TSE) is an alternative to municipal domestic water and may be used if available as makeup to the cooling towers. The TSE is characterized by its high TDS, suspended solids (SS), and might also contain bacteria. TSE will normally require polishing, disinfection, and filtration systems to enable its use, but it still will remain non-potable. With potable water becoming a more precious commodity, TSE is becoming more of a viable alternative and its use is becoming mandatory in several countries in the Arabian Gulf area.

The TSE water will require treatment prior its use and the treatment process is dependent upon the TSE water quality. The TSE water adopted in the Arabian Gulf is distributed with the following characteristics:

Influent Water:

- Total dissolved solids (TDS): 1250 ppm
- Feed water temperature: 25°C–35°C
- Total suspended solids: 1–20 ppm
- Chemical oxygen demand (COD): 5–40 ppm
- Biochemical oxygen demand (BOD): 0.5–5 ppm
- Total coliform: Maximum 2000 CFU/100 ml
- Fecal coliform: Maximum 220 CFU/100 ml
- E-coli: Maximum 200 CFU/100 ml

The water quality acceptable to the majority of manufacturers is as follows:

- Total dissolved solids (TDS): <500 mg/l
- pH: 6.5–8.0
- Chlorides: <750 mg/l
- Sulfates: <20.0 mg/l
- Sodium bicarbonate: <200.0 mg/l
- Suspended Solids: Negligible mg/l

To meet the above water quality requirements, a polishing reverse osmosis (RO) plant is required. The RO plant will reduce TDS of TSE, thus allowing the tower operator to circulate water several times prior to blowdown.

To prevent fast closing and damage of the RO membrane, a pretreatment and filtration process should take place. The permeate (water from the RO) should be subjected to chemical treatment to maintain the makeup water treatment.

The pretreatment normally consists of media filter (or two-stage filter) followed by activated-carbon filter to remove organic material that is harmful to the RO cells. Coagulants, antiscalant, and chlorine are added. The media filter is to be backwashed using filtered water. Due to heavy cake that will be collected in the media filter, air blowers are normally used in addition to the backwash pump to properly backwash the filters. Chemicals shall be injected on the discharge from the media filter as follows:

- Acid: to maintain acidic fluid to the membrane
- SBS: to remove the chlorine residuals
- Sodium: to protect the filter media of the ultra filter and the RO

The water from the media filter will pass through an ultra filter (UF) or a micron filter (MF), then will pass through an ultraviolet (UV) unit to kill the bacteria prior to passing to the RO. Water from the RO permeate is normally collected in a break tank then pumped to the CT basin. The total hardness of this water shall not be <50 ppm to enable

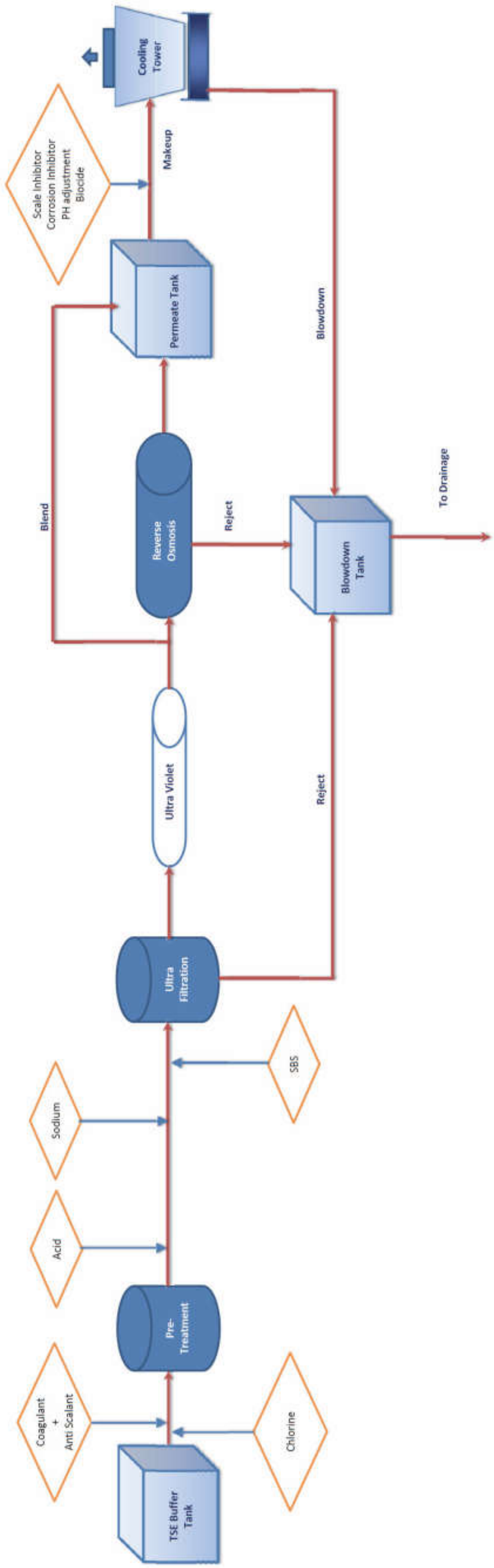



**Figure 3.59** TSE treatment for cooling tower use.

adding cooling tower chemicals. The following chemicals are normally injected as tower treatment:

- Scale inhibitor
- Corrosion inhibitor
- pH adjuster
- Biocide

A bypass should be provided around RO to reduce RO operating costs when treatment is not needed. The ejected water from the RO process should be blended with the media filter backwash and cooling tower bleed to achieve the level acceptable to the regulatory authority. The ejected water quality parameters should be obtained from local authorities of each municipality and country.

### Groundwater

If available, this source will often require the approval of the governing or regulating authority. The treatment of the system and storage requirement will be dictated by the well water quality and its possible peak discharge rate. Filtration may be needed, and RO could possibly be necessary, depending on the water quality. The use of alternative materials for chiller tubes, towers, piping, etc., as discussed above for seawater use, may be an alternative to treatment/purification under some circumstances. Furthermore, intake structures for use in lakes and rivers would follow the guidelines established for seawater.

## WATER FILTRATION SYSTEMS

Filtration of chilled- and condenser-water systems in DCSs is important to maintain clean chiller tubes and thus avoid degrading chiller performance.

A closed CHW system network must be properly flushed initially to remove all suspended dirt, particles, and debris from welding and installation, then subsequently filtered to continue the removal of any contaminants from the system. Flushing is mainly required at the startup of a DCS and might also be required if, due to development, building connections are taking place progressively and cross-contamination from the new direct connections cannot be avoided. Note that a minimum water velocity of 3 to 5 fps (0.9 to 1.5 m/s) must be achieved to have adequate force to flush pebble-sized debris from piping, hence a separate pumping source may be required during construction and startup to provide adequate flow to force the flush of debris.

The open type condenser-water system cooling in a DCS is subject to more dirt and debris accumulation particularly when the plant is located in areas having other construction activities. Water in condenser-water systems is extremely turbid. As mentioned previously, the towers are open to the atmosphere and act as air scrubbers, collecting all kinds of dirt that settles in the basin and low points of the system.

Conventional filtration system technologies are sand filters, centrifugal separators, multimedia filters, and multicartridge vessels. Depending on the technology used in sand and multimedia filters, one can experience bad fouling, high backwash requirements, and high initial and operating costs of duplex systems. Experience has shown that multicartridge vessels plug rapidly and are labor intensive and costly to replace.

Table 3.11 is an excerpt from the US Federal Energy Management Program that summarizes the characteristics of the major side stream filter equipment options. Any fine sediment that settles in cooling tower basins can act as a food source for *Legionella* bacteria. *Legionella* bacteria are less than two microns in size; therefore, filters providing efficiencies for particles below one micron are desirable.

Several approaches are utilized to filter the water of both CHW and condenser-water systems. Either side-stream or full-stream approaches are used depending on the severity

**Table 3.11** Side Stream Filtration System Characteristics

| Filter Type | Particle Removal Level | Self-Cleaning Features | Maintenance and Repairs | Water Loss from Backwash |
|---|---|---|---|---|
| Centrifugal Separators | 40-75 microns. Fine to coarse inorganics with a specific gravity of 1.62 or greater | Purged collected solids from the collection chamber | Purge components only—periodic inspection and servicing | No water loss due to backwashing. Water may be lost during the purging of the particle chamber |
| Automatic Screen Filter | Down to 10 microns | Automatic backwash by using a rotating suction scanner assembly | Regular maintenance may be required because of moving parts that enable automatic backwash | Requires much less water than other self-cleaning filters that utilize backwash cycles |
| Plastic Disk Filter | Down to 10 microns | Automatic backwash through releasing grooved discs and reversing water flow to wash collected solids off discs | Consumable discs can require frequent replacement | Requires much less water than other self-cleaning filters that utilize backwash cycles |
| Pressure Sand Filter | Down to 10 microns | Automatic backwash one a day or on pressure drop as needed | Requires regular inspection of stand media and electromechanical parts, and periodic replacement of sand media | Requires significant water for backwashing |
| High Efficiency Sand Filter | Down to 0.45 microns. Best for fine, light particles. Avoid heavy, coarse particle applications | Automatic backwash features, requires less time and water than other sand filters | Sand media must be monitored and periodically disposed and replaced | Requires more backwash water than centrifugal separators, automatic screen, and disc filters but about eight times less water than other sand filters |

*Recreated from data from October 2012 from the US Department of Energy, Energy Efficiency & Renewable Energy, Federal Energy Management Program (PNNL-SA-91274).*

of the filtration needs. The full-stream approach is to circulate all (100%) of the process water through the filtration system and the side-stream approach only circulates a part of the process water. Filter manufacturers should be consulted as to their recommended flow rate respective to their product.

The CHW network is typically a closed network and relatively clean; however, it still can accumulate sediment from the initial construction, adding customers and extending the distribution system. Therefore, a side-stream filter should be considered with a capacity of 5% to 10% of total water circulated. The filter could be hybrid sand filter, cartridge, or bag-type filters (see Chapter 8). Bag and cartridge type filters are available from several manufacturers in large sizes, yet the bags are disposable, and they offer a fine particles removal down to 5 μm that cannot be achieved by other technologies except sand filters.

An open type of condenser-water cooling system uses a side-stream filtration system with 5% to 15% capacity of circulated water or cooling tower sweeper system that consist of filters, pumps, and basin nozzles properly spaced and leveled in the tower basin to sweep out particles. When the environment contains a high level of particulate, a basin sweeper system may be used. Screens are available that offer some filtration measures for large particulate on the intake openings of cooling towers, but will require periodic main-

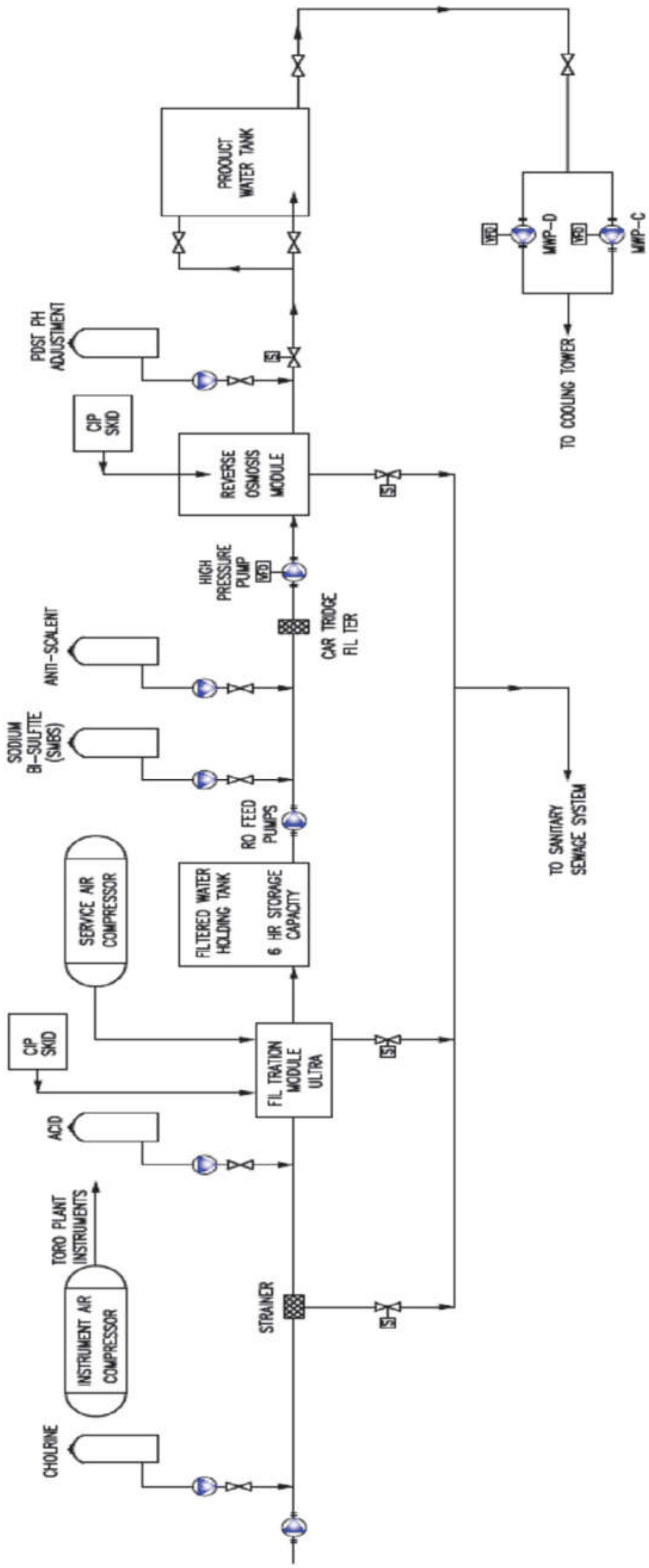



**Figure 3.60** Groundwater treatment flow diagram.

tenance to remove any debris collected. The screens protect the chiller and tower heat-transfer areas. Another very effective option is to use a common tower basin and sump for all towers where dirt can be easily collected and filtered from the system. The basin and sump should be segmented to facilitate cleaning and maintenance of cells without deenergizing the entire tower or pumps.

Figure 3.61 illustrates a typical condenser-water sump basin for a field erected cooling tower.

Where water-cooled DCPs are located in dirty or dusty areas and fine particulate in the cooling tower basin cannot be avoided, the chiller condensers will be susceptible to fouling and scaling. Fouling and scaling negatively impact the condenser tube-water heat transfer, increasing condenser temperature and increasing chiller energy consumption.

In such severe cases, regular condenser-tube cleaning might be difficult and automatic condenser-tube cleaning could be used. The automatic tube-cleaning system consists of plastic brushes that shuttle back and forth within the tubes using a reversing valve in the condenser-water line. The brushes should be cycled several times per day using an automatic controller. The automatic cleaning system will increase the condenser pump head by approximately 2 ft (0.6 m) but will avoid chiller capacity reduction by up to 10%.

## AIR VENTING

Air in water systems can cause major issues for district systems because it acts as an insulator for heat transfer and increases the corrosion in system components including chillers, pipes, and heat exchangers. Trapped air causes air pockets in the piping system that can accumulate at fittings and prohibit flow. This phenomenon is referred to as being "air bound." Figure 3.62 shows the impact of air in water on pump performance.

The air in a closed system is created through:

- Normal amount of air dissolved in the makeup water
- Air trapped in the system after the initial filling
- Diffusion

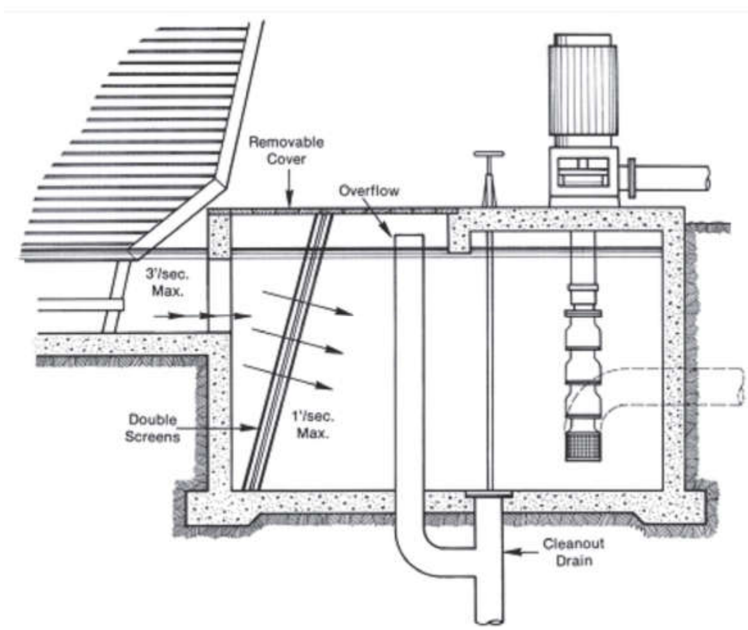



**Figure 3.61** Typical cross-section of concrete sump pit.
*SPX Cooling Technologies 2009*

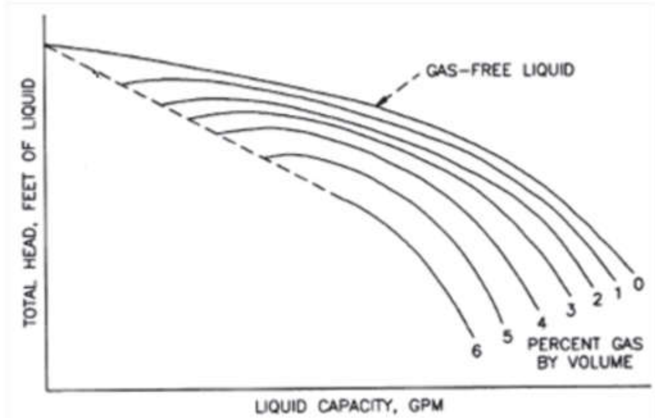



**Figure 3.62** Impact of gas pump curve and performance from HVAC Pump Handbook (Doolin 1963).
*Reprinted by special permission from CHEMICAL ENGINEERING (January 7, 1963)*


- Air ingress due to negative pressure

The air dissolved in the makeup water contains oxygen and nitrogen. The dissolved oxygen is quickly reduced by the initial corrosion process and will be completely used up within four to five hours of system fill. Air may be trapped in the system due to poor venting at the initial fill. The nitrogen content in cold water after several hours is three times higher than the initial filling.

Air diffusion mainly takes place through pipe materials that are not airtight like plastic and synthetic pipes and hoses, also through any expansion tank rubber bags/diaphragms. Piping installations constructed of traditional materials, such as steel and copper, have proven to be most reliable in avoiding air diffusion.

The loss of oxygen and nitrogen through diffusion in the expansion tank will result in loss of pressure in the expansion tank leading to a loss of pressure in the system. This loss in system pressure may lead to:

- A vacuum at the top of the system
- Drawing in large amounts of air
- Severe corrosion

When the pressure of the air cushion is lower than the static pressure, system water flows into the expansion tank. That water must come from somewhere like piping and heat emitters above the expansion tank. The automatic air vents located at the highest points in the system may open and become an automatic aerator, allowing large quantities of air into the system. To minimize the risk of diffusion, the designer should use expansion vessels with a bladder and not a membrane or diaphragm (see Figures 3.63 and 3.64).

The expansion tanks should be fully welded and not of the rolled-joint-seal type in order to provide the greatest possible gas tightness. The rubber bladder should be isobuteneisoprene rubber symmetrically suspended with vulcanized o-ring sealing.

Several devices are used to eliminate the presence of gas in CHW systems. Gas exists within the system in three forms: stagnant bubbles, gas entrained within the flow, and as dissolved gases. The large stagnant bubbles can be removed via manual or automatic venting if enough pressure is available. If bubble size exceeds the air vent passage, a capillary problem occurs and prevents the removal of the bubbles. In this case either a surface tension breaker (Figure 3.65) or a large bore vent (0.5 in. [12 mm]) (Figure 3.66) should be used.

**Figure 3.63** Expansion tank with a bladder.
*Courtesy of TA Hydronics*

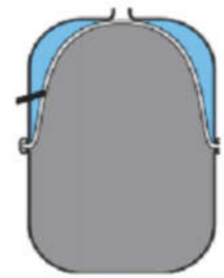


**Figure 3.64** Expansion tank with a membrane.
*Courtesy of TA Hydronics*

**Figure 3.65** Surface tension breaker.
*Courtesy of TA Hydronics*

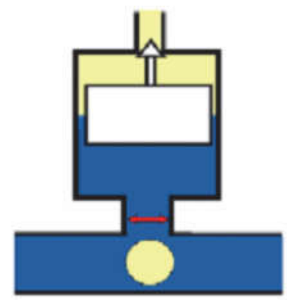


**Figure 3.66** Large bore vent used to pass air bubble.
*Courtesy of TA Hydronics*

The elimination of visible bubbles takes place most effectively when the water velocity is low (Figures 3.67 and 3.68).

Air vents will be effective for the elimination of stagnant bubbles that are separated from the flow and not totally dissolved in it, if the air vent passage is big enough to avoid a capillary problem. Also removal effectiveness will be improved in locations where velocity is lowest, such as plant headers.

Large and microbubbles in district systems are separated in water by the use of baffle separators, centrifugal separators, or wire mesh separators (Figures 3.69 through 3.71).

The separators are ideally suited for continuous gas venting in district plants. Figure 3.72 indicates the effectiveness of different approaches.

To enable proper separator function, the velocity through the separator should not exceed 6 fpm (0.03 m/s) and should allow for water turbulence. Some separators contain a zone to collect any dirt for blow down.

The air vents and separators are not effective tools for the separation and removal of the microbubbles. Degassers could be employed to remove and separate such gas bubbles.

Dissolved gas (invisible) is difficult to remove through separators or air vents but can be removed using a vacuum-pressure degasser. In this device, a fraction of the circulated water is put into a vacuum allowing the separation and removal of dissolved gas.

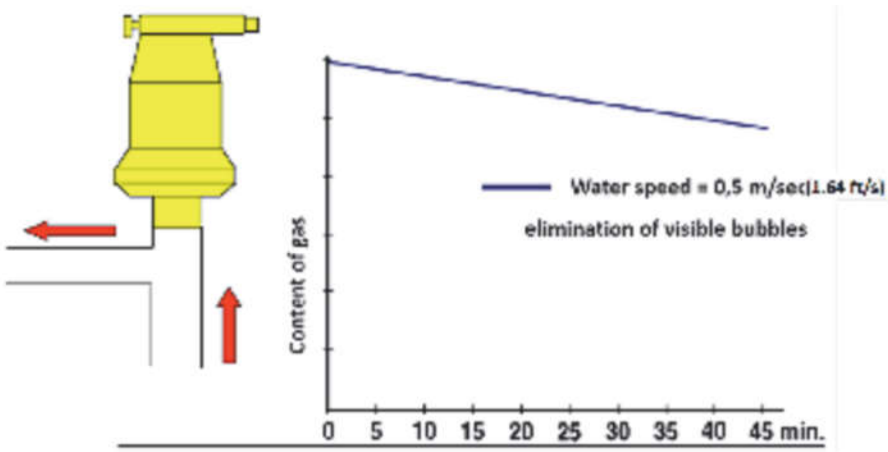



**Figure 3.67** Water speed versus removal time—ascending flow.
*Courtesy of TA Hydronics*

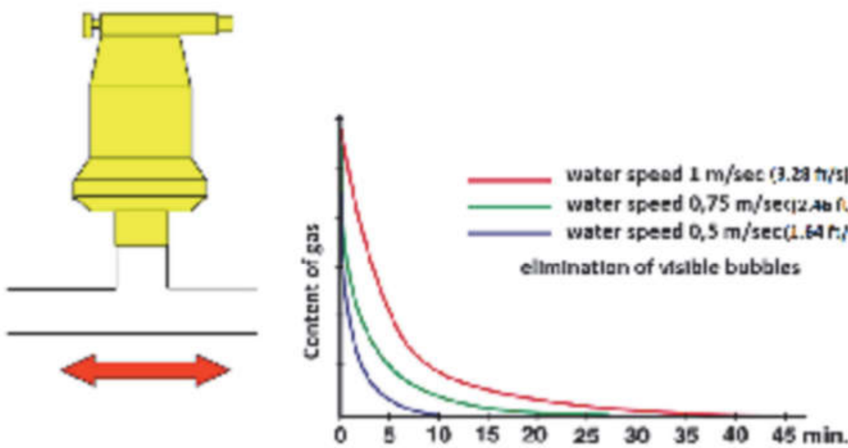



**Figure 3.68** Water speed versus removal time—horizontal pipe.
*Courtesy of TA Hydronics*

The microbubbles dissolved in water can be removed if the water is heated or put under negative pressure. Figures 3.73 through 3.75 show the nitrogen gas solubility at different pressures and temperature. It illustrates that gas contents in water will be reduced at a lower pressure. The gas removal device circulates part of the water and lowers its pressure, then separates the gas content, removes it, and returns water back to the system.

## PLANT PIPING AND INSULATION

Piping within the DCP is typically welded steel with flanged connections for valves and equipment. Sometimes mechanical couplings are used at equipment connections but their use should be minimized in a DCP because they are a weak link and potential point of failure in a robust welded piping system. Insulation materials commonly used in DCPs are polyurethane foam, polyisocyanurate, glass fiber, or cellular glass. Furthermore, to extend the life of the piping and mitigate external corrosion, it is recommended that the exterior surface of all piping be epoxy painted prior to insulating. This will prolong the

**Figure 3.69** Baffle separator.
*Courtesy of TA Hydronics*

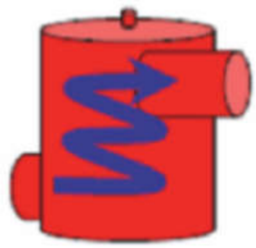


**Figure 3.70** Centrifugal separator.
*Courtesy of TA Hydronics*

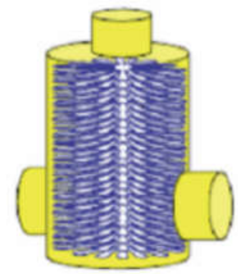


**Figure 3.71** Wire mesh separator.
*Courtesy of TA Hydronics*

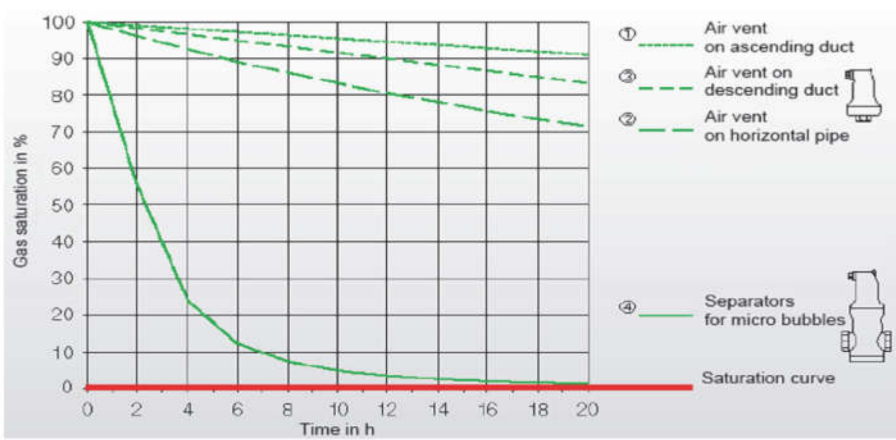



**Figure 3.72** Different trapped air venting tools.
*Courtesy of TA Hydronics*

life of the piping if the insulation or vapor barrier fails. A suitable vapor barrier and protective jacket (aluminum, stainless steel, or PVC) should also be used and the insulation, vapor retarder, and jacketing flame spread and smoke developed indices, as measured using ASTM Test Method E84, must comply with the applicable building codes or job specifications (ASTM 2012).

Refer to Chapter 4 for information on distribution piping systems.

## MECHANICAL ROOM DESIGN

The architectural features of a central plant will be somewhat dictated by the cost of land and its availability. District plants may be installed in a separate building or within a building used for other functions.

The plant building may only have a ground floor with the cooling towers located either adjacently on-grade or on the rooftop. Where minimizing the building footprint is a concern, plants are designed to be multifloor having basements to house tanks and water treatment pumps, a ground floor to contain chillers and electrical rooms, and cooling towers located on the roof of the building.

CHW TES tanks are commonly located on-grade, adjacent to the plant to keep costs down, but they also could be located below ground or anywhere along the distribution system. For ice-based TES, the tanks may be installed within the plant or outside the plant

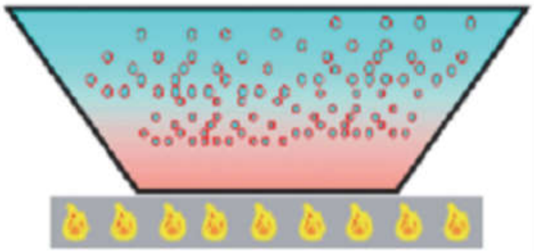


**Figure 3.73** Microbubble separation by heat.
*Courtesy of TA Hydronics*

**Figure 3.74** Microbubble separation by pressure reduction.
*Courtesy of TA Hydronics*

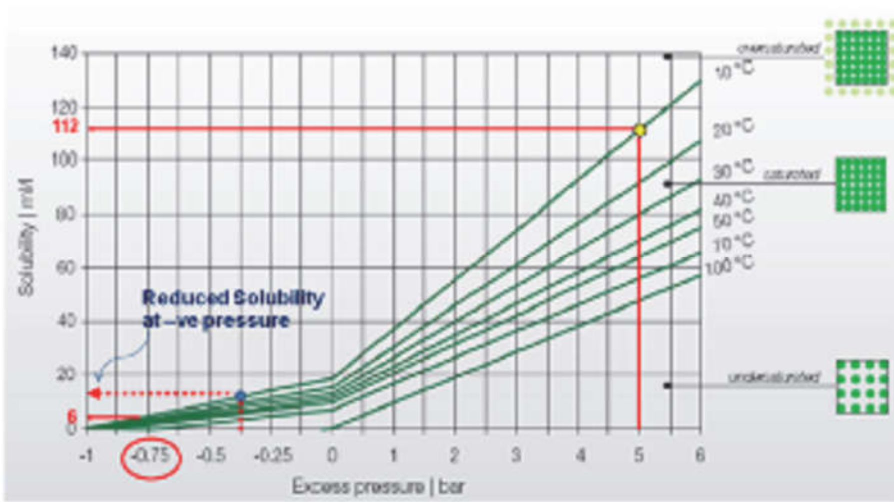



**Figure 3.75** Nitrogen reduction.
*Courtesy of TA Hydronics*

building as space permits. When ice storage is located within the building, a proper means of getting the ice banks in and out of the building must be implemented into the design.

In large district plants, pump room clear heights should not be less than 23 feet (7 m); whereas, chiller hall height should not be less than 26 to 30 feet (8 to 9 m) clear to allow for the installation of overhead cranes above chillers to remove the heaviest components, which are typically the motors and compressors. Monorails are also recommended running parallel to equipment to facilitate the removal of pump casings, pump motors, and chiller-marine water box covers.

Many times, a dedicated compressed air system is provided for the mechanics to use air-driven equipment for maintenance and cleaning purposes. This compressor should be separate from any temperature control air compressor that may be used. Strategically located hose bibbs should be provided for cleaning and flushing purposes and to connect a high-pressure water washer for chiller tube cleaning.

A chemical storage room should be provided having racks for chemical bulk storage tanks. The walls and floors of such rooms should have a suitable finish to withstand the stored material's corrosive characteristics. The room should have a lab-grade sink and cabinet for periodic testing of the water quality. All water-based coupon racks should also be located in this room. A chemical treatment contractor should be consulted as to how much space is required to house all the chemical tubs. Ideally, access to this room directly from the exterior of the plant is preferred so chemical tubs can be filled and tests can be run without interrupting plant operation staff. The chemical storage room should also house a wet-lab type sink with cabinets to have space and storage for equipment to test and analyze the quality of the water systems in the plant.

The plant should have a plant management zone where offices for the plant manager and supervisors are located, supply storage, a conference room, and control room should be located. The management zone floor should be isolated from other plant zones that vibrate or are noisy. Vibration matts and thick wall and floor construction should be used for this purpose. The control room should be adjacent to the chiller room and have a large acoustic window for observing the equipment.

Emergency eye wash and shower facilities should be provided and located close to chemical stores and chemical injection systems in the chiller hall and potentially the cooling tower basin.

The plant should contain toilet rooms with lockers and showers for operations staff, a workshop, and a spare parts storeroom. Where cooling towers are located on the building roof, a way to remove and download major parts (motor/blades/gear box) should be arranged. In some applications, a larger service elevator might be used or davits.

If the plant site is located adjacent to a development, care should be taken to abide by local environmental regulations related to noise and discharge of chemicals from cooling and CHW systems.

A chiller mechanical room contains extremely costly and hazardous plant components. The plant design should take into consideration the heat radiated from motors, starters, VFDs, transformers, etc., and keep the indoor design conditions below 90°F (32°C) to maintain a tenable environment for the operators. Furthermore, even though many VFDs have internal ventilation fans and air filters, the space should be relatively dust and dirt free to permit adequate cooling of the equipment.

Most DCP chiller halls exceed the refrigerant volume limits by ASHRAE Standard 15 (2016a) and are therefore defined as “machinery rooms.” ASHRAE Standard 15 (2016a) is a safety standard written to project people and property in the event of a refrigerant leak as refrigerants can be toxic and flammable. Because refrigerants are heavier than air, they displace the air in the space at floor level, which is hazardous if maintenance personnel are working on the floor or on their knees. They could suffocate due to lack of air. Hence, machinery rooms have specific design criteria including architectural, general exhaust/ventilation, and refrigerant purge requirements. A refrigerant purge exhaust system should be designed to activate upon detection of refrigerant leakage. Refrigerant monitors must be located adjacent to the chillers to detect any leakage to alarm and activate the refrigerant purge exhaust fan. Refrigerant exhaust openings should be located about 12 in. (0.305 m) above the finish floor as most refrigerants are heavier than air.

According to ASHRAE Standard 15 (2016a), the purge exhaust air rate to purge the machinery room of refrigerant is based on the refrigerant volume of the largest single chiller and should be estimated as follows:

$$Q = 100 \times G^{0.5} \tag{3.7a}$$

where:
$Q$ = airflow, cfm
$G$ = mass of refrigerant in the largest chiller, $lb_m$

$$Q = 70 \times G^{0.5} \tag{3.7b}$$

where:
$Q$ = airflow, L/s
$G$ = mass of refrigerant in the largest chiller, kg

Furthermore, ASHRAE Standard 15 (2016a) stipulates the ventilation rates of a machinery room to be:

- Operated, when occupied, to supply at least 0.5 cfm/ft$^2$ (2.54 L/s·m$^2$) of machine room area or 20 cfm (9.5 L/s) per person.
- Operated, when occupied, at a flow rate sufficient to avoid a max temperature rise of 18°F (10°C) above temperature of inlet air or temperature recommended by installed equipment suppliers.

Many times, the ventilation system and the refrigerant purge can be combined into a single system with separate distinct operating modes since the ventilation requirements typically exceed the refrigerant purge air volume. This conserves on installation cost but requires additional sophistication in controls and sequences. Furthermore, ASHRAE Standard 15 (2016a) requires the following general requirements:

- Adequate clearances for service, maintenance and operation is required and a minimum 7.25 ft (2.2 m) clear headroom below equipment situated over passageways.
- Tight fitting fire doors and no openings that would permit passage of escaping refrigerant.
- At least one door should open to the exterior or through a vestibule to the outdoors.
- Walls, floors, and ceilings shall be tightly sealed and at least one-hour fire resistance.
- Visual and audible alarms shall be activated upon detection of refrigerant.
- Multilevel alarms should be considered with warnings until the threshold is detected to finally start the refrigerant purge mode.
- Open flame combustion or flame producing device is not permitted in the machinery room unless combustion air is sealed and directly ducted from the outside.
- No airflow to or from an occupied space passing through the machinery room.
- Remote control of the mechanical equipment in the machinery room must be provided immediately outside the room to shutdown equipment in the event of an emergency.
- Ventilation fans should be on a separate power circuit and have a control switch immediately outside the machinery room.

If direct-fired absorption chillers are used, the ventilation rate should be sufficient to ventilate the space and maintain the required combustion air. Where direct-fired absorption chillers are used with vapor-compression chillers, the two different types of chillers should be in separate rooms to avoid the exposure of refrigerant vapor to open flame burners in order to comply with ASHRAE Standard 15 (ASHRAE 2016a).

**Table 3.12** Approximate Guide to Plant Floor Area Requirements

| Zone | Area ft$^2$/ton (m$^2$/MW) |
|---|---|
| Chillers (electrical) | 0.75 (0.00000172) |
| Pumps (primary only) | 0.32 (0.00000073) |
| Cooling towers | 0.43 (0.00000098) |
| Electrical | 0.54 (0.00000125) |

The plant space requirement should be adequate to allow for equipment access and proper maintenance. Table 3.12 may be used as a preliminary guide to allocate zone area when conceptually planning the plant.

## ELECTRICAL ROOM DESIGN

Electrical rooms within DCPs need to be properly located to serve the purpose of providing low-cost installation, yet meeting codes and local regulations.

The majority of electrical equipment, including incoming metering room, transformer room, switch gear, and generator rooms are normally located at ground floor for authority access reasons and ventilation equipment. In specific applications, some of the rooms mentioned could be located in floors other than the lower levels. There may be other reasons, such as being flood prone or in areas susceptible to tidal surges, to keep the electrical rooms higher and other non-critical areas lower in the building. Care should be taken to avoid having electrical rooms located directly below water tanks or below drain pipes serving floors above. Chillers are normally selected to operate on medium voltage (3.3 kV or 6.6 kV or 11 kV or 13.8 kV). In some cases of moderate chiller loads, 400 V chiller motors are selected. The 11 kV or the 13.8 kV chillers could operate on direct connection if AHJ permits this. To match with regulations in most countries, soft starters should be used for all medium voltage motors. Solid-state soft starters are common in DCP chiller application.

Motor control centers should be close to the corresponding cooling towers and pumps. Ideally chiller starters, if not unit mounted, should be located in the chiller hall or adjacent rooms such that the shortest electrical conduit routing to chillers is provided. Low-voltage starters are either solid-state soft starters or VS drives as applicable.

## REFERENCES

ASHRAE. 2016a. ANSI/ASHRAE Standard 15-2010, *Safety standard for refrigeration systems*. Atlanta: ASHRAE.

ASHRAE. 2016b. Standard 34-2016. *Safety Standard for refrigeration systems and designation and classification of refrigerants*. Atlanta: ASHRAE.

ASHRAE. 2016c. ANSI/ASHRAE/IES Standard 90.1-2016, *Energy standard for buildings except low-rise residential buildings*. Atlanta: ASHRAE.

ASHRAE. 2016d. Chapter 13, Hydronic heating and cooling. *ASHRAE handbook—HVAC systems and equipment*. Atlanta: ASHRAE.

ASHRAE. 2017. *ASHRAE handbook—Fundamentals*. Atlanta: ASHRAE.

ASTM. 2012. ASTM E84-12c, *Standard test method for surface burning characteristics of building materials*. West Conshohocken, PA: ASTM International.

Doolin, J. 1963. Centrifugal pumps and entrained-air problems. *Chemical Engineering* January 7: 103.

Grunewald, M., with Johnson Controls Inc./York International. Email communication, February 5, 2013.

SPX Cooling Technologies, Inc. 2009. *Cooling tower fundamentals*. Overland Park, Kansas: SPX Cooling Technologies, Inc.

Trane. 2017. Industry update next-generation refrigerants. Trane_HVAC_Industry_Refrigerant_Update_REFR-PRB001B_EN_March2017_.pdf

Weiman, J. Private Communication 11/2017. Chiller flows and pump pressure drops adjusted for various temperature differentials. Trane Company.

Werner. 2017. International review of district heating and cooling. *Energy* 137:617–31.

## BIBLIOGRAPHY

ASHRAE. 2012. ASHRAE Guideline 22-2012, *Instrumentation for monitoring central chilled-water plant efficiency*. Atlanta: ASHRAE.

ASHRAE. 2016. ANSI/ASHRAE Standard 184-2016, *Method of test for field performance of liquid-chilling systems*. Atlanta: ASHRAE.

ASHRAE. 2016b. Chapter 38, Compressors. *ASHRAE handbook—HVAC systems and equipment*. Atlanta: ASHRAE.

ASHRAE. 2017. Self Directed Learning course. Fundamentals of Design & Control of Chilled Water Plants SDL Course (Taylor Engineering).

Daikin. 2014. *Chiller application guide: Fundamentals of water and air cooled chillers*. Technical Report AG 31-003-4. Staunton, VA: Daikin Applied.

DOD. 2004. *Unified facilities criteria: Central heating plants*. UFC 3-430-08N. Washington, DC: US Department of Defense.DOE. 2003. The energy smart guide to campus cost savings. Washington, D.C.: US Department of Energy. www.nrel.gov/docs/fy03osti/34291.pdf.

DOE. 2004. How to buy an energy-efficient water-cooled electric chiller. Washington, DC: Department of Energy. http://infohouse.p2ric.org/ref/05/04186.pdf.

IDEA. 2008. *District cooling best practices guide*. Westborough, MA: International District Energy Association.

Kirsner, W. 1996. The demise of the primary-secondary pumping paradigm for chilled water plant design. *HPAC: Heating/Piping/AirConditioning*.

Kirsner, W. 1998. Designing for 42°F chilled water supply temperature—Does it save energy? *ASHRAE Journal* 40(1):37–42.

Lizardos, E.J. 1995. Engineering primary-secondary chilled water systems. *Engineered Systems* 12(8):30–33.

Schwedler, M. 1998. Chiller/tower interaction: take it to the limit...or just halfway? *ASHRAE Journal* 40(7):32–33.

Schwedler, M., and A. Yates. 2001. *Applications engineering manual: Multiple-chiller-system design and control*. La Crosse, WI: The Trane Company.

Smith, B. 2002. Economic analysis of hybrid chiller plants. *ASHRAE Journal* 44(7):42–45.

Spanswick, I. 2003. Advances in steam. *ASHRAE Journal* 45(9):20–24.

Taylor, S.T. 2002. Primary-only vs. primary-secondary variable flow systems. *ASHRAE Journal* 44(2):25–29.

Taylor, S.T. 2002. Degrading chilled water plant delta-t: causes and mitigation. *ASHRAE Transactions* 108(1):641–53.

Taylor, S.T. 2016. The fundamentals of expansion tanks. *ASHRAE Journal* 58(11):60–66.

Tredinnick, S. 2009. Horizontal or vertical: What's your angle? *District Energy* 4th quarter: 54–55.

Tredinnick, S. 2014. Pipe Sizing: Do you have the need for speed, or are you feeling groovy? *District Energy* 3rd quarter: 63–66.

# 4 Distribution Systems

## INTRODUCTION

While the moderate temperatures involved would suggest that a CHW distribution system could be constructed using the materials and methods used for potable water distribution, it is often not advisable to follow that inclination. A CHW distribution system should function as a closed loop where the water has been treated to minimize its impact on the distribution system, the chiller plant, and the consumer interface or building equipment. Potable water distribution systems routinely have significant rates of leakage, and even when newly constructed, a certain degree of leakage is considered acceptable. With age, potable water distribution systems tend to have increasing rates of leakage. Significant rates of leakage are well documented in the literature (Ghezzi 2005) and the press (Twedt 2002; Sicaras 2007; Long 2008). In potable water distribution, 10% water loss is the industry norm and losses can reach 25% (Twedt 2002). Clearly for a DCS that has water that is significantly more expensive to treat, as well as the investment in energy used to chill the water, such rates of leakage are unacceptable. In areas where feedwater for the DCS is a scarce resource, and thus a significant expense, it becomes even more imperative that the distribution system be of high integrity and leak free. There may also be environmental regulatory requirements that would prohibit significant losses of treated DCS water into the surrounding soil.

Most district cooling distribution systems will be directly buried. The inability to easily inspect the vast majority of the system once it has been constructed is a major drawback to a buried distribution system. In addition, the nearly universal presence of water in soil presents several challenges for a buried distribution system:

- Maintaining a dry environment for the thermal insulation where used
- Providing corrosion protection for all metallic portions of the buried system
- Providing dry environments for appurtenances such as valves, drains, vents, etc.

The joints within the piping system are normally the weak points of the system and this is especially true for insulated distribution systems. It is often difficult to execute a proper field joint of a prefabricated system under normal construction tolerance, practices, and field conditions.

All these factors conspire to make it more difficult to design and construct a buried CHW distribution system when compared to most other buried utilities; however, there are direct-buried systems in existence that have been successfully serving customers for over 40 years.

## DISTRIBUTION SYSTEM TYPES

The combination of aesthetics, first cost, safety, and life-cycle cost naturally divide distribution systems into two distinct categories—aboveground and underground distribution systems. Fortunately, the temperature range in which DCSs operate enables a wide variety of materials to be used, including efficient insulation materials and inexpensive pipe materials that resist corrosion.

The aboveground system typically has the lowest first cost and the lowest life-cycle cost because it can be easily maintained and constructed with readily available materials. Generally, aboveground systems are acceptable where they are hidden from view or can be hidden by landscaping. Poor aesthetics, physical security, right-of-way issues, and the risk of vehicle damage to the aboveground system removes them from contention for many projects, and thus they have seldom been used on DCSs. For more information on the construction of aboveground systems, see Phetteplace et al. (2013).

Several types of systems that are completely field fabricated include the walk-through tunnel, the concrete surface trench, and the deep-burial small tunnel and may be used for insulated district cooling distribution piping; however, due to high cost, they are seldom seen in practice. For more information on these systems and calculating heat gains for them, see Phetteplace et al. (2013).

The most common piping system for buried CHW distribution is preinsulated piping, although uninsulated piping has also been widely used in cooler climates where soil temperatures are low enough to prevent excessive heat gain; the impact of the heat gain on the decision to insulate or not is discussed later in this chapter. Insulated piping is much more costly than uninsulated piping due in part to the cost of its fabrication, but also it is costlier to install in many cases. The higher cost comes from the need to keep the insulation dry in order to preserve its insulating properties, and thus, the need to preserve the integrity of the jacketing material that protects the insulation at the fittings and the field joints in the piping systems. Some approaches do not insulate the field joints or fitting on an otherwise insulated system, but this obviously allows for elevated heat gain at those locations, and where metallic piping is used, due to corrosion there will still be a requirement to protect the piping itself from exposure to groundwater or soil moisture. When using this approach, end seals should be used on each section of preinsulated piping, and these end seals should have passed a pressure test commensurate with the burial conditions for the site.

Underground district cooling distribution systems should be designed for little or no leakage and must account for high pressure and transient shock waves, heat gain impacts, the presence of groundwater, axle loads from surface traffic, differential ground settlement, and the potential for corrosion.

Regardless of the type of construction, it is usually cost effective to route distribution piping through the basements of buildings, but only after liability and right-of-way issues are addressed. As discussed in Chapter 2, in the planning phase, the use of hydraulic analysis to optimize distribution system pipe sizing and to reduce the differential pressure between supply and return will normally be very beneficial. In laying out the main supply and return piping, redundancy of supply and return should be considered. If a looped system is used to provide redundancy, flow rates under all possible failure modes must be addressed when sizing and laying out the piping. Where system layout permits, a reverse return arrangement as discussed in Chapter 3 should be considered.

For larger systems, and in some instances where the piping is insulated, connections to the system are often made in manholes or valve vaults, as discussed in more detail in a section devoted to the topic later in this chapter. For small and moderately sized insulated-buried distribution systems, preinsulated fittings and valves suitable for direct burial may be used. An example of a preinsulated valve suitable for direct burial would be one meeting EN 488 (CEN 2011), as discussed later. Manholes are used to provide access to the distribution system at critical points, such as where there are high or low points on the system profile that vent the trapped air, or where the system can be drained; where there are elevation changes in the distribution system that are needed to maintain the required slope; and where there are major branches with isolation valves. Any appurtenances to a preinsulated system must assure the integrity of the waterproofing jacket. For any buried piping system, preinsulated or otherwise, corrosion of the components exposed to the soil must be considered and the metal piping must be protected, as discussed later in the section on cathodic protection. This complicates direct burial of fixtures such as vents and drains.

Preinsulated piping (Figure 4.1) encloses the insulation in an envelope that will not allow water to contact the insulation. Some systems rely on a watertight field joint between joining-casing sections to extend the envelope to a distant envelope termination point where the casing is sealed to the carrier pipe. Other systems form the waterproof insulation envelope in each individual preinsulated pipe section using the casing and the carrier pipe to form part of the envelope and a waterproof bulkhead to seal the casing to the carrier pipe. The typical insulation is polyurethane foam. It is important to note that the material properties for polyurethane foam can vary widely depending on the formulation, blowing agent, blowing process, and process quality control; not all polyurethane foams are equal nor can that assumption be safely made. It is important that the insulation be fully bonded to the jacket and the carrier pipe to prevent relative movement.

In Europe, a very successful preinsulated piping system used in low-temperature water district heating systems has been developed and standardized. This system is also well suited to insulated district cooling applications. These systems, which are available essentially worldwide, meet EN 253 (CEN 2009a) with regard to all major construction and design details. Standards also have been established for fittings (EN 448 [CEN 2009b]), preinsulated valves (EN 488 [CEN 2011]), and the field joint assemblies (EN 489 [CEN 2009c]).

Another district cooling piping system alternative that was developed for low temperature district heating applications is a flexible piping system. These are presently only

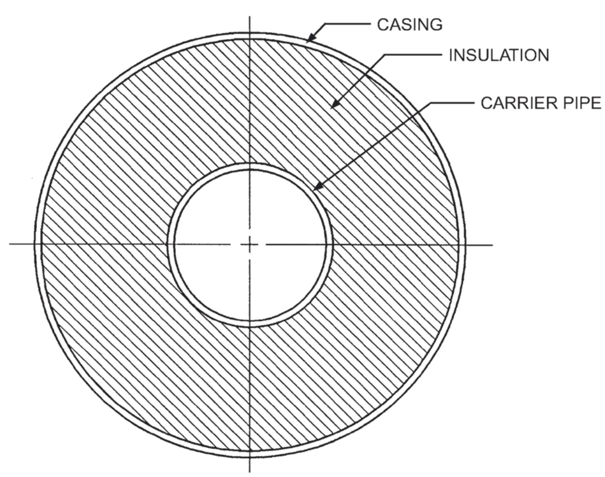



**Figure 4.1** Preinsulated piping system cross section.

available in smaller carrier pipe diameters, up to approximately 4 in. nominal diameter (110 mm) and are used primarily for connections between the main distribution system and the consumer. For these flexible systems, carrier pipe materials that have been used include corrugated steel and stainless steel, thin-walled steel, copper, aluminum, and cross-linked high-density polyethylene (sometimes referred to as PEX). The PEX pipes must be equipped with a diffusion barrier to prevent the diffusion of oxygen into the water. The flexible piping is normally delivered on rolls in lengths up to approximately 330 ft (100 m). The advantage of these flexible piping systems is the ease of installation, resulting primarily from fewer field joints, and to a lesser degree the ability of the system to conform to varying trench configurations. The disadvantages are the higher material cost, and in the case of PEX, the pressure limitations of the material which often require significantly greater wall thicknesses especially in larger diameters.

## PIPING AND JACKETING MATERIALS

A wide variety of materials have been used for district cooling distribution piping systems and jackets on insulated systems including:

- Steel
- Copper
- Ductile iron
- Cement-based products including reinforced concrete and polymer mortar
- Fiberglass-reinforced plastic (FRP), used for jackets as well
- Polyvinylchloride (PVC), used for jackets as well
- Polyethylene (PE) or high-density polyethylene (HDPE), used for jackets as well

Table 4.1 provides a summary of some of the important aspects of the various piping materials, and an approximate relative cost of the most popular materials is shown in Figure 4.2. In addition, the following is a list of the major advantages and disadvantages of each of these materials as well as applicable standards when used for the carrier pipe.

### Steel

*Advantages*: High strength and good flexibility, can be joined by welding for a high-integrity joint that can be inspected for quality control, widely available in all sizes, familiar material to most workforces.

*Disadvantages*: Relatively high cost, highly susceptible to corrosion and will require corrosion protection. Skilled labor force required for welding. Slower installation especially in larger diameters.

*Standards*: ASTM A53/A53M (2012a), ASTM A106A/106M (2011), C200 (2012).

### Copper

*Advantages*: Good flexibility, can be joined by soldering for a high-integrity joint, corrosion resistant but may still require protection, familiar material to most workforces.

*Disadvantages*: Expensive, only available/practical in small diameters (approximately 6 in. [150 mm] and smaller).

*Standards*: ASTM B88 (2009a).

### Ductile Iron

*Advantages*: Reasonable strength and flexibility, available in sizes from 4 in. (100 mm) up to about 64 in. (1.6 m), familiar material to many workforces. Faster installation.

**Table 4.1** Relative Merits of Piping Materials Commonly Used for District Cooling Distribution Systems

| Piping System | Carrier Pipe Joint Integrity | Joint Inspection | Insulated Joints Possible[1] | Corrosion Resistance | Installation Skill Level | Installation Time | Strength Under Burial Conditions | Relative Installed Cost |
|---|---|---|---|---|---|---|---|---|
| Welded Steel | Excellent | NDT (X-ray, etc), Pressure testing | Yes | Low, requires protection | High | High | Excellent | High |
| Soldered Copper | Medium | Pressure testing | Yes | Good | Medium | Medium | Good | Small D = High |
| Ductile Iron | Low | Pressure testing | No | Low, requires protection | Low–Medium | Low | Very Good | Low–Medium |
| Cement Pipe | Low | Pressure testing | No | Excellent | Low–Medium | Low | Good | Low–Medium |
| FRP | Low–Medium | Pressure testing | Yes | Excellent | Medium | Low–Medium | Low | Low–Medium |
| PVC | Low | Pressure testing | No | Excellent | Low–Medium | Low | Low | Low |
| HDPE | High | Pressure testing | Yes | Excellent | Medium | Small D = Low<br>Large D = Medium | Excellent | Small D = Low<br>Large D = Medium–High |

[1] *Insulated joints are not recommended for piping systems that have allowable leakage rates for the joints.*

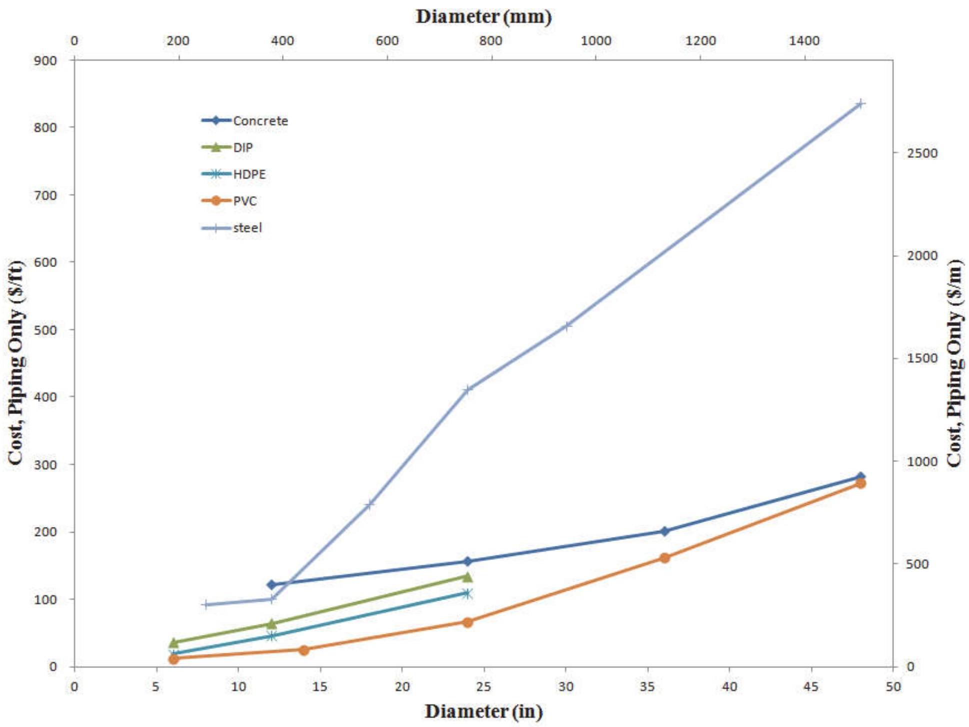



**Figure 4.2** Relative costs for piping alone, uninsulated. Includes joining, but does not include design, supervision and inspection, fittings, excavation, backfill, or surface restoration. *Cost data are from RSMeans-CostWorks® (www.rsmeansonline.com) for first quarter 2018 water utilities.*

*Disadvantages*: Heavy, susceptible to corrosion and will require corrosion protection, can only be joined by mechanical joints, some mechanical joint designs will require thrust blocks at all changes in direction. Fittings are expensive. Allowable leakage per standards.

*Standards*: AWWA C151 (2009).

## Cementitious Pipe

*Advantages*: Reasonable strength, available in all sizes, familiar material to many workforces.

*Disadvantages*: Heavy, poor flexibility, can only be joined by mechanical joints, thrust blocks will be required, difficult to add branch line piping to it. Lower pressure and velocity limits. Allowable leakage per standards.

*Standards*: AWWA C300 (2011a), AWWA C301 (2007a), AWWA C302 (2011b), AWWA C303 (2008a).

## FRP

*Advantages*: Light weight and high strength, available in all sizes.

*Disadvantages*: Poor flexibility, can only be joined by cement/field layup or mechanical joints, difficult to add branch line piping to, cemented joints must be kept clean and dry and may be slow to cure at low ambient temperatures, unfamiliar material to many workforces. Point of leakage may not be obvious.

*Standards*: AWWA C950 (2013), ASTM D2996 (2007) e1.

## PVC

*Advantages*: Light weight, low cost, available in sizes up to 48 in. (1.2 m).

*Disadvantages*: Low strength and poor flexibility, loses strength very quickly at elevated temperatures and becomes brittle at low temperatures, can only be joined by cement or mechanical joints, cemented joints must be kept clean and dry, difficult to add branch line piping to. Water hammer will fracture piping. Requires thrust blocks and has lower velocity limits.

*Standards*: AWWA C900 (2007b), AWWA C905 (2010), ASTM D1785 (2012c), ASTM D2241 (2009b).

## PE and HDPE

*Advantages*: Light weight, very flexible, can be fusion welded for high-integrity joints, available in sizes up to 63 in. (1.6 m). Leak free and fully restrained (no anchor blocks).

*Disadvantages*: Low strength when compared to steel and FRP results in significant wall thickness and thus cost in larger diameters. Additionally, increased wall thickness reduces the inside diameter, which will result in higher pressure losses and may require larger sizes for the same flow rates. Larger diameter fusion welding machines may be of limited availability. Cost will fluctuate with petroleum pricing.

*Standards*: AWWA C901(2008b), AWWA C906 (2007c).

# PIPING SYSTEM CONSIDERATIONS

## Factors to Consider when Choosing Piping Material for a DCS

### Pressure Testing and Acceptable Levels of Water Leakage

Pressure testing of the system will normally be conducted to verify that the system meets industry accepted levels of leakage. The pressure testing procedure will vary with the type of piping material used, for example, for steel-carrier piping the pressure testing procedure is described in Section 137 of ASME B31.1 (2012). The method of joining the carrier piping will largely determine the rates of leakage to be expected both initially as well as over the life of the system. For example, a system with a steel-carrier pipe with inspected-welded joints can be expected to be virtually leak free and remain that way over its useful lifetime. The same can be said for systems using butt-welded PE or HDPE carrier pipes, although the level of inspection methods for joints is not as highly developed as it is for welded-steel piping. Systems with cemented joints, such as is possible with PVC or FRP, can be expected to have higher rates of leakage than a welded system owing to factors like varied environmental conditions as well as the dirt and moisture that are common on construction sites. Finally, systems that use various types of mechanical methods of joining, such as is common in potable water distribution systems, can be expected to have the highest rates of initial leakage and can experience increases in the leakage rate over their lifetime as sealing materials degrade, sealing surfaces become fouled, and joints become further misaligned due to differential settlement.

### Insulated Systems

The addition of insulation to the system also requires that the insulation be protected from water intrusion by a high-integrity jacket. If the system will be insulated, preinsulated piping is recommended. Efforts at field insulating and providing a fully effective waterproofing jacket for the insulation on a construction site have not proven effective. The designer should always consider that the moisture in the soil will naturally migrate toward a CHW pipe when its temperature is lower than the soil temperature, which is normally the case. The choices of readily available preinsulated piping may be much more limited than the range of materials detailed above. The most common and widely available preinsulated piping systems use steel, PVC, HDPE carrier piping, and either HDPE or FRP jackets. PVC is also used as both a carrier pipe and a jacket material for preinsulated pipe. While it is possible for a fabrication shop to preinsulate any type of carrier pipe, unless the process is well established, the quality of the product could be inferior, especially with respect to providing a high-integrity waterproof jacket for the entire system.

### Field Joints

The field joint between successive lengths of piping and fittings is the most critical aspect of the system that must be executed on the construction site. It can be the Achilles heel of an otherwise superior design and construction effort, this cannot be overemphasized. As discussed above, the method of joining the carrier piping and the quality control exercised in the process will largely determine the leakage that can be expected from the system.

The type of method used for joining successive lengths of pipe and fittings will largely be dependent on the type of carrier piping used. Where welding can be used, as is the case for steel and HDPE carrier piping, it is recommended since it will produce the most secure joint. Copper is normally silver soldered, which also provides for a high-integrity joint. For PVC and FRP, cements or solvent cements are often used and are normally recommended over mechanical-type joints that may be available. Cemented/glued joints may not achieve the full tensile strength of the adjacent piping, and thus it may be advisable to use thrust blocks at changes in direction. For ductile iron and cementitious pipes, mechanical joints must be used. These joints are available in a number of designs as described in Nayyar (2000). Some designs provide for mechanical restraint so that the joint cannot be pulled apart by thrust forces, soil settlement/movement, water hammer, etc. Designs without this feature will require thrust blocks at all changes of direction. Mechanical joints have the advantage of rapid assembly and the ability to tolerate angular deflections normally in the 3.5° to 7.0° range (Nayyar 2000). However, despite the presence of sealing surfaces, mechanical joints often will leak and their specification normally includes accepted rates of leakage. The potential for leakage precludes mechanical joints from being insulated in most cases. Furthermore, leaks at mechanical joints provide an additional source of water that may infiltrate the insulation of the adjacent piping sections if the end-seal or jacket fails to perform its intended function. Slip-type mechanical joints, such as o-ring couplings on steel or copper carrier pipes are not recommended due to the potential for fouling of the o-ring/sealing surfaces by either soil from the exterior or debris/scale from the carrier fluid side. Fouling of the o-ring/sealing surfaces is more likely where the coupling is allowed to take up movement caused by piping expansion and contraction or surrounding soil movement.

For insulated systems, the field joint in the piping and the integrity of the water proofing jacket at those locations is very important and is a detail that must not be overlooked or underappreciated. Some methods of insulating the field joint and providing a jacket for that insulation allow the joint to be pressure tested to ensure that the method of extending the waterproof jacket over the joint is leak tight. These types of field joints are essential where installation will be below the water table and are generally preferable to designs that do not allow the joint to be leak tested in any application. In addition, for installation below the water table a leak detection system (discussed below) should be considered mandatory and increased inspection during the construction process is highly recommended. When selecting an insulated piping system, pay particular attention to the details that are provided regarding the field joint, both with regard to materials and instruction for completion. Normally, if requested in the procurement process, the preinsulated piping manufacturer will provide a kit with the necessary materials including both the insulation and jacketing materials; although on larger projects, a special machine may be provided for mixing polyurethane insulation in the proper proportions for field joints. The manufacturer of the system will normally also provide training to the construction crew. Aside from the joining of the carrier piping itself, the seal to the casing of the adjacent piping system is the most critical aspect of the field joint. Many methods are used, including heat-shrink materials and fusion of the jacket by electric heating or welding with filler material in the case of HDPE casing. For FRP casing field layups of fiberglass are common; extra care must be exercised to ensure that the casing of the adjacent sections of pipe are properly prepared, cleaned, and dried to ensure the success of the joint. The cemented joints of FRP casing can be slow to cure in cold ambient temperatures.

Where polyethylene shrink sleeves or shrink wrap are used on field joints it may be advisable to provide a protective sleeve of HDPE or other material over the field joint to protect the shrink sleeve from mechanical abrasion during the backfilling operation and in service. The polyethylene shrink sleeve material is more pliable than a HDPE or FRP jacket material for example and is somewhat more susceptible to mechanical abrasion/impact damage than the casing material. Where polyurethane insulation is used under the shrink sleeve it will provide adequate compressive force to resist any compressive forces from the overburden and support the shrink sleeve material in that regard, the purpose of the protective sleeve is only for mechanical abrasion/impact loading.

Polyethylene shrink sleeves or shrink wrap are available in various grades and thicknesses, be sure to check with the manufacturer of the material to be certain that the best product for your application has been selected. Shrink sleeves sized for each pipe jacket diameter to be used are the most desirable option, however when properly applied shrink wrap can make a high-quality joint approaching the integrity of a shrink sleeve. Proper materials, training and practice as well as control of the conditions of application will yield the high quality joints that are needed for this critical detail of the system construction; compromises to these prerequisites should not be allowed.

### System Size

The ultimate size and expanse of the system will have an impact on the piping material chosen. For the largest systems, where carrier pipe diameters can be 72 in. (1.8 m), only a few of the materials will be available, (e.g., steel and FRP). For larger diameters, unreinforced plastic piping requires a very thick wall to withstand the hoop stress induced by common pressures in DCSs. In addition, there can be issues with the carrier pipe deforming even under its own weight in larger diameters, let alone the pressures of the overburden as is discussed in more detail in the “Geotechnical Considerations” section.

### Material and Labor Availability

Steel is probably the most universally available piping material in either preinsulated or uninsulated pipe. It is also common to find qualified welders who can join steel pipe in any location where a DCS would be under consideration. It will be more difficult, for example, when butt-fusing HDPE piping to find equipment and laborers who are trained in its use. Many of the materials that are used primarily with mechanical joints, such as ductile iron, lend themselves to installation with the minimum amount of specialized labor or equipment. While materials that may use bonded or cemented joints, such as PVC or FRP, do not require specialized equipment or significant workforce training, diligence in cleaning and preparing the joint is required, and additional care and skill may be required under inclement weather conditions such as cold and rain/snow. Shipping costs may also be an issue where heavy piping materials such as ductile iron and cement pipe need to be shipped over long distances.

### Environmental Factors

When selecting a material, the environment where the piping will be installed should be considered. Corrosive soil conditions will be less favorable to metallic products due to cathodic protection requirements, as discussed later in the chapter. Note however, cathodic protection is only required at exposed metal/soil contact and thus fully insulated systems (i.e., those including insulated fitting and field joints) will not normally require cathodic protection. The joining of dissimilar metals must also be approached with caution and may require isolating couplings (Sperko 2009). On sites where it will be difficult to prevent differential settlement, piping systems with lower flexibility such as cement and FRP should be avoided. Painting or coating of CHW piping prior to insulating is recommended for aboveground installations in areas of high humidity and in all belowground installations. The designer should always consider that moisture will naturally migrate toward a CHW pipe when its temperature is lower than the surroundings, which is normally the case. Insulations used today for CHW include polyurethane and polyisocyanurate cellular plastics, phenolics, and fiberglass (aboveground only). With the exception of fiberglass, the rest can form acidic solutions (pH 2–3) once they hydrolyze in the presence of water. The acids emanate from the chlorides, sulfates, and halogens added in the manufacturing process to increase fire retardancy or expand the foam. Phenolics can be more than six times more corrosive than polyurethane due to the acids used in their manufacture and can develop environments to pH 1.8. The easiest way of mitigating corrosion is to coat the pipe exterior with a strong rust-preventative coating (two-part epoxy) prior to insulating. This is good engineering practice and most insulation manufacturers will suggest this, but it may not be in their literature and is an additional requirement that must be specified in the procurement process. For aboveground installations, a high-integrity vapor barrier is also paramount in order to minimize the amount of moisture migrating into the insulation and the pipe surface. Belowground systems will also require a high-integrity vapor barrier formed by a jacket that will also provide protection from the rigors of burial.

### Service and Maintainability

Various materials have differing operations and maintenance requirements. Nonmetallic piping is more difficult to locate than metallic piping, and it is recommended for nonmetallic piping that a locating wire/tape be buried directly over the piping. Some pipe materials may be more difficult to repair or to add branches to at later dates. In general, nonmetallic piping will have lower limits on permitted flow velocity in order to avoid erosion, although it may have lower relative roughness reducing pressure losses.

### Expansion and Contraction Forces and System Transient Stresses

While expansion and contraction are much more limited in DCSs than they are in heat distribution systems, pipe expansion and contraction still must be considered. Anchors will normally be required at entrances to buildings and valve vaults for example. Anchors may be located within a building or manhole. Nonmetallic piping systems will typically have much higher rates of expansion/contraction than metals, although the forces ultimately generated, if restrained, will be lower. For preinsulated systems, particularly those with steel pipes where the expansion/contraction forces can be significant, care must be exercised to assure that the forces generated are properly accounted for so that damage to the insulation and jacket will not occur. As previously noted, one option for insulated district cooling applications is a preinsulated piping system developed and standardized in Europe for use in low-temperature water district heating. With this system, the carrier pipe, insulation, and the casing are bonded together to form a single unit; special precautions are taken in the choice and preparation of the materials to ensure the strength of the bonds. A significant portion of the forces caused by thermal expansion/contraction are passed as shear forces to the mating components and ultimately to the soil. Thus, the system is restrained to a degree by the surrounding soil and hence the amount of expansion/contraction is significantly reduced and all the materials are selected such that the system will not be damaged by these forces. For this type of system to be successful, all the components as well as the processes used in its manufacture must be properly engineered and performance must be verified by testing. If for example, the foam insulation is not strong enough or if it is not adequately bonded to the piping or jacket, relative movement may occur with expansion and contraction. Relative movement may abrade the materials or crush them at changes in direction with ultimate failure of the jacket, and water intrusion into the insulation being one possibility.

DCSs, like any hydraulic system, are subject to flow transients that may cause significant pressure surges, as discussed in the Chapter 2. These can come about due to pump failures or rapid valve closure for example. For these reasons, it may be advisable to conduct transient hydraulic analyses on DCSs to assess the potential for system damage from such events.

### Cost

Ultimately cost will play a large role in the choice of piping material for DCSs. The design should consider the entire life-cycle cost of the piping system and not just the first costs. For example, systems that may offer reduced installation costs but higher rates of leakage may not be the best choice, especially in areas where water is an expensive and precious commodity. Leakage also may be a major concern where the treated DCS water is considered environmentally hazardous or is otherwise regulated. Insulated piping will be significantly more costly than uninsulated piping and its installation will also be slower and more costly. However, the increased heat gains over the life of the system may result in a lower life-cycle cost for the insulated systems; example calculations are provided later in this chapter.

## LEAK DETECTION

The distribution system may require excavation to repair construction errors after burial. Various techniques are available for detecting leaks in district cooling piping. They range from performing periodic pressure tests on the piping system to installing a sensor cable within the insulation along the entire length of the piping to continuously detect and locate leaks. Pressure testing should be performed on all piping to verify integrity during installation and during the life of the piping.

Where leaks cause damage such as sink holes, erosion, flooding of underground structures, etc., it may be possible to use dyes or tracer chemicals within the DCS water if there is a dispute over the source of the offending water. It may also be possible to effectively use the chemicals of the district cooling utility as part of their water treatment program as tracers.

Minor leaks can be very difficult to locate without the aid of a cable-type leak detector as discussed below. Finding a leak typically involves excavating major sections between valve vaults. Infrared detectors and acoustic detectors can help narrow down the location of a leak, but they do not work equally well for all underground systems. Also, they are not as accurate with underground systems as compared to aboveground systems.

For preinsulated piping with foam insulation, special wires or cables can be installed during fabrication to aid in detecting and locating liquid leaks. The wires may be insulated or uninsulated depending on the manufacturer. Some systems monitor the entire wire length while others only monitor at the joints of the piping system. The detectors either look for a short in the circuit using Ohm's Law or monitor for impedance change using time domain reflectometry (TDR). For this type of system to be effective, the system must be dry when the field closures are made and the field closures must be leak tight. Because CHW temperatures are normally lower than the adjacent soil temperature, moisture will be drawn from the surrounding soil towards the CHW system and even in burial situations where a heat distribution system might remain dry, water ingress into the CHW system insulation may occur.

## CATHODIC PROTECTION

If steel or ductile iron piping is directly buried, it must be coated or insulated to reduce the potential for corrosion. For ductile iron piping, PE sleeves are available and provide some degree of corrosion protection, although not to the same degree as a continuous-coated welded-steel piping system. In addition to any coating or protective sleeve systems, it may be necessary or advisable to provide a cathodic protection system for uninsulated steel or ductile iron piping as well. Corrosion is an electrochemical process that occurs when a corrosion cell is formed. A corrosion cell consists of an anode, a cathode, a connecting path between them, and an electrolyte (soil or water). The structure of this cell is the same as a dry-cell battery, and like a battery it produces a direct electrical current. The anode and cathode in the cell may be dissimilar metals, and due to differences in their natural electrical potentials, a current flows from anode to cathode. When the current leaves an anode, it destroys the anode material at that point. The anode and cathode may also be the same material. Differences in composition, environment, temperature, stress, or shape make one section of the same material anodic and an adjacent section cathodic. With a connection path and the presence of an electrolyte, this combination also generates a direct electrical current and causes corrosion at the anodic area. The cathodic protection system generates a reverse voltage strong enough to stop the corrosion cell.

Cathodic protection is a standard method used by the underground pipeline industry to further protect coated steel against corrosion. Cathodic protection systems are routinely designed for a minimum life of 20 years. Cathodic protection may be achieved by the sacrificial anode method or the impressed-current method.

Cathodic protection systems require maintenance. Although the maintenance manual that delineates the required maintenance may be furnished to the owner by the construction contractor, the EOR should be given responsibility for prescribing and reviewing its content. A National Association of Corrosion Engineers (NACE) registered engineer should design the cathodic protection system.

Sacrificial anode systems are normally used with well-coated structures. A direct current is induced to the outer surface of the steel structure with a potential driving force that prevents the current from leaving the steel structure. This potential is created by connecting the steel structure to another metal, such as magnesium, aluminum, or zinc, which becomes the anode and forces the steel structure to be the cathode. The moist soil acts as the electrolyte. These deliberately connected materials become the sacrificial anode and corrode. If the sacrificial anodes generate sufficient current, they adequately protect the coated structure, while their low current output is not apt to corrode other metallic structures in the vicinity.

Impressed-current systems use a rectifier to convert an alternating current power source to usable direct current. The current is distributed to the metallic structure to be protected through relatively inert anodes such as graphite or high-silicon cast iron. The rectifier allows the current to be adjusted over the life of the system. Impressed-current systems, also called rectified systems, are used on long pipelines in existing systems with insufficient coatings, on marine facilities, and on any structure where current requirements are high. Impressed-current systems are installed selectively in congested pipe areas to ensure that other buried metallic structures are not damaged.

The design of effective cathodic protection requires information on the diameter of the carrier pipe, the length of run, the number of pipes in a common trench, and the number of system terminations in access areas, buildings, etc. Soil from the construction area should be analyzed to determine the soil resistivity, or the ease at which current flows through the soil. Areas of low soil resistivity require fewer anodes to generate the required cathodic protection current, but the life of the system depends on the weight of anode material used. The design life expectancy of the cathodic protection must also be defined. All anode material is theoretically used up at the end of the cathodic protection system life. At this point, the corrosion cell reverts to the unprotected system and corrosion occurs at points along the system or buried metallic structure. Anodes may be replaced or added periodically to continue the cathodic protection and increase the conduit life.

A cathodically protected system must be electrically isolated at all points where the pipe is connected to building or access (manhole) piping and where a new system is connected to an existing system. Piping is generally tied to another building or access piping with flanged connections. Flange isolation kits, including dielectric gaskets, washers, and bolt sleeves, electrically isolate the cathodically protected structure. If an isolation flange is not used, any connecting piping or metallic structure will be in the protection system, *but protection may not be adequate*. Isolation flanges are extremely important when connecting to existing piping systems because new steel is slightly anodic when compared to older piping because the old steel has lost some of its electrons already.

The effectiveness of cathodic protection can only be determined by an installation survey after the system has been energized. Cathodically protected structures should be tested at regular intervals to determine the continued effectiveness and life expectancy of the system. Sacrificial anode cathodic protection is monitored by measuring the potential (voltage) between the underground metallic structure and the soil versus a stable reference. This potential is measured with a high resistance voltmeter and a reference cell. The most commonly used reference cell material is copper/copper sulfate. One criterion for protection of buried steel structures is a negative voltage of at least 0.85 V as measured between the structure surface and a saturated copper/copper sulfate reference electrode in contact with the electrolyte (the soil). Impressed-current systems require more frequent and detailed monitoring than sacrificial anode systems. The rectified current and potential output and operation must be verified and recorded at monthly intervals. NACE SPO169 (NACE 2007) has further information on the control of external corrosion on buried, metallic structures.

## GEOTECHNICAL CONSIDERATIONS

There are stringent burial requirements for underground district cooling distribution systems, as compared to most other utilities associated with buildings. The elevation of the trench bottom must not have slope reversals between valve vaults and building entry locations. These piping systems normally have the coatings or jackets needed for corrosion protection or insulation protection that must not be damaged by rocks, debris, or construction equipment. Thus, proper burial conditions must be established for the district cooling distribution system to achieve its design life. Requirements vary and manufacturers of the piping system should be able to provide guidance specific to their system. It is the responsibility of the EOR to assure that these requirements are included in the contract documents. It is recommended that the services of a licensed geotechnical engineer familiar with the local conditions be engaged to conduct a site survey in advance of construction, to recommend any soil testing required, and to develop the specifications for excavation and backfill. The geotechnical and structural engineer should also be responsible for the design of any thrust blocks or anchors that are needed based on forces provided to them by the EOR.

External loads will be imposed on the system both by the overburden (dead loads) and the traffic on the surface above (live loads). Minimum cover will normally be specified by the piping system supplier in order for the system to be able to survive live loads from traffic and activities on the surface.

It is beyond the scope of this guide to provide a complete treatment of the geotechnical design procedure needed to assure live and dead loads placed on the DC piping system externally do not damage the system. However, Figure 4.3 shows the impact that burial depth has on live loads at the surface to provide some perspective. From studying Figure 4.3, it is clear how burial depth reduces the impacts of live loads dramatically.

The dead load of the overburden soil for wide trenches or pipes buried on embankments can be estimated conservatively by the weight of the "prism" of soil directly over the pipe, i.e. a volume of soil per unit length of pipe equal to the burial depth multiplied by the pipe diameter (ASWP undated). This is a very conservative assumption as it ignores the effect of arching in the soil and essentially treats the soil as if it were a liquid. Detailed calculations using the soil properties can be made to determine the arching or

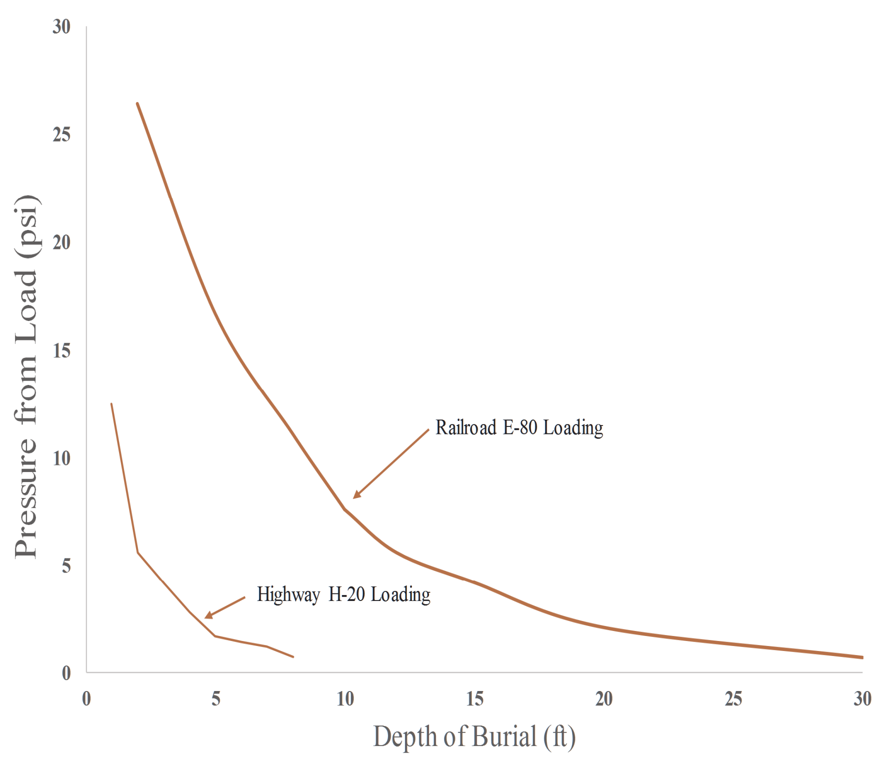



**Figure 4.3** The impact of burial depth on live loads from the surface.
*Data from ASWP (undated)*

bridging effect. The soil will also act as a support for the sides of the buried pipeline and in fact this effect may be used to reinforce the pipe against the burden of loads, see ASWP (undated) for details.

For piping insulated with continuous polyurethane foam (i.e. either foamed in place between the jacket and carrier pipe or spray applied directly to the carrier pipe before the jacket is applied) the pipe will be stiffened significantly. For piping systems with liners or outer coating such as the insulation, ASWP (undated) suggests adding the stiffness of the components together to yield the assembly stiffness.

For additional information on design of buried pipelines see either ASWP (undated) or Revie (2015).

In general, trenches must be overexcavated by a minimum of 4 in. (100 mm) in depth to remove any unyielding material; overexcavation may need to be greater at the locations of the field joints depending on the type of system being installed and the method of construction. The overexcavation is generally filled with a select backfill material; normally this would be a sandy, noncohesive material free of any stones greater than 0.75 in. (19 mm). If unstable materials are encountered in the excavation, those materials should be removed and properly backfilled and compacted. The select backfill in the trench bottom should be prepared to achieve the minimum slope for the carrier pipe of 1 in. in 20 ft (0.2%) (1 cm in 2.4 m [0.4%]) and compacted to 95% of laboratory maximum density per ASTM D 698 (2012b). Some of the methods of carrier pipe joining, such as welding, will require a working area around the entire circumference of the field joint. One method of achieving this is to overexcavate under the pipe, and potentially even at the sides of the trench at the locations of the field joints. If this is done, care must be taken to fully compact the backfill material under the field joint area. Another method used to provide working clearance for making the field joint is to block the piping up off the bottom of the trench during that process. When this method is used, care must be taken to block the pipe sufficiently to achieve proper alignment for joining and to emulate the pipe, as it will ultimately lay on the sloped trench bottom. Once the field joints have been completed, the blocking should be removed and the piping carefully and uniformly placed in the trench bottom that has been prepared as described above. The blocking should not be left in place as this creates point loading on the piping and may contribute to differential settlement as well. In some situations when welded-steel piping is being used, for example, it may be possible to join two sections of piping together adjacent to the trench and then lift the assembly in as a unit and thus reduce the work required in the trench.

After the piping is placed in the trench and all field joints and pressure tests have been completed, immediately preceding the backfilling, the elevation of the top invert of the pipe/jacket should be taken at each pipe section midpoint and field joint. These elevations should be recorded and subsequently transferred to the as-built drawings. Backfill of the piping should then be accomplished in layers of no more than 6 in. (150 mm) with the same select backfill material used for the pipe bedding. The select backfill should be extended to approximately 12 in. (300 mm) above the top of the pipe or jacketing. Buried utility warning tape should be buried in the trench at this depth. Compaction of this backfill material should also be to 95% of laboratory maximum density per ASTM D 698 (2012b). Care should be exercised to ensure that the backfill adequately fills the void created under the pipe and between the supply and return pipes. Care should also be exercised not to damage any pipe coating or insulation jacketing material and if any such damage does occur, it should be repaired per the pipe system manufacturer's field repair instructions. Final backfill of the remainder of the trench should be accomplished using the native soil, but removing any stones greater than 3 in. (75 mm), compacted in layers of no more than 6 in. (150 mm). This final backfill should be compacted to 95% of laboratory maximum density per ASTM D 698 (2012b) for noncohesive soils, or 90% of laboratory maximum density per ASTM D 698 (2012b) for cohesive soils. Note that it is not advisable to complete the final backfilling process with anything other than native soil as its permeability may be much different than the native material. For example, using a permeable backfill material in a native soil that is impermeable is essentially placing your DCS in a drainage ditch for surface water.

It should also be noted that horizontal boring, jacking, and microtunneling have become popular methods of installing buried pipelines where the normal cut and cover methods described above may be difficult or impossible, or simply cost prohibitive for one or more reasons. These alternate burial methods preclude the use of protective backfilling approaches discussed above, but in many cases may also negate such requirements. However, protection of the pipeline from corrosion will still be necessary for metallic pipelines and methods appropriate for the installation method must be used.

## VALVE VAULTS AND ENTRY PITS

Manholes or valve vaults may be required on underground distribution systems to provide access to underground systems at critical points, such as where there are high or low points on the system profile that vent trapped air or where the system can be drained; where there are elevation changes in the distribution system that are needed to maintain the required constant or nonreversing slope (a slope of 1:240 is a reasonable construction value that can be achieved); and where there are major branches with isolation valves.

A decision to use manholes or valve vaults in a system should not be taken lightly; while these features provide critical access as noted above, they come with an enormous maintenance burden in many climates/soil conditions and they are normally expensive to construct. Especially for smaller diameter piping, preinsulated valves and other appurtenances designed for direct burial will normally be lower both in initial costs as well as maintenance costs.

Preinsulated valves and fitting for direct burial are discussed earlier in this chapter under the “Distribution System Types” section.

To facilitate leak location and repair and limit damage caused by leaks, access points generally should be spaced no farther than 500 ft (150 m) apart; the manholes provide the only access to the buried system for testing such as leak location, isolation for emergency repairs, etc.

These recommendations are generalizations and the specific system used as well as the site-specific economics may dictate greater, or even lesser, spacing.

Special attention must be given to the safety of personnel who come in contact with distributions systems or who must enter spaces occupied by underground systems. The regulatory authority’s definition of a confined space and the possibility of exposure to high-pressure piping have a significant impact on the access design, which must be addressed by the project designer or EOR.

Entry permits will normally require not only the equipment, training, and personnel needed to affect a rescue should a problem occur, but also monitoring both before and during entry for safe oxygen levels and the absence of explosive gases.

As noted, valve vaults are often classified as a confined space, and for those entering may require training, obtaining entry permits, and following entry procedures. For additional information on Occupational Safety and Health Administration (OSHA) regulations and requirements, refer to Chapter 8.

Valve vaults should have electric sump pumps with adequate lighting and convenience outlets to inspect the system components located therein. Electric sump pumps may not be required if the manholes can be drained to a sanitary system. French drains are usually not acceptable because groundwater will backflow into the manhole when high groundwater levels occur.

Valve vaults allow a user to isolate one segment of a system rather than analyze the entire system or a large section thereof. Isolation may be for both the purpose of routine maintenance as well as operational problems or failures. The term vault is used to eliminate confusion with sanitary manholes. Valve vaults are important when the underground distribution system cannot be maintained without excavation, as is the case for all direct-buried district cooling piping systems. Valve vaults allow for step elevation change in the distribution system piping, while maintaining an acceptable slope on the system; they also allow the designer to better match the topography and avoid unreasonable and expensive burial depths. For all their positive attributes, valve vaults also have a significant potential to generate maintenance requirements of their own as well as to provide the potential for problems, and if left uncorrected, allow the problems to cascade into the adjacent buried portions of the distribution system.

## Valve Vault Issues

Valve vaults can be considered a necessary evil; while they provide the advantages discussed above, they bring many challenges in design, construction, and operation in order to successfully provide those advantages. The various aspects of design of the valve vault are detailed below.

### Penetrations

One of the basic functions of a valve vault is to provide a dry, corrosion-free environment for the piping and appurtenances that are located in the valve vault. This means that the vault walls must not let groundwater enter the vault. Typical types of penetrations of the vault walls are the district cooling piping, electrical service conduits, sump pump discharge pipes, and sanitary drains. All penetrations must have a method to provide a positive water seal between the vault wall and the pipes other utilities such as electrical conduits. Often a leak plate is welded to a steel sleeve that is cast into the concrete vault wall (see Figure 4.4). Existing vault walls or precast valve vaults that do not have the leak plate cast into the wall may still be sealed acceptably, as long as the surface of the wall penetration is smooth and an adequate seal can be achieved. The annular space between

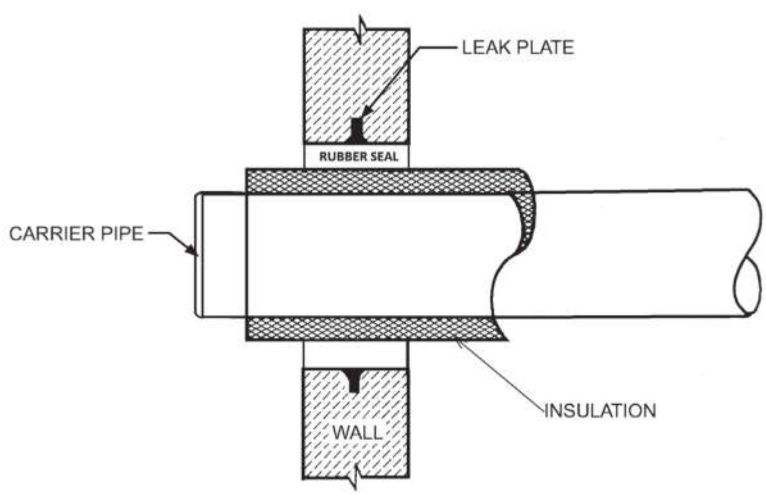



**Figure 4.4** Manhole penetration detail with leak plate and adjustable rubber seal shown.

the outside of the piping/conduit and the valve vault wall penetration is typically sealed with a link seal or other type of adjustable compressed rubber seal, see Figure 4.5. These rubber seals will work poorly where construction quality control is poor as the holes in the vault wall are often the wrong diameter or are irregular. Piping that penetrates the vault wall at an angle causes especially difficult sealing problems for precast construction when the hole in the vault wall is not perpendicular to the conduit. In addition, in the case of some nonmetallic systems, the casing of the district cooling pipe tends to deflect when the seal exerts the radial compressive load it needs to achieve the water seal. When plastic piping or casings are used, a special design is needed to prevent radial deflection, or some other type of sealing method is required.

## Ponding Water

The most significant problem with valve vaults is that water ponds in them. Ponding water may be from either carrier pipe leaks or the intrusion of surface or groundwater into the valve vault. When the hot and CHW distribution systems share the same valve vault, plastic CHW lines often fail because ponded water heated by the hot water or steam distribution system piping heats the plastic to failure. Historical experience indicates that water gathers in the valve vaults irrespective of climate; therefore, a sound design strives to eliminate the water for the entire life of the underground distribution system. Where possible, the most successful water removal systems are those that drain to sanitary or storm drainage systems; this technique is successful because the system is affected very little by corrosion and has no moving parts to fail. Backwater valves are recommended in case the drainage system backs up.

Duplex sump pumps with lead-lag controllers and a failure annunciation system are used when storm drains and sanitary drains are not accessible. Because pumps have a history of frequent failures, duplex pumps help eliminate short cycling and provide standby pumping capacity. A labeled, lockable, dedicated electrical service should be used for electric pumps. The circuit label should indicate what the circuit is used for; it should also warn of the damage that will occur if the circuit is deenergized. Alarms that indicate failure of the sump pump or its circuitry are desirable. Where the application permits, these can simply be warning lights aboveground at the manhole location. Alternately, a more desirable solution is where the manholes can be connected to a SCADA (supervisory con-

**Figure 4.5** Rubber seal between outer pipe casing and steel sleeve through conduit wall. Bolts are tightened to compress seal in annulus.

trol and data acquisition) system (see Chapter 7). In this case, alarms can be displayed at a central location, which is normally the central plant control room.

Electrical components have experienced accelerated corrosion in the high humidity of closed, unventilated vaults. A pump that works well under normal conditions often performs poorly or not at all when subjected to an environment of 100% relative humidity. To resolve this problem, one approach specifies components that have demonstrated high reliability at 100% relative humidity with a damp-proof electrical service. The pump should have a corrosion-resistant shaft (when immersed in water), an impeller, and have demonstrated 200,000 cycles of successful operation, including the electrical switching components, at 100% relative humidity. The pump must also pass foreign matter; therefore, the requirement to pass a 0.375 in. (1 cm) ball should be specified. The sump pump intake should be screened to prevent the entry of foreign matter that could prevent the pump from working as intended.

Redundant methods may be necessary if maximum reliability is needed or future maintenance is questionable. The pump can discharge to the sanitary or storm drain or to a splash block near the valve vault; local codes should be checked before discharge to storm drains or sewers is planned. Water pumped to a splash block has a tendency to enter the vault, but this is not a significant problem if the vault construction joints have been sealed properly. Extreme caution must be exercised if the bottom of the valve vault has French drains. These drains work backward when the groundwater level is high and allow groundwater to enter the vault and flood the insulation on the distribution system during a high groundwater condition. Adequate ventilation of the valve vault is also important as it will help reduce humidity, as discussed below in the “High Humidity” subsection.

### Crowding of Components

The valve vault must be laid out in three dimensions, considering the space needed for the standing room for the worker, wrench swings, the size of valve operators, variation between manufacturers in the size of appurtenances, and all other variations that the specifications allow with respect to any item placed in the vault. Access to the vault must be adequate to comply with confined space entry requirements where they exist. To achieve desired results, the vault layout must be shown to scale on the contract drawings.

### High Humidity

High humidity develops in a valve vault when it has no positive ventilation. High humidity is particularly problematic in CHW distribution systems as pipe surfaces are almost always below dew-point temperatures, and thus condensation will occur on uninsulated piping and fixtures or under insulation if it is not fully sealed. Open structural grate tops are the most successful covers for ventilation purposes and safety (see Figure 4.6). Open grates allow debris and rain to enter the vault; however, the techniques mentioned in the section on ponding water are sufficient to handle the rainwater. Some vaults have a closed top and use screened, elevated sides to allow free ventilation (see Figure 4.7). In this design, the solid vault sides extend slightly above grade, then a screened window is placed in the wall on at least two sides. The overall above-grade height may be only 18 in. (45 cm).

Both of the manhole cover designs shown in Figures 4.6 and 4.7 have the advantage of being able to remove the entire top of the manhole for access to components for maintenance and service work. Open grates or other designs with high ventilation rates should not be used in environments where freezing is possible.

They may also be problematic where blowing sand is common. However, their ability to provide a well ventilated, low humidity environment may offset this disadvantage if periodic cleaning of the vault is adopted as a policy. Cleaning should be included in peri-

**Figure 4.6** Manhole with an open grate cover.

**Figure 4.7** Manhole with a raised cover and vented sides.

odic inspection of valve vaults, a practice which is recommended regardless of valve vault type and construction. An appropriate schedule for inspections will depend on many factors such as climate, system age, operating conditions, etc., and should be established based on experience at the site in question.

### Deep Burial

When a valve vault is buried too deeply, the structure is exposed to potentially greater groundwater pressures, entry and exit often become a safety problem, construction becomes more difficult, and the cost of the vault is greatly increased. The most common way to limit burial depth is to place the valve vaults closer together. Steps in the distribution system slope are made in the valve vault (i.e., the carrier pipes come into the valve vault at one elevation and leave at a different elevation). If the slope of the distribution system is changed to more closely match the earth's topography, the valve vaults will be shallower; however, the allowable range of slope of the carrier pipes restricts this method. In most systems, the slope of the distribution system can be reversed in a valve vault, but not out in the system between valve vaults without other provisions for vents and drains for example. The minimum slope for the carrier pipes is 1 in. in 20 ft (1 cm in 2.4 m [0.4%]). Lower slopes are outside the range of normal construction tolerance. If the entire distribution system is buried too deeply, the designer must determine the maximum allowable burial depth of the system and survey the topography of the distribution system to determine where the maximum and minimum depth of burial will occur. All elevations must be adjusted to limit the minimum and maximum allowable burial depths.

### Freezing Conditions

In cold climates, the failure of distribution systems due to water freezing in the components is common. The designer must consider the coldest temperature that may occur at a site and not the 99% or 99.6% condition used in building design (ASHRAE 2017). Insulation should be on all items that can freeze, and it must be kept in good condition. Electrical heat tape and pipe-type heat tracing can be used under insulation. In extremely cold climates during the winter, the water may have to be circulated (possibly with heat addition) or drained. Circulation may be provided by the use of shunts or bypasses around the control valves at the location of the building interconnect.

### Safety and Access

Owing to high pressure, in addition to the potential for drowning, the water used in district cooling underground distribution systems can cause severe injury and/or death if accidentally released in a confined space such as a valve vault. The shallow valve vault with large openings is desirable because it allows personnel to escape quickly in an emergency. The layout of the pipes and appurtenances must allow easy access for maintenance without requiring maintenance personnel to crawl underneath or between other pipes. The task of the designer is to keep clear work spaces for maintenance personnel so that they can work efficiently and, if necessary, exit quickly. Engineering drawings must show pipe insulation thickness to scale, otherwise they will give a false impression of the available space.

The location and type of ladder are important for safety and ease of egress. It is best to lay out the ladder and access openings when laying out the valve vault pipes and appurtenances as a method of exercising control over safety and ease of access. Ladder steps, when cast in the concrete vault walls, may corrode if not constructed of the correct material. Corrosion is most common in steel rungs, or plastic rungs that have steel reinforcing at their core. Either cast iron or prefabricated, OSHA-approved, galvanized steel ladders that sit on the valve vault floor and are anchored near the top to hold it into position are best. See Figure 4.8 for an example. If the valve vault design uses lockable access doors, the locks must be operable from inside or have some keyed-open device that allows workers to keep the key while working in the valve vault.

### Valve Vault Construction

The most successful valve vaults are those constructed of cast-in-place reinforced concrete. These vaults conform to the earth excavation profile and show little movement when backfilled properly. Leakproof connections can be made with pipe or insulated pipe jackets, even though they may enter or leave at oblique angles. In contrast, prefabricated valve vaults often settle and move after construction is complete. Penetrations for prefabricated vaults, as well as the angles of entry and exit, are difficult to locate exactly in advance. As a result, much of the work associated with penetrations is not detailed and must be done by construction workers in the field, which can greatly lower the quality, in turn greatly increasing the chances of a groundwater leak into the valve vault. Both cast-in-place and prefabricated valve vaults can use concrete additives and coatings that are used for water proofing and sealing the concrete to mitigate permeability and water leakage into the vault through the concrete structure. However, the penetrations will normally be the most vulnerable feature with respect to waterproofing and will warrant extra care in both design and construction, refer to the earlier discussion the in the "Penetrations" subsection.

### Construction Deficiencies

Construction deficiencies that go unnoticed in the buildings can destroy a cooling distribution system; therefore, the designer must clearly convey to the contractor that a valve vault does not behave like a sanitary manhole. A design that is sufficient for a sanitary manhole will cause the appurtenances in district cooling distribution system manhole to fail prematurely because many of the requirements mentioned above are not provided.

### Construction of Systems without Valve Vaults

Uninsulated systems have been designed where features such as valves are directly buried and then remotely operated, much the same as is frequently done for potable water distribution systems. In order for this approach to be successful for an insulated system, a continuous, high-integrity jacket system as well as the necessary seals for operable shafts,

**Figure 4.8** Steel manhole ladder, hot dip galvanized after fabrication.

etc., must be provided. If the waterproofing is not entirely successful, groundwater will enter the system causing corrosion and deterioration of the insulation thermal properties. Achieving adequate waterproofing of features, such as direct burial valves and vents/drains, will be very difficult to achieve in the field. Factory prefabricated systems that use this approach are the standardized European preinsulated piping systems developed originally for low-temperature, hot-water systems, as discussed earlier.

In using these systems it should be considered that the operating conditions for CHW distributions systems are more severe in terms of the drive for moisture ingress due to their operating temperatures often being below even surrounding soil temperatures. Additionally, preinsulated valves, vents, etc., are usually limited to smaller diameter pipes and may not be available in the size needed for a typical DCS in the Middle East for example.

## THERMAL DESIGN CONDITIONS

The thermal design conditions for a CHW distribution system may vary greatly from one system to another, and thus the decision to insulate and to what degree should not be based on prior experience, except for systems located in similar climates with similar load characteristics, similar load densities, and similar CHW generation costs. Special circumstances, such as an isolated building with a long lateral piping run uncharacteristic of the remainder of the system, may often require specific provisions with regard to the thermal design.

Another special circumstance that must be considered in the design is the startup of a large system where the initial loads may be small. Heat gains in this case may represent a large portion of the load, and due to low demand, the consumers' flow rates will be low, therefore supply-temperature degradation may be unacceptable.

For a CHW distribution system, two thermal design conditions must be considered to ensure satisfactory system performance (example calculations are provided later):

1. The normal condition used for the life-cycle cost analysis determines appropriate insulation thickness. Average values for the temperatures, burial depth, and thermal properties of the materials are used for design. If the thermal properties of the insulating material are expected to degrade over the useful life of the system, appropriate allowances should be made in the cost analysis. Polyurethane foam's thermal conductivity will increase over time as the blowing agent diffuses through the plastic jacket and is replaced by air. This aging can be reduced by systems that utilize an aluminum diffusion barrier between the polyurethane foam and jacket.
2. Because heat transfer in piping is not related to the instantaneous connected load, it can be a large part of the total load at times of low load. The maximum heat transfer rate determines the load on the central plant due to the distribution system. It also determines the temperature increase in the CHW distribution system, which determines the delivered temperature to the consumer. For this calculation, the thermal conductivity of each component must be taken at its maximum value, and the temperatures must be assumed to take on their extreme values, which would result in the greatest temperature difference between the CHW and the soil or air. The burial depth will normally be at its lowest value for this calculation.

Uncertainty in heat transfer calculations for CHW distribution systems results from the uncertainty in the thermal properties of the materials involved as well as approximations that must be made in the calculation procedure. Generally, the designer must rely on the manufacturers' specifications and handbook data to obtain approximate values. The data in this chapter should only be used as guidance in preliminary calculations until specific products have been identified, then specific data should be obtained from the manufacturer of the product in question. In the case of soils, as much information as is practical should be gathered to determine the thermal properties. Uninsulated systems will be more sensitive to soil properties in heat gain calculations than will systems with insulation.

## SOIL THERMAL PROPERTIES

Heat transfer in buried systems is determined largely by the thermal conductivity of the soil and by the depth of burial for uninsulated systems. For insulated systems, if the thermal insulation of the system has low thermal resistance, the soil thermal conductivity and burial depth will also have a significant impact on heat transfer. Soil thermal properties are principally a function of three factors:

1. The type of soil (grain size and composition),
2. The moisture content of the soil, and
3. The density (state of compaction) of the soil.

### Soil Thermal Conductivity

In the absence of specific information on the soil type, moisture content, and density, the thermal conductivity factors in Table 4.2 may be used as an estimate. Because dry soil is rare in most areas, low moisture content should be assumed only where it can be validated for calculation of heat gains in the normal operational condition. Because moisture will migrate

**Table 4.2** Soil Thermal Conductivities

| Soil Moisture Content (By Mass) | Thermal Conductivity, Btu/h·ft·°F (W/m·K) | | |
|---|---|---|---|
| | Sand | Silt | Clay |
| Low, <4% | 0.17 (0.29) | 0.08 (0.14) | 0.08 (0.14) |
| Medium, 4%–20% | 1.08 (1.87) | 0.75 (1.30) | 0.58 (1.00) |
| High, >20% | 1.25 (2.16) | 1.25 (2.16) | 1.25 (2.16) |

toward a chilled pipe, a thermal conductivity value of 1.25 Btu/h·ft·°F (2.16 W/m·K) is recommended for CHW systems in the absence of any site-specific soil data.

Ideally, when confronted with an engineering problem requiring soil thermal properties, one would obtain samples from the site and have them tested. For most applications, this is much too expensive, although in-situ thermal properties testing has become very popular for the design of ground-source heat pump systems (see Kavanaugh 2000). If an analysis of the soil is available or can be done, the thermal conductivity of the soil can be estimated from published data for soils with similar composition and gradation (Kersten 1949; Farouki 1981; Lunardini 1981). For steady-state analyses, only the thermal conductivity of the soil is required. If a transient analysis is required, the specific heat and density are also required. If neither laboratory tests for thermal properties or soil composition and gradation are available, the approximate equations developed by Kersten (1949) may be used. Kersten (1949) presents one set of equations for fine-grained soils (silts and clays) and another set for coarse-grained soils (sands and gravels). These equations are presented below. These equations apply to moisture contents of 7% or more for silts and clays and 1% or more for sands and gravels. Soils with more than 50% silt and clay would fall into the fine-grained group. For sandy soil with relatively high silt and clay content (i.e., 40%), Kersten (1949) suggests that the average of the fine-grained and coarse-grained equations would provide a reasonable prediction. If applied judiciously, Kersten (1949) suggests that the resulting thermal conductivity predictions from Equations 4.1 through 4.4 should be within ±25%.

For I-P units:

$$k_s = 0.083(0.9 \log w - 0.2)10^{0.01\gamma_d} \quad \text{unfrozen silts-clay} \tag{4.1a}$$

$$k_s = 0.00083(10)^{0.022\gamma_d} + 0.0071(10)^{0.008\gamma_d}w \quad \text{frozen silts-clay} \tag{4.2a}$$

$$k_s = 0.083(0.7 \log w + 0.4)10^{0.01\gamma_d} \quad \text{unfrozen sand-gravel} \tag{4.3a}$$

$$k_s = 0.0063(10)^{0.013\gamma_d} + 0.0027(10)^{0.0146\gamma_d}w \quad \text{frozen sand-gravel} \tag{4.4a}$$

where

$k_s$ = thermal conductivity, Btu/h·ft·°F
$\gamma_d$ = dry density of soil, $lb_m$/ft
$w$ = moisture content of soil,% (dry basis)

For SI units (Farouki 1981):

$$k_s = 0.1442(0.9 \log w - 0.2)10^{0.6243\gamma_d} \quad \text{unfrozen silts-clay} \tag{4.1b}$$

$$k_s = 0.001442(10)^{1.373\gamma_d} + 0.01226(10)^{0.4994\gamma_d}w \qquad \text{frozen silts-clay} \tag{4.2b}$$

$$k_s = 0.1442(0.7 \log w + 0.4)10^{0.6243\gamma_d} \qquad \text{unfrozen sand-gravel} \tag{4.3b}$$

$$k_s = 0.01096(10)^{0.8116\gamma_d} + 0.00461(10)^{0.9115\gamma_d}w \qquad \text{frozen sand-gravel} \tag{4.4b}$$

where

$k_s$ = thermal conductivity, W/m·°C
$\gamma_d$ = dry density of soil, g/cm$^3$
$w$ = moisture content of soil,% (dry basis)

Note that in Equations 4.1a through 4.4a and Equations 4.1b through 4.4b the moisture content of the soil is expressed as a percentage of the dry weight of the soil, which is the convention that is normally used.

## Temperature Effects on Soil Thermal Conductivity and Frost Depth

It should also be noted that the thermal conductivity of soils, like most materials, is a function of temperature. Kersten (1949) provides data for the soils he studied; however, normally for moderate temperature ranges near normal ambient temperature it is not necessary to compensate for this effect. What must not be ignored is the dramatic increase in thermal conductivity between unfrozen and frozen soils when there is significant moisture present. Frozen soils may be encountered in the immediate vicinity of buried district cooling pipelines in areas of seasonal frost and permafrost. In such climates, consideration must be given to freeze protection for CHW piping that is not in operation during the periods when frost penetration is expected to reach the burial depth of the pipelines. The calculation of frost depth is beyond the scope of this guide; the reader is referred to Lunardini (1981), ASCE (1996), or Andersland and Ladanyi (2004). Furthermore, the reader is cautioned against using overly generalized frost depth maps; in addition to the large microclimatic variations that such maps fail to capture, the surface type, soil type, and soil moisture content can have substantial impacts on frost depth.

## Specific Heat of Soils

The specific heat of soils with moisture within them is a function of the soil solids and water content and the specific heat of those two components in the soil/water system. The following equation is a commonly used, simple proration of these impacts:

$$c_s = c_{ss} + c_w\left(\frac{w}{100}\right) \tag{4.5}$$

where

$c_s$ = specific heat of soil,% (dry basis)
$c_{ss}$ = specific heat of dry-soil solids, Btu/lb$_m$·°F (kJ/kg·K)
$c_w$ = specific heat of water, Btu/lb$_m$·°F (kJ/kg·K)
$w$ = moisture content of soil,% (dry basis)

Note that in Equation 4.5 the moisture content of the soil is expressed as a percentage of the dry weight of the soil, which is the convention normally used. The specific heat of liquid water, $c_w$ can be taken as 1 Btu/lbm·°F (4.18 kJ/kg·K) for our purposes. Because the specific heat of dry soil is nearly constant for all types of soil, $c_{ss}$ may be taken as 0.175 Btu/lbm·°F (0.73 kJ/kg·K).

**Example 4.1: Soil Thermal Property Calculations**

As an example of the use of the soil thermal conductivity equations, consider the case of an unfrozen sandy soil with a moisture content of 10% at a dry density of 100 $lb_m/ft^3$ (1.60 $g/cm^3$).

Working the problem in I-P units using Equation 4.3a:

$$k_s = 0.083(0.7 \log w + 0.4)10^{0.01\gamma_d} = 0.083[0.7 \log(10) + 0.4]10^{0.01(100)}$$
$$= 0.91 \text{ Btu/h·ft·°F}$$

Repeating this calculation using metric units with Equation 4.3b:

$$k_s = 0.1442[0.7 \log w + 0.4]10^{0.6243\gamma_d}$$
$$= 0.1442[0.7 \log(10) + 0.4]10^{0.6243(1.60)} = 1.58 \text{ W/m·K}$$

Calculating the specific heat for this same soil using Equation 4.5 yields:

$$c_s = c_{ss} + c_w\left(\frac{w}{100}\right) = 0.175 + 1.0\left(\frac{10}{100}\right) = 0.275 \text{ Btu/lb}_m\text{·°F} \qquad \text{(I-P)}$$

$$c_s = c_{ss} + c_w\left(\frac{w}{100}\right) = 0.73 + 4.18\left(\frac{10}{100}\right) = 1.15 \text{ kJ/kg·K} \qquad \text{(SI)}$$

The calculation of the thermal diffusivity using the thermal conductivity, the specific heat, and the density is illustrated in the next section.

## UNDISTURBED SOIL TEMPERATURES

Before any heat gain calculations may be conducted, the undisturbed soil temperature at the site must be determined. The choice of soil temperature is guided primarily by the type of calculation being conducted; see the section on thermal design considerations. The appropriate choice of undisturbed soil temperature also depends on the location of the site, time of year, depth of burial, and thermal properties of the soil. Some methods for determining undisturbed soil temperatures are as follows:

The average annual air temperature may be used to approximate the average annual soil temperature. The groundwater temperature is also a very good approximation of the average annual soil temperature. However, groundwater must not be confused with shallow perched water tables. Use of the average annual soil temperature is appropriate when the objective of the calculation is to yield the average heat gain over the yearly weather cycle. Average annual air temperatures may be obtained from various sources of climatic data; *ASHRAE Handbook—Fundamentals* contains a summary of climatic design information for 8118 locations worldwide on a CD included with the publication (ASHRAE 2017).

Use the maximum/minimum air temperature as an estimate of the maximum/minimum undisturbed soil temperature for pipes buried at a shallow depth. This approximation is normally a conservative assumption. Maximum and minimum expected air temperatures may be found on the CD included with ASHRAE (2017).

For systems that are buried at other than shallow depths, maximum/minimum undisturbed soil temperatures may be estimated as a function of depth, soil thermal properties, and prevailing climate. The following equations may be used to estimate the minimum and maximum expected undisturbed soil temperatures.

For maximum temperature:

$$T_{s,z} = T_{ms} + A_s e^{-z\sqrt{\frac{\pi}{\alpha\tau}}} \tag{4.6}$$

For minimum temperature:

$$T_{s,z} = T_{ms} - A_s e^{-z\sqrt{\frac{\pi}{\alpha\tau}}} \tag{4.7}$$

where:

| | | |
|---|---|---|
| $T_{s,z}$ | = | temperature, °F (°C) |
| $z$ | = | depth, ft (m) |
| $\tau$ | = | annual period length, 365 days |
| $\alpha$ | = | thermal diffusivity of the soil, ft2/day (m2/day) |
| $T_{ms}$ | = | mean annual surface temperature, °F (°C) |
| $A_s$ | = | surface temperature amplitude, °F (°C) |

Values for the climatic constants $T_{ms}$ and $A_s$ for 8118 locations worldwide on the CD included with *ASHRAE Handbook—Fundamentals* (ASHRAE 2017) may be found at https://tc0602.ashraetcs.org/Climatic_constants_using_ASHRAE_CD_Ver_6.0.pdf. Alternately, Phetteplace et al. (2013) provides a method to find the climatic constants given any set of observed or derived climatic data.

Thermal diffusivity for soil may be calculated as follows:

$$\alpha = \frac{24k_s}{\rho_s c_s} \quad \text{(I-P)} \tag{4.8a}$$

$$\alpha = \frac{86.4k_s}{\rho_s c_s} \quad \text{(SI)} \tag{4.8b}$$

where:

| | | |
|---|---|---|
| $\rho_s$ | = | soil density, $lb_m/ft^3$ ($kg/m^3$) |

For instances where specific temperatures other than the maximum or minimum are needed, the undisturbed soil temperatures may be estimated for any time of the year as a function of depth, soil thermal properties, and prevailing climate. This temperature, as well as those calculated with Equations 4.6 and 4.7, may be used in lieu of the soil surface temperature normally called for by the steady-state heat transfer equations when estimates of heat loss/gain as a function of the time of year are desired. The substitution of the undisturbed soil temperatures at the pipe depth allows the steady-state equations to be used as a first approximation to the solution to the actual transient heat transfer problem with its annual temperature variations at the surface. The following equation may be used to estimate the undisturbed soil temperature at any depth at any point during the yearly weather cycle (ASCE 1996).

$$T_{s,z} = T_{ms} + A_s e^{-z\sqrt{\frac{\pi}{\alpha\tau}}} \sin\left(\frac{2\pi(t - t_{lag})}{\tau} - z\sqrt{\frac{\pi}{\alpha\tau}}\right) \tag{4.9}$$

Note: argument for sin function is in radians.

where:

| | | |
|---|---|---|
| $t$ | = | Julian date, (i.e., numerical day of year, 1 to 365, 0 = 1 January), days |
| $t_{lag}$ | = | phase lag of soil surface temperature, days |

Values for the climatic constants $T_{ms}$, $A_s$, and $t_{lag}$ are available from https://tc0602.ashraetcs.org/Climatic_constants_using_ASHRAE_CD_Ver_6.0.pdf for 8118 locations worldwide on the CD included with *ASHRAE Handbook—Fundamentals*. In addition, Phetteplace et al. (2013) provides a method to find the climatic constants given any set of observed or contrived climatic data. Equation 4.8 may be used to calculate soil thermal diffusivity.

Equation 4.9 does not account for latent heat effects due to freezing, thawing, or evaporation. For buried CHW systems, freezing may be a consideration, and thus systems that are not used or drained during the winter months should be buried below the seasonal frost depth. For systems that are in use but are buried above the frost depth, freezing may still occur if flow is low or nonexistent in portions of the system and/or the burial depth is shallow when compared to the frost depth.

In the calculation of the climatic constants available at https://tc0602.ashraetcs.org/Climatic_constants_using_ASHRAE_CD_Ver_6.0.pdf, the ground surface temperature is assumed to equal the air temperature, which is an acceptable assumption for most design calculations. If a more accurate calculation is desired, the methods presented below may be used to compensate for the convective thermal resistance to heat transfer at the ground surface and the impacts of the type of surface cover. Appendix A provides additional information on methods to account for the surface type.

**Example 4.2: Soil Temperature Calculations**

To illustrate the use of the equations presented in this section, consider the most general case of determining soil temperatures both as a function of time and depth using Equation 4.9. We will assume a coastal Massachusetts climate, which yields the following values for the climatic constants:

$$T_{ms} = 49.0°\text{F } (9.5°\text{C})$$

$$A_s = 20.6°\text{F } (11.4 \text{ K})$$

$$t_{lag} = 115.9 \text{ days}$$

For soil, this example will use the same unfrozen, sandy soil with a moisture content of 10% at a dry density of 100 $\text{lb}_m/\text{ft}^3$ (1600 $\text{kg/m}^3$) from the example in the previous section, which yielded

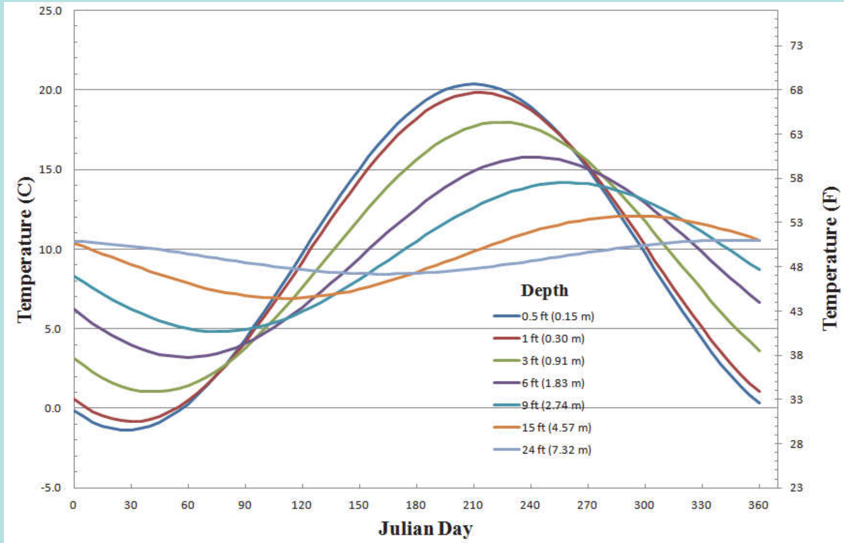



**Figure 4.9** Soil temperatures calculated with Equation 4.9 for a coastal Massachusetts climate and the assumption that surface temperature is equal to air temperature.

the following thermal properties:

$$k_s = 0.91 \text{ Btu/h·ft·°F } (1.58 \text{ W/m·K})$$

$$c_s = 0.275 \text{ Btu/lb}_m\text{·°F } (1.15 \text{ kJ/kg·K})$$

The thermal diffusivity is calculated with Equation 4.8a:

$$\alpha = \frac{24k_s}{\rho_s c_s} = \frac{24(0.91)}{100(0.275)} = 0.79 \text{ ft}^2\text{/day } (0.074 \text{ m}^2\text{/day})$$

Substituting these thermal property and climatic constant values into Equation 4.9 we have:

$$T_{s,z} = 49.0 + 20.6e^{-z\sqrt{\frac{\pi}{0.79(365)}}}\sin\left[\frac{2\pi(t-115.9)}{365} - z\sqrt{\frac{\pi}{0.79(365)}}\right]$$

$$= 49.0 + 20.6e^{-0.10z}\sin\{[0.0172(t-115.9)] - 0.10z\}$$

This result is for I-P units, and for simplicity, the result for the SI version of Equation 4.9 is omitted.[1] With this result, we can now evaluate the soil temperature for any depth, $z$ (ft) and time, $t$ (Julian day). A series of calculations have been made using this result and they are presented in Figure 4.9. These calculations are based on the assumption that the ground surface temperature is equal to the air temperature; to adjust for a convective coefficient, the value calculated by Equation 4.10 would simply be added to the burial depth of interest to calculate the appropriate depth value in Figure 4.9.

1. For consistency and simplicity, thermal conductivities in I-P units are given in Btu/h·ft·°F with all dimensions in feet (not the more traditional Btu·in./h·ft$^2$·°F). By doing so, conversion factors within the equations themselves are not needed and in most instances the equations apply for either I-P or SI units.

## Heat Transfer at Ground Surface

Heat transfer between the ground surface and the ambient air occurs by convection. In addition, heat transfer also takes place as a result of incident precipitation and radiation. The heat balance at the ground surface is too complex to warrant detailed treatment in the design of buried district heating and cooling systems, the interested reader is referred to Lunardini (1981). For many district cooling applications calculations using the simplified method presented below will provide adequate results. However, for shallow burial under pavement, especially of large diameter pipes, the simplified method below may not be adequate as it will likely significantly underpredict soil temperatures under the pavement and hence the heat gain to the buried district cooling piping will also be underpredicted. For uninsulated piping the impacts will be much greater than for those of insulated systems. Additionally, the greater the level of insulation the lesser the impacts will be. Thus, one method for dealing with the unknown impact of surface pavements is to use insulation thicknesses greater than what would be indicated by an analysis that ignores the impact of surface type. Appendix A presents additional details on the surface heat balance and how surface type will impact subsurface temperatures.

As a first approximation for the convective heat transfer, an effective thickness of a fictitious soil layer may be added to the burial depth to account for the effect of the convective heat transfer resistance at the ground surface. The effective thickness is calculated as follows:

$$\delta = \frac{k_s}{h} \tag{4.10}$$

where

$\delta$ = effective thickness of fictitious soil layer, ft (m)

$h$ = convective heat transfer coefficient at ground surface, Btu/h·ft$^2$·°F (W/m$^2$·K)

The effective thickness calculated with Equation 4.10 is simply added to the actual burial depth of the pipes in calculating the soil thermal resistance using equations presented in the "Steady-State Heat Gain Calculations for Systems" section.

# INSULATIONS AND THEIR THERMAL PROPERTIES

Insulation normally provides the primary thermal resistance against heat gain in insulated district cooling distribution systems. The thermal properties and other characteristics of insulations normally used in district cooling distribution systems are listed in Table 4.3. The material properties provided in Table 4.3 should only be used in the absence of data specific to the actual insulation material to be used/in use. Material properties such as thermal conductivity, density, compressive strength, moisture absorption, dimensional stability, and combustibility

**Table 4.3** Insulations for Buried DCSs

| Mean Temperature | Thermal Conductivity, Btu/h·ft·°F (W/m·K) | |
|---|---|---|
| | Urethane Foam | Cellular Glass ASTM C 552 |
| 0°F (–18°C) | 0.0145 (0.0251) | 0.0206 (0.0356) |
| 50°F (10°C) | 0.0138 (0.0238) | 0.0231 (0.0399) |
| 100°F (40°C) | 0.0138 (0.0238) | 0.0258 (0.0445) |
| 150°F (66°C) | 0.0150 (0.0260) | 0.0285 (0.0493) |
| Density, lb/ft$^3$ (kg/m$^3$) | 2.0 (32) | 7.5 (120) |

*Data from Nayyar 2000*

are typically reported in the ASTM standard for the respective material. Some properties have more than one associated standard. For example, thermal conductivity for insulation material in block-form may be reported using ASTM C 177 (2010a), C518 (2010b), or C1114 (2006). Thermal conductivity for insulation material that is fabricated or molded for use on piping is reported using ASTM C335 (2010c).

Table 4.3 contains the two most common insulations used in the construction of district cooling piping systems. For other insulations, refer to the insulation manufacturer, or for preliminary calculations, Phetteplace et al. (2013) or Nayyar (2000).

## STEADY-STATE HEAT GAIN CALCULATIONS FOR SYSTEMS

This section presents the formulas necessary to calculate steady-state heat gains from the most common district cooling piping system geometries; for other types of buried systems refer to Phetteplace et al. (2013). The most important factors affecting heat transfer are the difference between earth and fluid temperatures and the thermal insulation. Other factors that affect heat transfer are depth of burial related to the earth temperature and soil thermal resistance; soil thermal conductivity related to soil moisture content and density; and distance between adjacent pipes.

For complex geometries and to compute transient heat gains, numerical methods that approximate any physical problem and include factors such as the effect of temperature on thermal properties, provide the most accurate results. For most designs, numerical analyses are probably not warranted, except where simpler steady-state analyses predict that the potential exists to thermally damage the CHW distribution system due to an adjacent heating system or heated object. Albert and Phetteplace (1983), Minkowycz et al. (1988), and Rao (1982) have further information on numerical methods.

Steady-state calculations are appropriate for determining the annual heat gain from a buried DCS if the average annual earth temperatures are used. Steady-state calculations may also be appropriate to approximate the maximum heat gain during the peak summer period to establish the design cooling load.

The following steady-state methods of analysis use resistance formulations developed by Phetteplace and Meyer (1990) that simplify the calculations needed to determine temperatures within the system. Each type of resistance is given a unique subscript and is defined only when introduced. In each case, the resistances are on a unit-length basis so that heat flows per unit length result directly when the temperature difference is divided by the resistance.

### Single Uninsulated Buried Pipe

For this case (Figure 4.10), an approximation for the soil thermal resistance has been used extensively. This approximation is sufficiently accurate (within 1%) for the ratios of burial depth to pipe radius indicated next to Equations 4.11 and 4.12. Both the actual resistance and the approximate resistance are presented, along with the depth/radius criteria for each.

$$R_s = \frac{\ln\left\{\left(\frac{d}{r_o}\right) + \left[\left(\frac{d}{r_o}\right)^2 - 1\right]^{\frac{1}{2}}\right\}}{2\pi k_s} \qquad \text{for } \frac{d}{r_o} > 2 \tag{4.11}$$

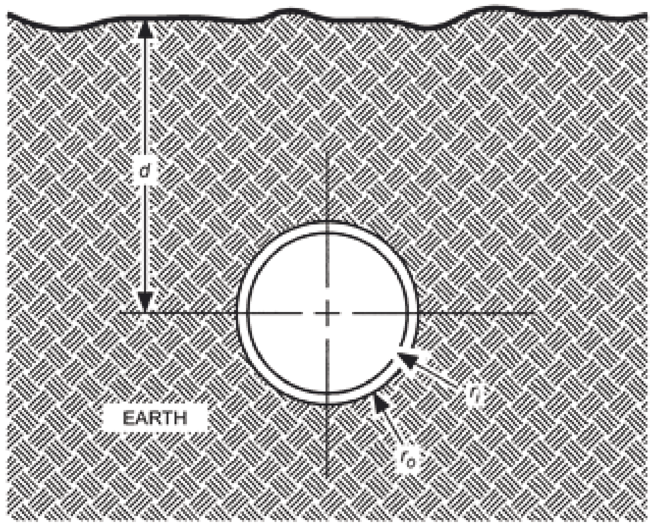



**Figure 4.10** Single uninsulated buried pipe.

$$R_s = \frac{\ln\left(\frac{2d}{r_o}\right)}{2\pi k_s} \qquad \text{for } \frac{d}{r_o} > 4 \tag{4.12}$$

where

| | | |
|---|---|---|
| $R_s$ | = | thermal resistance of soil, h·ft·°F/Btu (m·K/W) |
| $k_s$ | = | thermal conductivity of soil, Btu/h·ft·°F (W/m·K) |
| $d$ | = | burial depth to centerline of pipe, ft (m) |
| $r_o$ | = | outer radius of pipe or conduit, ft (m) |

Include the thermal resistance of the pipe if it is significant when compared to the soil resistance. The thermal resistance of a pipe or any concentric circular region is given by

$$R_p = \frac{\ln\left(\frac{r_o}{r_i}\right)}{2\pi k_p} \tag{4.13}$$

where

| | | |
|---|---|---|
| $R_p$ | = | thermal resistance of pipe wall, h·ft·°F/Btu (m·K/W) |
| $k_p$ | = | thermal conductivity of pipe, Btu/h·ft·°F(W/m·K) |
| $r_i$ | = | inner radius of pipe, ft (m) |

**Example 4.3:** Consider an uninsulated, 3 in. (75 mm) Schedule 40 PVC CHW supply line carrying 45°F (7°C) water. Assume the pipe is buried 3 ft (0.91 m) deep in soil with a thermal conductivity of 1 Btu/h·ft·°F (1.7 W/m·K), and no other pipes or thermal anomalies are within close proximity. Assume the average annual soil temperature is 60°F (16°C).

$r_i$ = 1.54 in. = 0.128 ft (0.0390 m)
$r_o$ = 1.75 in. = 0.146 ft (0.0445 m)
$d$ = 3 ft (0.91 m)
$k_s$ = 1 Btu/h·ft·°F (1.7 W/m·K)
$k_p$ = 0.10 Btu/h·ft·°F (0.17 W/m·K)

**Solution:** Calculate thermal resistance of the pipe using Equation 4.13:

$$R_p = 0.21 \text{ h·ft·°F/Btu } (0.12 \text{ m·K/W})$$

Calculate the thermal resistance of the soil using Equation 4.12. (*Note*: $\dfrac{d}{r_o} = 21$ is greater than 4; thus Equation 4.12 may be used in lieu of Equation 4.11.)

$$R_s = 0.59 \text{ h·ft·°F/Btu } (0.34 \text{ m·K/W})$$

Calculate the rate of heat transfer by dividing the overall temperature difference by the total thermal resistance:

$$q = \frac{t_f - t_s}{R_t} = \frac{(45 - 60)}{0.80 \text{ h·ft·°F/Btu}} = -19 \text{ Btu/h·ft } (-18 \text{ W/m})$$

where:

$R_t$ = total thermal resistance (i.e., $R_s + R_p$ in this case of pure series heat flow), h·ft·°F/Btu (m·K/W)
$t_f$ = fluid temperature, °F (°C)
$t_s$ = average annual soil temperature, °F (°C)
$q$ = heat loss or gain per unit length of system, Btu/h·ft (W/m)

The negative result indicates a heat gain rather than a loss. Note that the thermal resistance of the fluid/pipe interface has been neglected. This is a reasonable assumption, as such resistances tend to be very small for flowing fluids. Also note that, in this case, the thermal resistance of the pipe comprises a significant portion of the total thermal resistance. This results from the relatively low thermal conductivity of PVC, compared with other piping materials and the fact that no other major thermal resistances exist in the system to overshadow it. If any significant amount of insulation were included in the system, its thermal resistance would dominate and it might be possible to neglect that of the piping material.

## Single Buried Insulated Pipe

Equation 4.13 can be used to calculate the thermal resistance of any concentric, circular region of material, including an insulation layer. When making calculations using insulation thickness, actual thickness rather than nominal thickness should be used to obtain the most accurate results.

**Example 4.4:** Consider the effect of adding 1 in. (2.5 cm) of urethane foam insulation and a 0.125 in. (3 mm) thick PVC jacket to the CHW line in Example 4.3. Using the thermal conductivity value for an insulation mean temperature of 50°F (10°C), calculate the thermal resistance of the insulation layer from Equation 4.13 as follows:

$$R_i = \frac{\ln\left(\frac{0.229}{0.146}\right)}{2\pi \times 0.0138} = 5.19 \text{ h·ft·°F/Btu} \ \ (3.00 \text{ m·K/W})$$

For the PVC jacket material assume a thermal conductivity of 0.10 Btu/h·ft·°F and use Equation 4.13 again:

$$R_j = \frac{\ln\left(\frac{0.240}{0.229}\right)}{2\pi \times 0.10} = 0.07 \text{ h·ft·°F/Btu} \ \ (0.04 \text{ m·K/W})$$

The thermal resistance of the soil as calculated by Equation 4.12 decreases slightly to $R_s$ = 0.51 h·ft·°F/Btu (0.29 m·K/W) because of the increase in the outer radius of the piping system. The total thermal resistance is now:

$$R_t = R_p + R_i + R_j + R_s = 0.21 + 5.19 + 0.07 + 0.51 = 5.98 \text{ h·ft·°F/Btu} \ \ (3.45 \text{ m·K/W})$$

The heat gain by the CHW pipe is reduced to about 2.5 Btu/h·ft (2.4 W/m). In this case, the thermal resistance of the piping material and the jacket material could be neglected with a resultant error of <5%. Considering that the uncertainties in the material properties are likely greater than 5%, it is usually appropriate to neglect minor resistances such as those of piping and jacket materials when insulation is present.

## Two Buried Pipes or Conduits

This case (Figure 4.11) may be formulated in terms of the thermal resistances used for a single buried pipe or conduit and some geometric and temperature factors. The factors needed are

$$\theta_1 = \frac{(t_{p2} - t_s)}{(t_{p1} - t_s)} \tag{4.14}$$

$$\theta_2 = \frac{1}{\theta_1} = \frac{(t_{p1} - t_s)}{(t_{p2} - t_s)} \tag{4.15}$$

$$P_1 = \frac{1}{2\pi k_s} \ln\left[\frac{(d_1 + d_2)^2 + a^2}{(d_1 - d_2)^2 + a^2}\right]^{0.5} \tag{4.16}$$

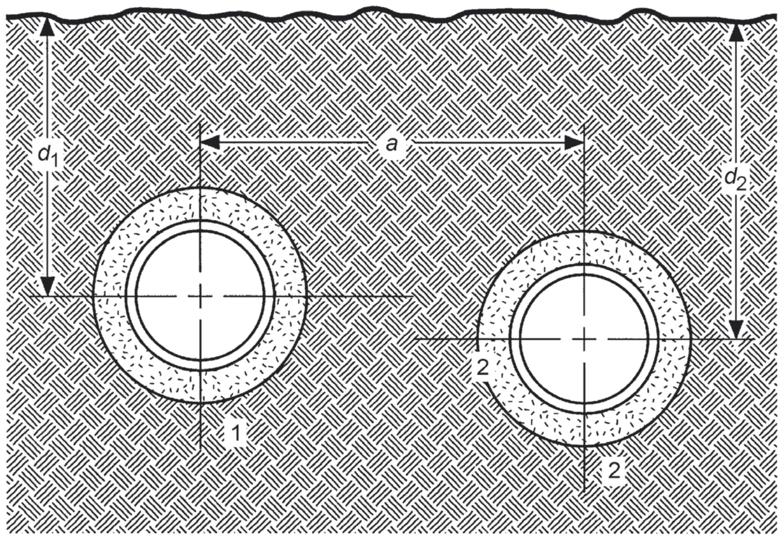



**Figure 4.11** Two buried pipes or conduits.

$$P_2 = \frac{1}{2\pi k_s}\ln\left[\frac{(d_2+d_1)^2+a^2}{(d_2-d_1)^2+a^2}\right]^{0.5} \tag{4.17}$$

where

$a$ = horizontal separation distance between centerline of two pipes, ft (m)

And the thermal resistance for each pipe or conduit is given by

$$R_{e1} = \frac{R_{t1}-\left(\frac{P_1^2}{R_{t2}}\right)}{1-\left(\frac{P_1\theta_1}{R_{t2}}\right)} \tag{4.18}$$

$$R_{e2} = \frac{R_{t2}-\left(\frac{P_2}{R_{t1}}\right)}{1-\left(\frac{P_2\theta_2}{R_{t1}}\right)} \tag{4.19}$$

where

$\theta$ = temperature factor, dimensionless

$P$ = geometric/material factor, h·ft·°F/Btu (m·K/W)

$R_e$ = effective thermal resistance of one pipe/conduit in two-pipe system, h·ft·°F/Btu (m·K/W)

$R_t$ = total thermal resistance of one pipe/conduit if buried separately, h·ft·°F/Btu (m·K/W)

Heat flow from each pipe is then calculated from

$$q_1 = \frac{(t_{p1} - t_s)}{R_{e1}} \tag{4.20}$$

$$q_2 = \frac{(t_{p1} - t_s)}{R_{e2}} \tag{4.21}$$

**Example 4.5:** Consider the buried supply and return lines for a DCS typical of a Middle Eastern application. The carrier pipes are 42 in. (1050 mm) standard weight steel (42 in. outer diameter [1067 mm]) with 2 in. (50 mm) of urethane foam insulation. The insulation is protected by a 0.25 in. (6.4 mm) thick HDPE jacket. The thermal conductivity of the insulation is per Table 4.3: 0.0138 Btu/h·ft·°F (0.0238 W/m·K) mean annual temperature of 82°F (27.8°C). The horizontal distance between the pipe centerlines is 7 ft (2.13 m). The supply water is 40°F (4.4 °C), and the return water is 55°F (12.8°C).

**Solution:** Neglect the thermal resistances of the carrier pipes and the HDPE jacket. Because the pipes are large but burial is not deep, we need to check to see if Equation 4.11 or 4.12 should be used.

First, we calculate the outer radius of the piping system:

$$r_o = \frac{(21 + 2 + 0.25)}{12} = 1.94 \text{ ft (0.59 m)}$$

Now the ratio of the depth-to-pipe outer radius is calculated:

$$\frac{d}{r_o} = \frac{6}{1.93} = 3.1$$

Since this result is less than 4, we must use the more complicated Equation 4.11 in lieu of Equation 4.12 for calculating the soil thermal resistances. Using Equation 4.11 as if the pipes were independent of each other:

$$R_{s1} = R_{s2} = \frac{\ln\left[(3.1) + (3.1^2 - 1)^{\frac{1}{2}}\right]}{2\pi \times 1.25} = 0.23 \text{ h·ft·°F/Btu (0.13 m·K/W)}$$

The insulation resistances are calculated with Equation 4.13, again as if the pipes were independent of each other:

$$R_{i1} = R_{i2} = \frac{\ln\left(\frac{1.92}{1.75}\right)}{2\pi \times 0.0138} = 1.07 \text{ h·ft·°F/Btu (0.618 m·K/W)}$$

Now the total resistance of the soil and insulation is calculated, once again as if the pipes were independent:

$$R_{t1} = R_{t2} = 0.23 + 1.07 = 1.30 \text{ h·ft·°F/Btu} \ \ (0.751 \text{ m·K/W})$$

From Equations 4.16 and 4.17, the geometric and temperature factors are:

$$P_1 = P_2 = \frac{1}{2\pi \times 1.25} \ln\left[\frac{(6+6)^2 + 7^2}{(6-6)^2 + 7^2}\right]^{0.5} = 0.0873 \text{ h·ft·°F/Btu} \ \ (0.0504 \text{ m·K/W})$$

$$\theta_1 = \frac{(55-82)}{(40-82)} = 0.643$$

$$\theta_2 = \frac{1}{\theta_1} = 1.56$$

Calculate the effective total thermal resistances from Equations 4.18 and 4.19 as:

$$R_{e1} = \frac{1.30 - \left(\frac{0.0873^2}{1.30}\right)}{1 - \left(\frac{0.0873 \times 0.643}{1.30}\right)} = 1.35 \text{ h·ft·°F/Btu} \ \ (0.782 \text{ m·K/W})$$

$$R_{e2} = \frac{1.30 - \left(\frac{0.0873^2}{1.30}\right)}{1 - \left(\frac{0.0873 \times 1.56}{1.30}\right)} = 1.45 \text{ h·ft°F/Btu} \ \ (0.836 \text{ m·K/W})$$

The heat flows are then given by Equations 4.20 and 4.21:

$$q_1 = \frac{(40-82)}{1.35} = -31.1 \text{ Btu/h·ft} \ \ (-29.9 \text{ W/m})$$

$$q_2 = \frac{55-82}{1.45} = -18.6 \text{ Btu/h·ft} \ \ (-17.9 \text{ W/m})$$

$$R_{e2} = \frac{1.30 - \left(\frac{0.0873^2}{1.30}\right)}{1 - \left(\frac{0.0873 \times 1.56}{1.30}\right)} = 1.45 \text{ h·ft°F/Btu} \ \ (0.836 \text{ m·K/W})$$

The heat flows are then given by Equations 4.20 and 4.21:

$$q_1 = \frac{(40 - 82)}{1.35} = -31.1 \text{ Btu/h·ft } (-29.9 \text{ W/m})$$

$$q_2 = \frac{55 - 82}{1.45} = -18.6 \text{ Btu/h·ft } (-17.9 \text{ W/m})$$

$$q_t = (-31.1) + (-18.6) = -49.7 \text{ Btu/h·ft } (-47.8 \text{ W/m})$$

Note that when the resistances and geometry for the two pipes are identical, the total heat flow into the two pipes is the same if the temperature corrections are used or if they are set to unity. The individual heat gains will vary somewhat, however.

## WHEN TO INSULATE DISTRICT COOLING PIPING

The decision to insulate buried CHW piping is driven by three principal issues:

- The energy use impact of heat gain into the system
- The cost of the additional chiller plant capacity needed due to the system heat gain
- The impact that heat gains will have on the ability to deliver an adequate supply temperature to the consumer and the subsequent impact on the consumer's equipment

### Energy Cost Impact of Heat Gain

Calculating the heat gain into the distribution system is a straightforward exercise using the methods outlined earlier in this chapter. Where the system is in operation for 12 months per year, this calculation would be done using the average annual soil temperature, which can be approximated by the average annual air temperature as discussed earlier. If the system will only be in operation for a portion of the year, then the average annual temperature should not be used, but rather the average soil temperature for the season of operation should be used; otherwise, the heat gain will be underestimated as the cooling season will be the warmest time of the year. When operation is seasonal, the soil temperatures at the depth of burial should be used for the most accurate analysis. These would be computed using Equation 4.9 and then averaged for the period of operation. Regardless of the period of operation, if significant portions of the system will be located under paved areas the adjustment described in Appendix A.

As an example cost of the heat gain from insulated versus uninsulated piping, consider the piping of Example 4.5. Buried under the same conditions but without insulation, the average annual heat gain from the piping system would be 153 Btu/h·ft (147 W/m) for the supply pipe and 56 Btu/h·ft (54 W/m) for the return pipe, or a total of 209 Btu/h·ft (201 W/m). The total heat gain is thus 159 Btu/h·ft (153 W/m) more for the uninsulated system. Assuming year-round operation, average overall chiller plant performance of 1 kW/ton, and electricity at $0.12/kWh, the additional heat gain would amount to $13.70 per year per foot of piping ($45 per year per meter). Over a 30-year system life, simple payback indicates this would justify an additional expense for the insulated piping of about $400 per foot ($1312 per meter). This is only the energy consumption and it does not account for additional costs at the chiller plant of generating the CHW, for example condenser water and its treatment. These costs should be considered when comparing the cost of both uninsulated and insulated systems fully installed. For example, uninsulated steel piping will require a coating be installed, something that would be part of the insulated system. An uninsulated system might also require cathodic protection. Another factor that should be considered is that the insulated system will require more labor for installation due to the need to insulate and jacket the field joints.

In theory, such a calculation could be carried out for each pipe size (and in instances of seasonal operation, pipe size and burial depth combination) in the system; however, in practicality, a decision will be reached based on insulated or uninsulated piping for essentially the entire system. It is possible that in some circumstances insulation may not be warranted due to the economic impact of the heat gain, but rather it may be required due to unacceptable supply-temperature degradation, as discussed later. In those cases, it may turn out that it will be necessary to insulate the smaller diameter piping where the heat gain is much greater in proportion to the heat capacity of the water being transported (transport capacity varies with the square of pipe diameter, while heat gain varies in a nearly linear fashion).

## Cost of Additional Chiller Plant Capacity

This calculation must proceed under the design conditions, i.e., the worst condition that will be encountered, thus the use of the average annual soil temperature is not appropriate. The following is an example using the system of Example 4.5 where the maximum annual soil temperature at the depth of the pipes is calculated.

**Example 4.6:** Calculate the maximum rate of heat gain for the system of Example 4.5 for the design condition in both the uninsulated and insulated condition.

**Solution:** The climatic constants $T_{ms}$, $A_s$, and $t_{lag}$ are found at http://tc62.ashraetcs.org/pdf/ASHRAE_Climatic_Data.pdf as:

$T_{ms}$ = 81.8°F (27.7°C)
$A_s$ = 13.6°F (7.5°C)
$t_{lag}$ = 114 days

Since the pipe is not buried at a shallow depth, the temperature at the depth of the pipe needs to be calculated. Now that we have the climatic constants, the temperature at the depth of the pipe can be calculated; Equation 4.6 is used to obtain the maximum temperature, or Equation 4.9 may be used to obtain the temperature for any time of the year. Before doing so, a density and moisture content for the soil will need to be assumed in order to compute the specific heat and then the thermal diffusivity (the thermal conductivity is taken from Example 4.5). The density will be assumed to be 110 $lb_m/ft^3$ (1760 $kg/m^3$) and the moisture content will be taken as 10%. With this, the specific heat can be calculated from Equation 4.5:

$$c_s = c_{ss} + c_w\left(\frac{w}{100}\right) = 0.175 + 1.0\left(\frac{10}{100}\right) = 0.275\ \text{Btu/lb}_m\cdot°\text{F} \quad (1.15\text{kJ/kg}\cdot\text{K})$$

And now we may compute the thermal diffusivity using Equation 4.8:

$$\alpha = \frac{24k_s}{\rho_s c_s} = \frac{24(1.25)}{110(0.275)} = 0.99\ \text{ft}^2/\text{day} \qquad (0.092\ \text{m}^2/\text{day})$$

Now we use Equation 4.6 to obtain the maximum soil temperature at our depth of 6 ft (1.83 m) to the centerline:

$$T_{s,z} = T_{ms} + A_s e^{-z\sqrt{\frac{\pi}{\alpha\tau}}} = 81.8 + 13.6\left(e^{-6\sqrt{\frac{\pi}{0.99 \times 365}}}\right) = 89.6°\text{F} \ \ (32.0°\text{C})$$

Equation 4.9 has also been used to prepare the following graph showing the calculated soil temperatures at various depths for the assumptions of this example.[3]

Now recalculate the temperature factors from Example 4.5:

$$\theta_1 = \frac{(55 - 89.6)}{(40 - 89.6)} = 0.698$$

$$\theta_2 = \frac{1}{\theta_1} = 1.43$$

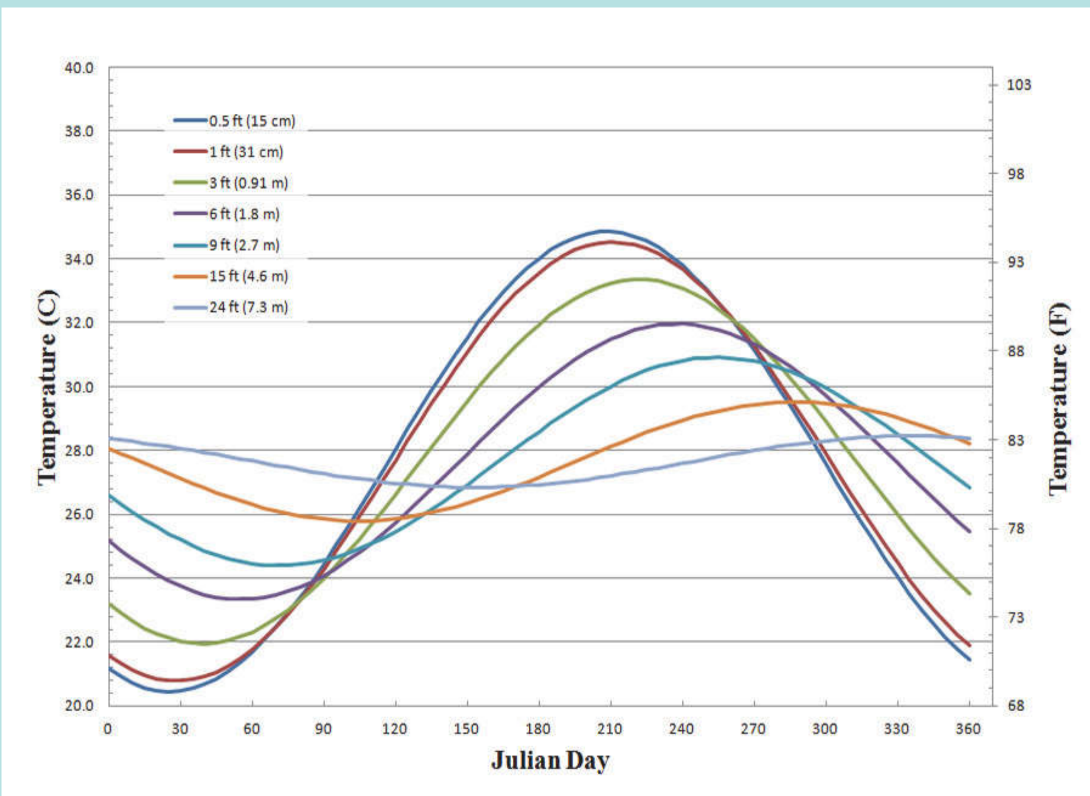



**Figure 4.12** Annual soil temperature variation with depth for Example 4.6.

Next we calculate the effective total thermal resistances from Equations 4.18 and 4.19 as:

$$R_{e1} = \frac{1.30 - \left(\frac{0.0873^2}{1.30}\right)}{1 - \left(\frac{0.0873 \times 0.698}{1.30}\right)} = 1.36\ \text{h·ft·°F/Btu}\ \ (0.786\ \text{m·K/W})$$

$$R_{e2} = \frac{1.30 - \left(\frac{0.0873^2}{1.30}\right)}{1 - \left(\frac{0.0873 \times 1.53}{1.30}\right)} = 1.44\ \text{h·ft·°F/Btu}\ \ (0.834\ \text{m·K/W})$$

The heat flows are then given by Equations 4.20 and 4.21:

$$q_1 = \frac{(40 - 89.6)}{1.36} = -36.5\ \text{Btu/h·ft}\ \ (-35.0\ \text{W/m})$$

$$q_2 = \frac{(55 - 89.6)}{1.44} = -24.0\ \text{Btu/h·ft}\ \ (-23.1\ \text{W/m})$$

$$q_1 = (-36.5) + (-24.0) = -60.5\ \text{Btu/h·ft}\ \ (-58.1\ \text{W/m})$$

Thus this maximum total heat gain is approximately 22% higher than the average total heat gain of 49.7 Btu/h·ft (47.8 W/m) calculated in Example 4.5. Repeating the calculation for uninsulated piping buried under the same conditions results in the maximum heat gain from the piping system of 177 Btu/h·ft (170 W/m) for the supply pipe and 80.0 Btu/h·ft (76.9 W/m) for the return pipe, or a total of 257 Btu/h·ft (247 W/m). This maximum heat gain represents an increase of approximately 23% above the average total heat gain of 209 Btu/h·ft (201 W/m) calculated above for the uninsulated system.

---

3. Note that it could be argued that since the maximum soil temperature at some significant burial depth will always occur after the theoretical peak of the cooling season, the maximum load imposed by the distribution network should not be added to the plant capacity required to meet the design cooling day. Rather, one might suggest that the load from the network on the theoretical design cooling day should be used. For example, in this case the theoretical peak of the cooling season would occur on approximately Julian day 205 (the lag time of 114.2 days plus one-fourth the period of 365 days); yet, close examination of the data used to generate the figure of Example 4.6 reveals that the peak temperature at the 6 foot (1.8 m) burial depth occurs at approximately Julian day 238, over a month later. Given normal climate variations it would not be difficult to imagine a design cooling day occurring a month or more from the theoretical peak of the cooling season.

The maximum total heat gain calculated in Example 4.6 is thus 197 Btu/h·ft (189 W/m) more for the uninsulated system. To calculate the impact of this higher heat gain rate of the uninsulated system on the chiller plant, it would be necessary to make this calculation for each pipe diameter in the distribution system and multiply these results by the lengths and then sum them. In terms of the incremental cost of the cooling plant capacity that would be needed per unit length of uninsulated versus insulated pipe, if we use the median cost for a chiller plant of $2650/ton from

*ASHRAE Handbook—HVAC Systems and Equipment* (ASHRAE 2012), we find that an additional cost of approximately $44/ft ($144/m) of pipe will be incurred over the effective life of the chiller plant. This is somewhat overshadowed by the energy cost impacts of the uninsulated system, which we estimated above to be approximately $400/ft ($1312/m); however, it is still significant. Note that under a different set of assumptions, the relative results would likely change.

## Impacts of Heat Gain on Delivered Supply Water Temperature

This calculation is predicated by yet another design condition, i.e., the worst condition that will be encountered not just over the annual cycle, but also in terms of the connected loads. The most distant customer from a supply-temperature degradation standpoint will establish the location in the distribution system where the design condition occurs, the maximum soil temperature at the depth of burial must be applied at that location. In calculating temperature degradation in the distribution network, the impacts will normally be greatest in the piping most distant from the plant and closest to the consumer as this piping will be the smallest in diameter, and as noted earlier, transport capacity varies with the square of pipe diameter while heat gain varies in a nearly linear fashion. The impact of insulation on the service pipes will be illustrated in Example 4.7.

**Example 4.7:** Following the Middle Eastern district cooling application of Example 4.5 and Example 4.6, now consider buried supply and return laterals from the main distribution system to a small consumer with a 25 ton (88 kW) peak cooling load. The objective is to calculate the supply-temperature degradation in the supply lateral which is 1000 ft (305 m) long under the design condition. Given our temperature difference between supply and return of 15°F (8.3°C), the 25 ton (88 kW) peak heat load will require a flow rate of 40 gal/min (2.51 l/s). From *ASHRAE Handbook—Fundamentals* (ASHRAE 2017)*,* we see that a steel 2 in. (50 mm) NPS (nominal pipe size) pipe is able to carry that flow at a head loss of approximately 3.2 ft per 100 ft of pipe (313 Pa per meter of pipe), which is within the criteria of ASHRAE (2017). Thus, carrier pipes of 2 in. (50 mm) standard weight steel (2.375 in. outer diameter [60 mm]) will be used and both insulated and uninsulated cases will be examined. For the insulated case, the pipes are insulated with 1 in. (25 mm) of urethane foam insulation. The insulation is protected by a 0.25 in. (6.4 mm) thick HDPE jacket. The thermal conductivity of the insulation is per Table 4.3: 0.0138 Btu/h·ft·°F (0.0238 W/m·K) and is assumed constant with respect to temperature. The pipes are buried 3 ft (0.91 m) deep to the centerline in soil with a thermal conductivity of 1.25 Btu/h·ft·°F (2.16 W/m·K). The soil temperature to be used is the maximum annual temperature at the depth of the pipes as determined from the calculations performed for the figure of Example 4.6, which yields 92.1°F (33.4°C). The horizontal distance between the pipe centerlines is 2 ft (0.61 m). As before, the supply water is at 40°F (4.4°C), and the return water is at 55°F (12.8°C), i.e., temperature degradation up to this point in the system is being neglected for the purpose of this calculation.

**Solution:** Neglect the thermal resistances of the carrier pipes and the HDPE jacket. Unlike Example 4.5 and Example 4.6 where the pipes were large relative to the burial depth, here we have much smaller pipes so even at one half the burial depth, the criteria for the use of Equation 4.12 rather than Equation 4.11 is easily met. Otherwise, the calculations follow the procedure used in Examples 4.5 and 4.6. For the sake of brevity, the details of the calculations have been omitted here. However, so that the reader may duplicate the calculations as an exercise, the following intermediate results are provided:

For the uninsulated case:

$$R_{t1} = R_{t2} = R_{s1} = R_{s2} = 0.523 \ \text{h·ft·°F/Btu} \ (0.302 \ \text{m·K/W})$$

$$P_1 = P_2 = 0.147 \ \text{h·ft·°F/Btu} \ (0.0850 \ \text{m·K/W})$$

$$\theta_1 = 0.712$$

$$\theta_2 = 1.40$$

$$R_{e1} = 0.602 \ \text{h·ft·°F/Btu} \ (0.348 \ \text{m·K/W})$$

$$R_{e2} = 0.794 \ \text{h·ft·°F/Btu} \ (0.459 \ \text{m·K/W})$$

For the insulated case:

$$R_{s1} = R_{s2} = 0.431 \ \text{h·ft·°F/Btu} \ (0.249 \ \text{m·K/W})$$

$$R_{i1} = R_{i2} = 7.04 \ \text{h·ft·°F/Btu} \ (4.07 \ \text{m·K/W})$$

$$R_{t1} = R_{t2} = 7.47 \ \text{h·ft·°F/Btu} \ (4.32 \ \text{m·K/W})$$

$$P_1 = P_2 = 0.147 \ \text{h·ft·°F/Btu} \ (0.0850 \ \text{m·K/W})$$

$$\theta_1 = 0.712$$

$$\theta_2 = 1.40$$

$$R_{e1} = 7.57 \ \text{h·ft·°F/Btu} \ (4.38 \ \text{m·K/W})$$

$$R_{e2} = 7.68 \ \text{h·ft·°F/Btu} \ (4.44 \ \text{m·K/W})$$

The results of the calculations are summarized in the table:

Uninsulated versus Insulated Heat Gains for a Typical Middle Eastern DC Application

| | Heat Gain, Btu/h·ft (W/m) | | |
|---|---|---|---|
| | Supply | Return | Total |
| Uninsulated | –86.5 (83.1) | –46.7 (–44.9) | –133 (128) |
| Insulated | –6.9 (–6.6) | –4.8 (–4.6) | –11.7 (11.2) |

From these results we can calculate the temperature degradation in the supply pipe using the following equation:

For the uninsulated case:

$$\Delta T = \frac{q_s L}{\dot{m} c_p} = 86.5 \times \frac{1000}{20000} \times 1 = 4.32°\text{F} \ \ (2.4°\text{C})$$

For the insulated case:

$$\Delta T = \frac{q_s L}{\dot{m} c_p} = 6.9 \times \frac{1000}{20000} \times 1 = 0.35°\text{F} \ \ (0.19°\text{C})$$

The temperature degradation calculated for the uninsulated case of Example 4.7 will likely be unacceptable since it would be added to any temperature degradation that would have occurred in the main supply piping up to the point of this lateral to the consumer. Note that if this lateral were located under a paved surface, or even one free of vegetation, the soil temperatures could be significantly higher and thus the heat gains and temperature degradation would be higher. Furthermore, it must be pointed out that this is the temperature degradation at design load conditions; at lower loads and lower flows the situation will likely be worse. While the temptation would be to assume that the lower load conditions would also occur at a time of the year of lower heat gain into the piping due to lower soil temperatures, such may not be the case. Normal climatic variations could easily generate significantly lower loads coincident with the peak soil temperatures, and if these occur at times of high humidity, it will become increasingly difficult to meet the consumer's required conditions for comfort. The likely outcome of such circumstances is overflowing the consumer with the accompanying loss of system $\Delta T$ and excess pumping energy that results.

It could easily be argued that the above example used to illustrate supply-temperature degradation is unrealistic due to the long lateral required to serve such a small load. However, it is not entirely unrealistic and it illustrates the point while also highlighting yet another difficulty in serving low-density loads, especially with uninsulated piping.

## REFERENCES

Andersland, O.B., and B. Ladanyi. 2004. *Frozen ground engineering*, 2d ed. Hoboken, NJ: John Wiley and Sons.

Albert, M.R., and G.E. Phetteplace. 1983. *Computer models for two-dimensional steady-state heat conduction*. CRREL Report 83-10. Hanover, NH: U.S. Army Cold Regions Research and Engineering Laboratory.

ASCE. 1996. *Cold regions utilities monograph*. D.W. Smith, Technical Editor. Reston, VA: American Society of Civil Engineers.

ASHRAE. 2012. *ASHRAE handbook—HVAC systems and equipment*. Atlanta: ASHRAE.

ASHRAE. 2017. *ASHRAE handbook—Fundamentals*. Atlanta: ASHRAE.

ASHRAE. 2018. Climatic constants for calculating subsurface soil temperatures. ASHRAE Technical Committee 6.2, District Energy. https://tc0602.ashraetcs.org/Climatic_constants_using_ASHRAE_CD_Ver_6.0.pdf.

ASWP. Undated. Section Two: Steel pipe design, American SpiralWeld pipe. In *ASWP manual*. Available at http://www.american-usa.com/system/assets/342/original/ASWP_Manual_-_Section_2_-_Steel_Pipe_Design_(6-1-13).pdf

ASME. 2012. B31.1, *Power piping*. New York: ASME.

ASTM. 2006. C1114, *Standard test method for steady-state thermal transmission properties by means of the thin-heater apparatus*. West Conshohocken, PA: ASTM International.

ASTM. 2007. ASTM D2996 - 01(2007)e1, *Standard specification for filament-wound "fiberglass" (glass-fiber-reinforced thermosetting-resin) pipe.* West Conshohocken, PA: ASTM International.

ASTM. 2009a. B88, *Standard specification for seamless copper water tube*. West Conshohocken, PA: ASTM International.

ASTM. 2009b. D2241, *Standard specification for poly(vinyl chloride) (PVC) pressure-rated ripe (SDR series).* West Conshohocken, PA: ASTM International.

ASTM. 2010a. C177, *Standard test method for steady-state heat flux measurements and thermal transmission properties by means of the guarded-hot-plate apparatus*. West Conshohocken, PA: ASTM International.

ASTM. 2010b. C518, *Standard test method for steady-state thermal transmission properties by means of the heat flow meter apparatus*. West Conshohocken, PA: ASTM International.

ASTM. 2010c. C335/C335M - 10e1, *Standard test method for steady-state heat transfer properties of pipe insulation*. West Conshohocken, PA: ASTM International.

ASTM. 2011. A106/A106M, *Standard specification for seamless carbon steel pipe for high-temperature service*. West Conshohocken, PA: ASTM International.

ASTM. 2012a. A53/A53M, *Standard specification for pipe, steel, black and hot-dipped, zinc-coated, welded and seamless*. West Conshohocken, PA: ASTM International.

ASTM. 2012b. D698, *Standard test methods for laboratory compaction characteristics of soil using standard effort (12 400 ft-lbf/ft$^3$ (600 kN-m/m$^3$))*. West Conshohocken, PA: ASTM International.

ASTM. 2012c. D1785, *Standard specification for poly(vinyl chloride) (PVC) plastic pipe, schedules 40, 80, and 120*, West Conshohocken, PA: ASTM International.

AWWA. 2007a. C301, *Prestressed concrete pressure pipe, steel-cylinder type*. Denver, CO: American Water Works Association.

AWWA. 2007b. C900, *Polyvinyl chloride (PVC) pressure pipe and fabricated fittings, 4 in. through 12 in. (100 mm through 300 mm), for water transmission and distribution*. Denver. CO: American Water Works Association.

AWWA. 2007c. C906, *Polyethylene (PE) pressure pipe and fittings 4 in. (100 mm) through 63 in. (1,600 mm) for water distribution and transmission*. Denver. CO: American Water Works Association.

AWWA. 2008a. C303, *Concrete pressure pipe, bar-wrapped, steel-cylinder type*. Denver, CO: American Water Works Association.

AWWA. 2008b. C901, *Polyethylene (PE) pressure pipe and tubing, ½ in. (13 mm) 3 in. through (76 mm) for water service*. Denver, CO: American Water Works Association.

AWWA. 2009. C151, *Ductile iron pipe, centrifgually cast, for water.* Denver, CO: American Water Works Association.

AWWA. 2010. C905, *Polyvinyl chloride (PVC) pressure pipe and fabricated fittings, 14 in. through 48 in. (350 mm through 1,200 mm) for water transmission and distribution*. Denver. CO: American Water Works Association.

AWWA. 2011a. C300, *Reinforced concrete pressure pipe, steel-cylinder type*. Denver, CO: American Water Works Association.

AWWA. 2011b. C302, *Reinforced concrete pressure pipe, noncylinder type*. Denver, CO: American Water Works Association.

AWWA. 2012. C200, *Steel water pipe 6 inch (150 mm) and larger*. Denver, CO: American Water Works Association.

AWWA. 2013. C950, *Fiberglass pressure pipe*. Denver, CO: American Water Works Association.

Chyu, M.C., X. Zeng, and L. Ye. 1997a. Performance of fibrous glass insulation subjected to underground water attack. *ASHRAE Transactions* 103(1):303–308.

Chyu, M.C., X. Zeng, and L. Ye. 1997b. The effect of moisture content on the performance of polyurethane insulation on a district heating and cooling pipe. *ASHRAE Transactions* 103(1):309–17.

Chyu, M.C., X. Zeng, and L. Ye. 1998a. Behavior of cellular glass insulation on a DHC pipe subjected to underground water attack. *ASHRAE Transactions* 104(2):161–67.

Chyu, M.C., X. Zeng, and L. Ye. 1998b. Effect of underground water attack on the performance of mineral wool pipe insulation. *ASHRAE Transactions* 104(2):168–75.

CEN. 2009a. EN 253, *District heating pipes—Preinsulated bonded pipe systems for directly buried hot water networks—Pipe assembly of steel service pipe, polyurethane thermal insulation and outer casing of polyethylene*. Brussels, Belgium: CEN.

CEN. 2009b. EN 448, *District heating pipes—Preinsulated bonded pipe systems for directly buried hot water networks—Fitting assemblies of steel service pipes, polyurethane thermal insulation and outer casing of polyethylene*. Brussels, Belgium: CEN.

CEN. 2009c. EN 489, *District heating [ipes—Preinsulated bonded pipe systems for directly buried hot water networks—Joint assembly for steel service pipes, polyurethane thermal insulation and outer casing of polyethylene*. Brussels, Belgium: CEN.

CEN. 2011. EN 488, *District heating pipes—Preinsulated bonded pipe systems for directly buried hot water networks—steel valve assembly for steel service pipes, polyurethane thermal insulation and outer casing of polyethylene*. Brussels, Belgium: CEN.Farouki, O.T. 1981. *Thermal properties of soils, CRREL monograph 81-1*. Hanover, NH: U.S. Army Cold Regions Research and Engineering Laboratory.

Ghezzi, P. 2005. Draining resources. *American City and County*, January, Penton Media, Inc.

Kavanaugh, S.P. 2000. *Investigation of methods for determining soil and rock formation thermal properties from short-term field tests*. ASHRAE research project 1118 TRP final report, Atlanta: ASHRAE.

Kersten, M.S. 1949. *Laboratory research for the determination of the thermal properties of soils, engineering experiment station*. Contract Report for the US Army Corps of Engineers, St. Paul District, University of Minnesota.

Long, C. 2008. US water pipelines are breaking. *Associated Press*, April 8.

Lunardini, V.J. 1978. Theory of n-factors and correlation of data. *Proceeding of the Third International Conference on Permafrost*, Edmonton, Alberta. National Research Council of Canada publication no. 16529.

Lunardini, V.J. 1981. *Heat Transfer in Cold Climates*. New York: Van Nostrand Reinhold.

McCabe R.E., J.J. Bender, and K.R. Potter. 1995. Subsurface ground temperature—Implications for a district cooling system. *ASHRAE Journal* 37(12):40–45.

Minkowycz, W.J., E.M. Sparrow, G.E. Schneider, and R.H. Pletcher. 1988. *Handbook of numerical heat transfer*. New York: John Wiley & Sons.

NACE. 2007. SP0169, *Control of external corrosion on underground or submerged metallic piping systems*. Houston: NACE International.

Nayyar, M. 2000. *Piping handbook*, 7th ed. New York: McGraw-Hill Book Co.

Phetteplace, G., and V. Meyer. 1990. *Piping for thermal distribution systems*. CRREL Internal Report 1059. Hanover, NH: U.S. Army Cold Regions Research and Engineering Laboratory.

Phetteplace, G., P. Mildenstein, J. Overgaard, K. Rafferty, D.W. Wade, I. Oliker, P.M. Overbye, V. Meyer, and S. Tredinnick. 2013. *District heating guide*. Atlanta: ASHRAE.

Rao, S.S. 1982. *The finite element method in engineering*. New York: Pergamon.

Revie, R. W, Editor. 2015. *Oil and gas pipelines: Integrity and safety handbook*. Hoboken, NJ: John Wiley & Sons, Inc.

Sicaras, V. 2007. Every drops counts. *Public Works*. March 12.

Sperko, W. 2009. Dissimilar metals in heating and AC piping systems. *ASHRAE Journal* 51(4):28–32.

Twedt, S. 2002. A sea of drinking water lost between treatment, tap. *Pittsburgh Post-Gazette*, July 15.

# 5
# End User Interface

## TEMPERATURE DIFFERENTIAL CONTROL

The chilled water (CHW) energy that is produced at the central plant is transported via the distribution network and is finally transferred to the connected buildings or to consumers. The district cooling interconnection to the building has been called many different terms, such as the energy transfer station (ETS), the end user interface, or a customer/consumer interface; however, the purpose is the same—to transfer energy and custody of the chilled water from the provider to the customer. The consumer interconnection to the system is a critical aspect of district cooling that has impacts not only at the consumer building but also far beyond. The success of a DCS is often measured in terms of the temperature differential achieved between supply and return chilled-water temperatures at the central plant. Generally, maintaining a high temperature differential ($\Delta T$) between supply and return lines is most cost-effective because it allows smaller pipes to be used in the distribution system and may also reduce pumping energy consumed.

Largely, the consumer interconnection and the in-building equipment at the consumer's location determine the $\Delta T$ between supply and return CHW at that connection. Low $\Delta T$ is a chronic problem in DCSs and has earned the moniker "low $\Delta T$ syndrome." Low $\Delta T$ may be the Achilles' heel of a DCS that is otherwise well designed and operated. For this reason, the consumer's building HVAC equipment and the interconnection with the DCSs should be afforded much more attention in design than might initially be apparent. Successful operation of a DCS will normally require that the customer's $\Delta T$ be monitored, controlled, and optimized in most circumstances.

When DCSs are connected to existing buildings, the HVAC equipment within those buildings will often place severe limitations on the ability of the DCSs to achieve adequate $\Delta T$ at that connection while maintaining acceptable comfort within the building. Retrofit and potential optimization of building HVAC equipment to achieve acceptable $\Delta T$ may be expensive, but should be fully considered in the context of the impacts on $\Delta T$. The savings possible from increased $\Delta T$ must be weighed against any higher building conversion costs that may result. For DCSs operating as commercial ventures where consumers are not able to achieve adequate $\Delta T$, tariff structures should be considered that adequately compensate the district cooling utility for the impacts.

When new buildings that will be connected to a DCS are constructed, it is possible to design the in-building HVAC equipment to achieve acceptable $\Delta T$s by following the guidance in this chapter. While the solution to achieving a high $\Delta T$ is similar to an existing building connection, the additional costs for doing so in a new building will not be as onerous; however, it may be difficult to convince the building owner and/or designer to make the necessary modifications to the conventional design used with in-building CHW generation. The district cooling utility/proponent must be prepared to make additional effort to convince others of the necessity of designing the building for district cooling service.

To optimize the $\Delta T$ and meet the customer's CHW demand, the flow from the central plant should be varied. Varying the flow also saves pump energy. CHW flow in the customer's side must be varied and will be discussed later in this chapter. Terminal units in the building connected to the CHW loop (i.e., air-handling units, fan-coils, etc.) may require modifications (e.g., changing three-way valves to two-way, etc.) to operate with variable water flow to ensure a maximum return-water temperature.

For cooling coils, six-row 12–14 fins-per-in. (5–6 fins-per cm) coils are the minimum size coil applied to central station air-handling units to provide adequate performance. With this type of coil, the $\Delta T$ should range from 12°F–16°F (6.7°C–8.9°C) at full load. Coil performance at reduced loads should be considered as well, and fluid velocity in the tube should remain high to stay in the turbulent flow range. To maintain a reasonable temperature differential at design conditions, fan-coil units should be sized for an entering-water temperature several degrees above the main CHW plant supply temperature. Often, the most cost-effective retrofit is to replace the CHW control valve with a pressure independent control valve (PICV). PICVs are essentially two valves in one, combining a pressure-reducing device with a flow throttling valve.

A summary of additional suggestions that will aid in achieving a high $\Delta T$ is provided below. Some have been discussed above and others are discussed later in this chapter:

- Use variable district side flow
- Use variable customer side flow
- Provide cooling coils with a minimum of 6 rows and 8–10 fins-per-in. (3–4 fins-per-cm)
- Size fan-coils for 2°F–3°F (1.1°C–1.7°C) above the main CHW supply temperature
- Eliminate three-way valves from building terminal equipment to maximum extent possible and use high quality two-way PICVs
- Use direct connections at the building where possible
- Size heat exchangers for low approach temperatures (2°F [1.1°C]), where a customer requires low supply temperature
- Where a CHW demand control valve is used, the valve should be a high quality (industrial) valves capable of control and positive shut-off under the highest expected pressures

## CONNECTION TYPES

CHW may be used directly by the building HVAC system or process loads or indirectly via a heat exchanger that transfers energy from one media to another. When CHW is used directly, it may be reduced in pressure commensurate with the buildings' systems. The design engineer must perform an analysis to determine which connection type is best. Table 5.1 outlines the relative merits of direct versus indirect connections; additional details are contained in the following sections devoted to each type of connection, and an overview of various types of connections and their control is provided by Rishel (2007).

For commercially operated systems, a contract boundary or point of delivery divides responsibilities between the energy provider and the customer. This point can be at a

**Table 5.1** Relative Merits of Direct and Indirect Consumer Interconnection

| Issue | Direct Connection | Indirect Connection |
|---|---|---|
| Water Quality | DCS water is exposed to a building system which may have lower levels of treatment and filtering. Components within existing building systems may have scale and corrosion. | Water quality of the DCS is isolated from building system and can be controlled. |
| Water Consumption | Leakage and consumption of DCS water within the building may be difficult to control and correct. | Water leakage is within the control of the district heating utility. |
| Contractual | Demarcation of consumer's building system may not be clear. | Clear delineation between the consumer and district cooling utility equipment. |
| Cost | Generally lower in overall cost due to the absence of a heat exchanger and possible deletion of building pumps and controls. | Higher cost due to a heat exchanger and additional controls. |
| Reliability | Failures within the building may cause problems or potentially even outages for the district system. | The DCS is largely isolated of any problems in the building beyond the interconnection. |
| Pressure Isolation | Building systems may need to be protected from higher pressure in a DCS or for tall buildings, a DCS may be subjected to higher pressures by the building system. | The heat exchanger provides isolation from building system pressure from the DCS pressure and each may operate at their preferred pressures without influence from the other. |
| $\Delta T$ | Potential for greater $\Delta T$ due to absence of heat exchanger. | Approach temperature in heat exchanger is a detriment to $\Delta T$. |
| In-building Space Requirements | Low space requirements. | Additional space required for heat exchanger and controls. |

piece of equipment, as in a heat exchanger with an indirect connection, or flanges as in a direct connection. A chemical treatment analysis must be performed (regardless of the type of connection) to determine the compatibility of each side of the system (district and consumer) prior to energizing. Cathodic isolating flanges should also be provided; normally these flanges would be located at the point where the CHW distribution system first enters the building. This is especially important where the CHW distribution system has a cathodic protection system. For more information on corrosion and cathodic protection, see Tredinnick (2008) and Sperko (2009).

## Direct Connection

Because a direct connection offers no barrier between the district water and the building's own system (e.g., air-handling unit cooling coils, fan-coils, process loads), the water circulated at the district plant has the same quality as the customer's water. Direct connections, therefore, are at a greater risk of incurring damage or contamination based on the poor water quality of either party. Typically, district systems have contracts with water treatment vendors or trained in-house staff and monitor water quality continuously. This may not be the case with all consumers. A direct connection is often more economical than an indirect connection because it is not burdened by the installation of heat exchangers, additional circulation pumps, or water treatment systems; therefore, investment costs are reduced and return temperatures identical to design values are possible (i.e., there is no loss of $\Delta T$ in a heat exchanger). In general, a direct connection should be considered when a number of the following parameters are true: the building owner is the DCS owner or they are related entities, first cost control is important; buildings are generally low rise in nature; building systems are new or in good condition; in-building space for the interconnection is limited; and the building owner, if different than the district cooling utility, respects the need for high $\Delta T$ and will maintain the building systems accordingly and retrofit the building equipment where necessary to achieve adequate $\Delta T$.

A form of a direct connection is shown in Figure 5.1. This traditional connection includes secondary or tertiary pumping within the building with a variable-speed drive controlled by a differential pressure sensor at a location that is representative of the most hydraulically remote point; hence, if the differential pressure sensor is satisfied, then all coils are satisfied. In this example, the consumer's return temperature is being controlled. With this type of control, it may also be necessary to provide an override control on the consumer supply temperature in order to maintain adequate humidity control in some circumstances. A thermostatic control valve is used to control each individual in-building terminal unit (cooling coil control valve, fan-coil control valve, etc.).

It is possible to have an interconnection without secondary or tertiary pumping or an ETS return temperature control valve such as shown in Figure 5.2. The ETS return temperature control valve is not required if the return temperature from the building meets the system design parameters. It is a more simple and holistic approach to ETS connections and relies on the individual thermostatic control valves at each individual in-building terminal unit for ultimate control. This type of connection is the Holy Grail of ETS designs; if it

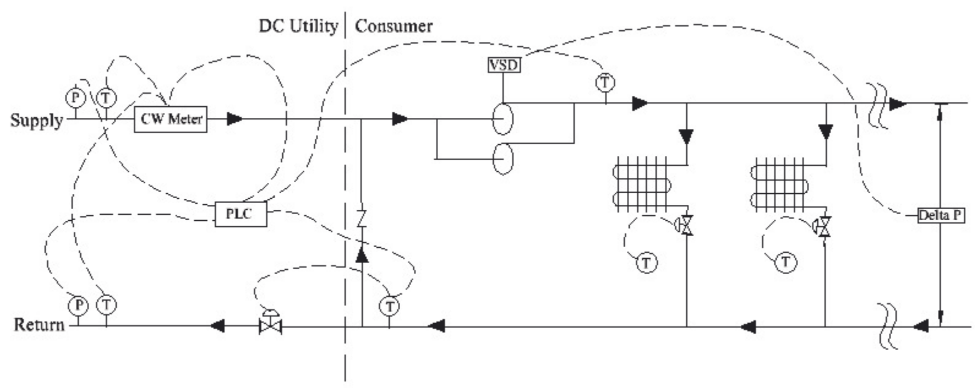



**Figure 5.1** Direct consumer interconnection with in-building pumping.

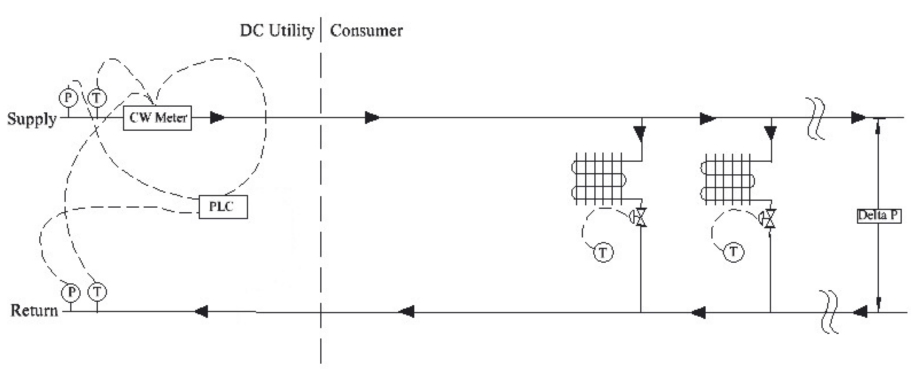



**Figure 5.2** Direct consumer interconnection without in-building pumping or $\Delta T$ control.

works correctly, it is the most efficient and cost-effective design. However, it is only successful if the building EOR has done a superb job of ensuring proper coil and control valve selections at the terminal units. Hence, a connection of this type leaves the district cooling utility entirely at the mercy of the consumer's equipment with respect to $\Delta T$ control. If the in-building heat exchangers are undersized or not the proper configuration (i.e., does not have enough rows of coils), adequate $\Delta T$ may not be achieved. If the cooling coil control valve is not properly sized or is a 3-way type, excess flow will be present, which will cause degradation of CHW $\Delta T$. This can occur even in an otherwise well designed and balanced system when loads less than the design load are encountered, as discussed later. There are many caveats for this type of connection with a big concern being that each terminal unit control valve must be able to close-off tightly against the significant system pressure that the DCP distribution pumps will create. This is not only specific to the valve but also its actuator. Typical commercial-grade terminal unit control valves and actuators may only close-off against 40 to 50 psid (2.8 to 3.4 bar); therefore, it is up to the building EOR to design the system with this in mind.

As shown in Figures 5.1 and 5.2, most commercial systems have a flowmeter installed as well as temperature sensors and transmitters to calculate the energy used; metering is discussed later in this chapter. Temperature and pressure indicators should always be installed at the location of the consumer interconnection. In addition, the district cooling provider typically will have access to the ETS equipment and will monitor the operation from the DCP; instrumentation and control at the consumer's location are discussed in more detail later in this chapter.

With a direct connection, particular attention must be paid to connecting high-rise buildings because they induce a static head. Pressure-control devices should be investigated carefully. It is not unusual to have a DCS with a mixture of direct and indirect connections in which heat exchangers isolate the systems hydraulically for the taller buildings.

In a direct system, the pressure in the main distribution system must meet local building codes to protect the customer's installation and the reliability of the district system. To minimize noise, cavitation, and control problems, constant-pressure differential control valves could be installed in the buildings. Because the pressure energy consumed by the control valves can be extreme, noise can become an issue, hence special attention should be given to potential noise problems at the control valves. A proper valve selection will reveal the calculated noise level of the valve. These valves must correspond to the design pressure differential in a system that has constantly varying distribution pressures because of load shifts. Similar to steam pressure-reducing stations, multiple valves may be required in order to serve the load under all flow and pressure ranges. Industrial quality valves and actuators should be used for this application due to the high pressure drops, turndown, and controllability requirements.

If the temperature in the main distribution system is lower than that required in the consumer cooling systems, a larger temperature differential between supply and return occurs, thus reducing the required pipe size. In the connection type shown in Figure 5.1, the consumer's desired supply temperature can be attained by mixing the return water with the district cooling supply water. Depending on the size and design of the main system, elevation differences, and types of customers and building systems, additional safety equipment, such as automatic shutoff valves on both supply and return lines, may be required.

When buildings have separate circulation pumps, primary/secondary piping, and pumping-isolating techniques are used (cross-connection shunts between return and supply piping, decouplers, and bypass lines). This ensures that two-way control valves are subjected only to the differential pressure established by the customer's building (tertiary) pump. Figure 5.3 shows a connection using an in-building pumping scheme with a $\Delta T$ control valve (i.e., return water temperature control valve).

The use of the $\Delta T$ control valve is typically controversial during the design of DCS and contract negotiations with customers because it benefits the DCP but may negatively affect the customer's CHW supply temperature. Ideally, if the customer's CHW temperature was returned per contract requirements, the $\Delta T$ control valve would not be required. Unfortunately, this does not occur often in reality due to the customer's HVAC system design and operation; hence, a control device is required to actively manage the energy transferred. In order to placate all parties, the $\Delta T$ control valve has several operating scenarios.

The $\Delta T$ control valve's primary function is to provide the CHW supply temperature that the building requires per the contract and is controlled from a temperature transmitter in the customers CHW supply piping. The secondary control function of the $\Delta T$ control valve is to ensure that the customer's return water temperature is also per the contract. If the CHW is returned too cool or below the contracted value (as sensed by a temperature transmitter in the customer's CHW return pipe), the $\Delta T$ control valve will throttle back and send a portion of the return water into the supply through the decoupler until the district return water temperature reaches the desired setpoint. This recirculation action will increase the customer's supply water temperature due to the blending of return water. Under most part-load conditions this is acceptable, but not in hot and humid climates because there is a danger of the supply water temperature being too high to provide dehumidification. Hence, an upper limit set point must be programmed that will only allow the customer's supply water to increase to a point where dehumidification is lost, usually around 48°F (8.9°C).

The bottom line is that the customer must take an active role in ensuring that their water is returned to the DCP per the contract requirements. This means that the terminal unit control valves must be two-way design and of a high quality (pressure independent or

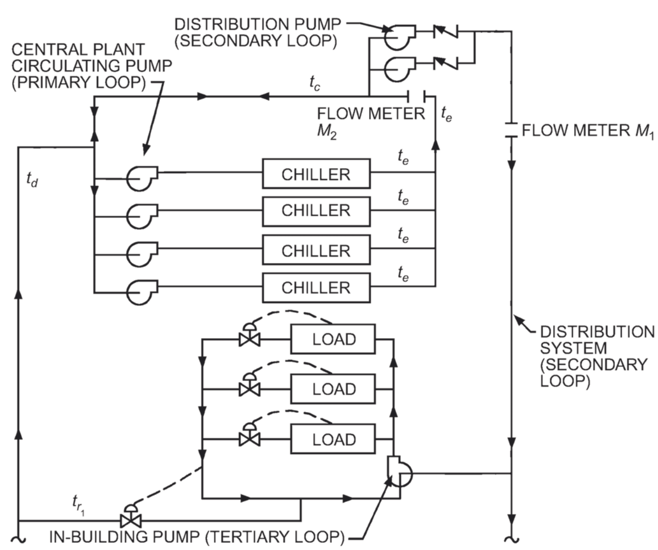



**Figure 5.3** Direct connection with in-building and primary-secondary pumping of district cooling CHW.

characterized ball valve) to be able to throttle the flow through the cooling coils while maintaining control valve authority (see Hegberg 2000 and Hegberg and Hegberg 2015). Furthermore, all bypasses within the customer's chilled-water system must be removed including a 3-way valve at the end of the system to prevent pump dead-heading. There are several alternative methods to perform the same function, including 1) adding a bypass two-position valve around the pump tied to the pump's VFD low-speed signal or 2) adding a constant flow control valve at the end of the system that bypasses water at a specific pressure differential at low load but is closed during all other load conditions.

When tertiary (i.e., in-building) pumps are used, all series connections between the district system pumps should be removed or closely scrutinized for proper pump controls. Without proper pump control, a series connection may cause the district system return to operate at a higher pressure than the distribution system supply and disrupt normal flow patterns. Series operation usually occurs during improper use of 3-way mixing valves in the primary to secondary connection.

## Indirect Connection

Many of the components are similar to those used in the direct connection applications with the exception that a heat exchanger performs one or more of the following functions: heat transfer, pressure interception, and a buffer between potentially different qualities of water treatment. Figure 5.4 shows a form of an indirect connection.

Identical to the direct connection, the rate of energy extraction in the heat exchanger is governed by a control valve that reacts to the building load demand. Once again, the control valve usually modulates to maintain a temperature setpoint on either side of the heat exchanger depending on the contractual agreement between the consumer and the producer. In Figure 5.4, the return temperature of the district cooling network water is being controlled primarily; however, an override is provided if the supply temperature of

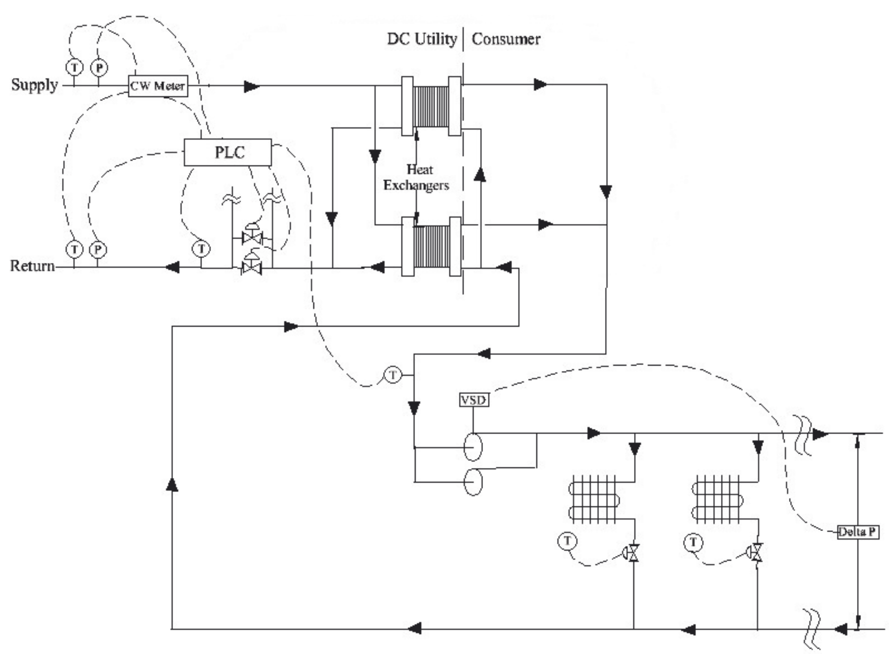



**Figure 5.4** Indirect connection of a building to a DCS.

the water on the consumer side becomes too high to provide adequate comfort or dehumidification.

The three major advantages of using heat exchangers are 1) the static head influences of a high-rise building are eliminated, 2) the two water streams are separated, and 3) consumers must make up all of their own lost water and chemicals used in its treatment. The primary disadvantages of using an indirect connection are 1) the additional cost of the heat exchanger, 2) the increase in supply temperature to the consumer due to the heat-exchanger approach temperature limitation, and 3) the increased pumping pressure due to the addition of another heat transfer surface.

Whether direct or indirect connections are used, many times for critical customers that cannot be without cooling, emergency connections are extended from the district side to an outside wall or service yard of the customer's building to provide a quick interconnection to the building to provide chilled water from mobile rental equipment.

## COMPONENTS

### Heat Exchangers

Heat exchangers, as shown in Figure 5.4, typically act as the line of demarcation between ownership responsibility of the different components of an indirect system. The heat exchangers transfer thermal energy and act as pressure interceptors for the water pressure in high-rise buildings. They also keep fluids from each side (that may have different chemical treatments) from mixing.

Reliability of the installation is increased if multiple heat exchangers are installed. The number selected depends on the types of loads present, the magnitude and shape of the loads, and how those loads vary throughout the year. When selecting all equipment for the building interconnection, but specifically heat exchangers, the designer should:

- Size the unit's capacity to match the given load and estimated load turndown as closely as possible. Oversized units may not perform as desired at maximum turndown; therefore, several smaller units will optimize the installation. However, it must also be considered that multiple units will cost more and require more floor space.
- Verify existing building design CHW temperature relative to the planned district CHW delivery temperatures. If CHW temperature reset will be used, heat exchangers may require rerating at a higher district CHW supply temperature at the reduced load during off-peak hours.
- Assess the critical nature of the load/operation/process to address reliability and redundancy. For example, if a building has 24-hour process loads (i.e., computer-room cooling, water-cooled equipment, etc.), consider adding a separate heat exchanger for this load. Also, consider operation and maintenance of the units.
- If the customer is a hotel, hospital, casino, or data center, select a minimum of two units of at least 50% load each to allow one unit to be cleaned without interrupting building service. For customers with critical loads, multiple heat exchangers should be considered with each having the capacity to fully meet the load when one unit is out of service. Separate heat exchangers should be capable of automatic isolation during low-load conditions to increase part-load performance. Isolating a redundant heat exchanger at part-load will also keep the internal velocity higher and mitigate some internal fouling of the unit.
- Determine the customer's temperature and pressure design conditions. Some gasket materials for plate heat exchangers (PHEs) have low pressure and temperature limits.

- Investigate if existing building pumps and motors can be reused or if new pumps and motors must be installed.
- Consider operation and maintenance requirements. Since PHEs require tipping the plates to remove them, adequate clearance must be available on the sides of the units to remove/add plates for cleaning or capacity growth. Multiple units may share this clearance to economize on the installation footprint. A general rule of thumb is 150% of the plate heat exchanger width.
- Select the heat-exchanger approach. If the customer requires a low water-supply temperature within 2°F (1.1°C) of the district system supply temperature, then select a heat exchanger with such an approach. Otherwise, choose a 3°F or 4°F (1.7°C or 2.2°C) approach because the equipment is smaller and less expensive. In any case, the benefits to the customer and the owner should be compared to the construction cost to determine if the benefits justify the additional cost of a close approach heat exchanger.
- Evaluate customer's water quality (i.e., use appropriate fouling factor).
- Determine available space and structural factors of the mechanical room.
- Recognize that removable insulated housings and drip trays for PHEs are desirable due to the ease of disassembly and reassembly for periodic maintenance requirements.
- Consider using port strainers or automatic backflushing strainers on PHEs to mitigate clogging plates and extend the duration between full disassembly cleaning events.
- Verify existing building system design pressure classification. If the building is above 20 floors tall, the design pressure may exceed 150 psig (10.34 bar); therefore the heat exchanger and appurtenances must be designed for this higher pressure class.
- Calculate the allowable pressure drop on both sides of heat exchangers. The customer's side is usually the most critical for pressure drop. The higher the pressure drop, the smaller and less expensive the heat exchanger. However, the pressure drop must be kept in reasonable limits (15 psig [1.0 bar] or below) if existing pumps are to be reused in retrofit situations. Investigate the existing chiller evaporator pressure drop in order to assist in this evaluation.

All heat exchangers should be sized with future expansion in mind. When selecting heat exchangers, be cognizant that closer approach temperatures or low pressure drops require more heat transfer area, and hence cost more and take up more space. Strainers should be installed in front of any heat exchanger and control valve to keep debris from fouling surfaces.

### PHEs

Plate heat exchangers (PHEs) are the most common type of heat exchanger used in DCSs. Normally, shell-and-tube or shell-and-coil heat exchangers are not able to achieve low enough approach temperatures within space constraints to be suitable for district cooling applications (Skagestad and Mildenstein 2002). PHEs are available as gasketed units and in two gasket-free designs (brazed and all-welded or semi-welded construction). All PHEs consist of metal plates compressed between two end frames and sealed along the edges. Alternate plates are inverted and the gaps between the plates form the liquid flow channels. Fluids never mix as DCS water flows on one side of the plate and the consumer's water flows countercurrent on the other side. Ports at each corner of the end plates act as headers for the fluid. One fluid travels in the odd-numbered plates and the other in the even-numbered plates.

Because PHEs require turbulent flow for good heat transfer, pressure drops may be higher than that for a comparable shell-and-tube model. High efficiency leads to a smaller package. The designer should consider specifying that the frame be sized to hold 20% additional plates. PHEs require very little maintenance because the high velocity of the fluid in the channels tends to keep the surfaces clean. PHEs generally have a cost advantage and require one-third to one-half the surface required by shell-and-tube units for the same operating conditions. PHEs normally achieve closer approach temperatures.

Gasketed PHEs (also called plate-and-frame heat exchangers) consist of a number of gasketed embossed metal plates bolted together between two end frames. Gaskets are placed between the plates to contain the two media in the plates and to act as a boundary. Gasket failure will not cause the two media to mix; instead the media will leak to the atmosphere. Gaskets can be either glued or clip-on. Designers should select the appropriate gasket material for the design temperatures and pressures expected. Plates are typically stainless steel; however, plate material can be varied based on the chemical makeup of the heat transfer fluids. For maintenance recommendations for PHEs, see Tredinnick (2010).

Gasketed PHEs are typically used for district cooling with water and cooling tower water heat recovery (free cooling). PHEs have three to five times greater heat transfer coefficients than shell-and-tube units and are capable of achieving 1°F (0.56°C) approach temperatures. This type of PHE can be disassembled in the field to clean the plates and replace the gaskets. The gaskets can be either glued on or clipped on.

Brazed PHEs feature a close approach temperature (within 2°F [1.1°C]), large temperature drop, compact size, and a high heat transfer coefficient. Construction materials are stainless steel plates and frames brazed together with copper or nickel. Tightening bolts are not required as in the gasketed design. These units cannot be disassembled and cleaned; therefore, adequate strainers must be installed ahead of an exchanger, and it must be periodically flushed clean in a normal maintenance program. Brazed PHEs typically have a peak capacity of under 2,500,000 Btu/hr (about 220 plates and 120 gpm [7.6 L/s]).

## Heat-Exchanger Load Characteristics

In order to provide high $\Delta T$ under multiple load conditions, variable flow is required on both sides of the heat exchanger (Skagestad and Mildenstein 2002; Tredinnick 2007; Perdue and Ansbro 1999). Without variable flow on the customer side, it becomes necessary to flow more water on the DCS side at conditions of reduced load. This condition results in both increased pumping for the district cooling utility as well as reduced $\Delta T$. In addition, the customer side also experiences increased pumping costs without the use of variable flow. The specific degradation in $\Delta T$ and the increases in flow depend on the actual heat-exchanger selection and can easily be determined for a specific heat exchanger by use of selection and sizing software available from the heat-exchanger manufacturer. An example provided by Skagestad and Mildenstein (2002) for a 427 ton (1500 kW) design load indicates that at 50% load, for constant flow on the consumer side, 75% of the design flow would be required on the DCS side; whereas, with variable flow on the consumer side, the required flow on the district cooling side would be reduced to 45% of the design. In addition, the $\Delta T$ in this case is reduced from the design value of 15°F (8.3°C) to just 10°F (5.6°C) at 50% load with constant flow on the consumers side. For this case when variable flow is used on the consumer's side of the PHE, the $\Delta T$ is actually increased from the design value of 15°F to 16.7°F (8.3°C to 9.3°C).

Another example of the need for variable-flow pumping on the consumer's side of PHE is provided by Tredinnick (2007) for a 500 ton application. In Figure 5.5, the consumer side of the heat exchanger has constant flow with a design supply temperature of 42°F (5.6°C) on the consumer's side. The PHE has been sized such that at 100% of design load, the district cooling return temperature will be 54°F (12.2°C); thus, a $\Delta T$ of 14°F (7.8°C) will be achieved at this maximum load condition assuming a 2°F (1.1°C) approach temperature. However, with constant flow on the consumer side of the PHE at 50% of the design load, over 83% of the peak design flow on the district cooling side will be required and the district cooling return temperature will have decreased to 49.6°F (9.8°C), and thus the $\Delta T$ on the district cooling side will have decreased to 9.6°F (5.3°C).

Figure 5.6, also from Tredinnick (2007), illustrates the situation under identical conditions but with variable flow on the consumer side of the PHE. As before, with the consumer-side design supply temperature of 42°F (5.6°C), and at 100% design load, the PHE has been sized to yield a district cooling return temperature of 54°F (12.2°C); thus, at the design condition, the $\Delta T$ will be 14°F (7.8°C), again assuming a 2°F (1.1°C) approach temperature. However, under the 50% load condition with variable flow on the consumer's side, only 51% of the design flow is required on the district cooling side of the PHE and the return temperature has actually increased to 56°F (13.3°C). Thus, the $\Delta T$ for the DCS has increased from 9.6°F (5.3°C) with constant flow under this load condition to 16°F (8.9°C) for variable consumer side flow.

Variable flow also saves electrical pump energy and aids in controlling comfort. These examples, as well as others (see Perdue and Ansbro 1999), should clarify the need for variable flow on the consumer side of a PHE in an indirect connection of district cooling.

Typical constant-flow systems are found in older buildings and may be converted to simulate a variable-flow system by blocking off the bypass line around the air handler heat-exchanger coil on the three-way control valve. At low operating pressures, this potentially may convert a three-way, bypass-type valve to a two-way modulating shutoff

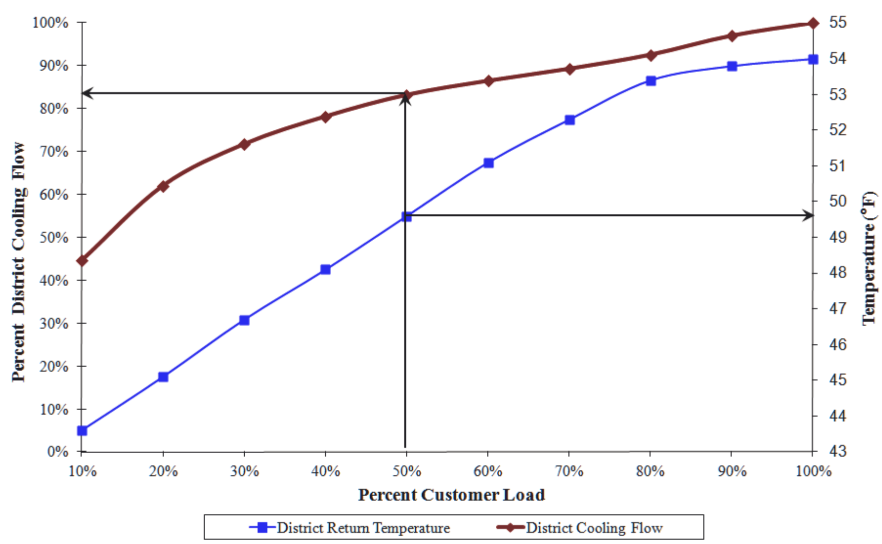



**Figure 5.5** PHE performance with constant flow on the consumer side and a consumer side supply temperature of 42°F (5.6°C) (Tredinnick 2007).

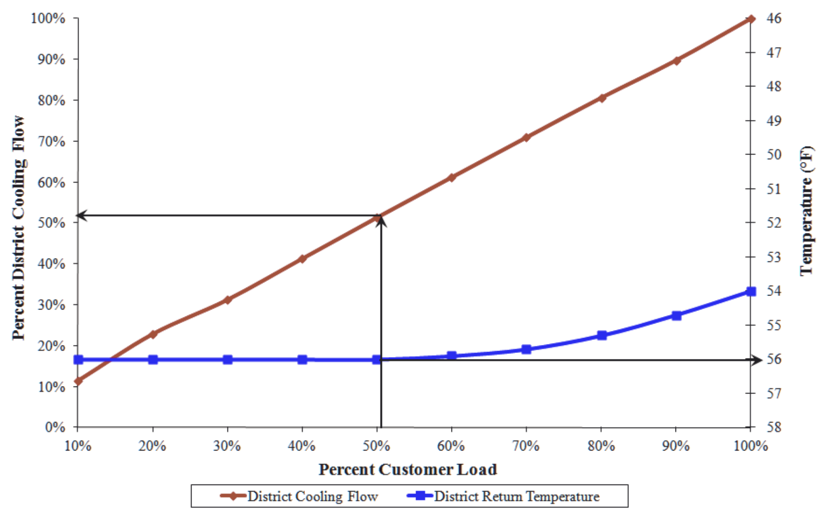



**Figure 5.6** PHE performance with variable flow on the consumer side and a consumer-side supply temperature of 42°F (5.6°C) (Tredinnick 2007).

valve. Careful analysis of the valve actuator must be undertaken because the shut-off requirements and control characteristics are totally different for a two-way valve compared to a three-way valve. More information on building conversion may be found in the work by Skagestad and Mildenstein (2002).

In theory, a partially-loaded cooling coil should have higher return-water temperature than at full load since the coil is oversized for the duty and hence has closer approach temperatures. In many real systems, as the load increases, the return-water temperature tends to rise, and with a low-load condition the supply water temperature rises. Consequently, process or critical humidity control systems may suffer when connected to a system where return-water temperature control is used to achieve high temperature differentials. Other techniques, such as separately pumping each CHW coil, may be used where constant supply water temperatures are necessary year-round.

## Flow Control Devices

In commercial district energy systems, second to the flowmeter, control valves are the most important element within the interface with the district energy system because proper valve adjustment and calibration save energy by ensuring high $\Delta T$ under all load conditions. High-quality, industrial-grade control valves provide more precise control, longer service life, and minimum maintenance.

All control valve actuators should take longer than 60 s to close from fully open to mitigate pressure transients or water hammer, which occurs when valves close more rapidly.

Actuators should also be sized to close against the anticipated system pressure so the valve seats are not forced open, thus forcing water to bypass and degrading temperature differential. Buildings near the central plant may require additional pressure-reducing valves upstream of the control valves.

The wide range of flows and pressures expected makes the selection of control valves difficult. Typically, only one control valve is required; however, for optimal response to load fluctuations and to prevent cavitation, two or more valves in parallel are often needed, as shown in Figure 5.4. The two valves operate in sequence and for a portion of the load (i.e., one valve is sized for two-thirds of peak flow and the other is sized for one-third of peak flow). The designer should review the occurrence of these loads to size the proportions correctly. The possibility of overstating customer loads complicates the selection process, so accurate load information is important. It is also important that the valve selected operates under the extreme pressure and flow ranges foreseen. Because most commercial-grade valves will not perform well for this installation, industrial-quality valves should be specified.

Electronic control valve actuators should remain in a fixed position when a power failure occurs and should be manually operable. Pneumatic control valve actuators should close upon loss of air pressure. A manual override on the control valves allows the operator to control flow with either loss of power or control air. All CHW control valves must fail in the closed position. Thus, when any secondary in-building systems are deenergized, the valves close and will not bypass CHW to the return system. The valves must close slowly as rapid closure at peak load conditions may cause damaging water hammer. Oversizing of control valves reduces valve life and causes valve hunting. Select control valves having a wide range of control; low leakage; and proportional-plus-integral control for close adjustment, balancing, temperature accuracy, and response time. Control valves should have actuators with enough force to open and close under the maximum pressure differential in the system. The control valve should have a pressure drop through the valve equal to at least 10%–30% of the static pressure drop of the distribution system.

This pressure drop gives the control valve the authority it requires to properly control flow. The relationship between valve travel and capacity output should be linear, with an equal percentage characteristic.

In CHW systems, control valves can be installed either in the supply or return line; typically, however, they are installed in the return line to reduce the potential for condensation on any exposed external surfaces and to minimize any water turbulence upstream of the flowmeter.

## Instrumentation and Control

In many systems where energy to the consumer is measured for billing purposes, temperature sensors assist in calculating the energy consumed as well as in diagnosing performance. Sensors and their transmitters should have an accuracy range commensurate to the accuracy of the flowmeter. In addition, pressure sensors are required for variable-speed (VS) pump control or valve control for pressure-reducing stations. Detailed recommendations on pressure, temperature, flow, and power transducers may be found in IDEA (2008). Temperature sensors need to be located by the heat exchangers being controlled rather than in the common pipe. Improperly located sensors will cause one control valve to open and others to close, resulting in unequal loads in the heat exchangers.

Table 5.2 provides a list of common measuring points and derivative parameters for remote monitoring and control of an indirect consumer interconnection. Other measuring points and derivative parameters may be required/recommended where the district heating utility assumes some responsibility for operation of the building systems or the building and DCSs belong to a single owner.

At the consumer's location, the controller performs several functions, including recording demand and the amount of energy used for billing purposes, monitoring the differential pressure for plant pump control, performing energy calculations, alarming for parameters outside normal, and monitoring and controlling all components.

Typical control strategies include regulating district flow to maintain the customer's supply temperature (which results in a fluctuating customer return temperature) or maintaining the customer's return temperature (which results in a fluctuating customer supply temperature). When controlling return flow, the impact on the customer's ability to dehumidify properly with an elevated entering coil temperature should be investigated care-fully. As discussed earlier, proper design and control of the customer's heat transfer coils are necessary to ensure adequate $\Delta T$ for efficient operation of the DCS.

## Temperature Measurement

Temperature measurements should be made with sensors located in wells to allow for change-out. Both resistance temperature detectors (RTDs) and thermistors have been used for temperature sensing elements. Temperature measurement transducers should be specified to provide accuracy of ±1°F (0.56°C) (IDEA 2008). For thermal meters (discussed in the "Metering" section in more detail) temperature measurement may be made by differential methods in order to achieve higher accuracy of the differential temperature which is the parameter of interest rather than the absolute temperature. Furthermore, a matched pair of RTDs should be used with flowmeters when calculating energy usage in order to get a more accurate reading.

## Pressure Measurement

Aside from the customary mechanical pressure gauges that should be provided at the end user interface for on-site diagnostics, pressure transducers are normally provided for remote monitoring and, in many instances, control. Pressure transducers should be specified to provide accuracy of ±1% of full scale and typically resolution of 0.1 psi (6.9 mbar) (IDEA 2008).

**Table 5.2** Measuring Points and Derivative Parameters for Remote Monitoring and Control of an Indirect Consumer Interconnection

| Measured Point/Parameter | Location |
|---|---|
| Temperature | District cooling side supply<br>District cooling side return (Optional)<br>Consumer side supply<br>Consumer side return |
| Pressure | District cooling side supply<br>District cooling side return<br>Consumer side supply<br>Consumer side return |
| Differential pressure | District cooling side at building entrance<br>District cooling side of heat exchanger(s)<br>Consumer side of heat exchanger(s)<br>District cooling side control valve(s)<br>District cooling side strainer |
| Flow rate | District cooling side water |
| Energy transfer | District cooling side water |
| Position of control valve(s) | District cooling side |
| Variable speed drive percentage(s) | Consumer side |

## Pressure-Control Devices

If the water pressure delivered to the customer is too high for direct use, it must be reduced. Similarly, pressure reducing or sustaining valves may be required if building height creates a high static pressure and influences the DCS return-water pressure for buildings directly connected. Water pressure can also be reduced by control valves or regenerative turbine pumps. The risks of using pressure-regulating devices to lower pres-sure on the return line is that if they fail, the entire distribution system (and other directly connected buildings) will be exposed to their pressure and overpressurization will occur.

In high-rise buildings, all piping, valves, coils, and other equipment may be required to withstand higher design pressures. Where system static pressure exceeds safe or economical operating pressure, either the heat- exchanger method or pressure-sustaining valves in the return line may be used to minimize the impact of the pressure. Vacuum vents should be provided at the top of the building's water risers to introduce air into the piping in case the vertical water column collapses.

## METERING

All CHW delivered by a commercially operated DCS to customers or end users for billing or revenue must be metered. For such systems, the meter is the most important component of the end user interface; it is often referred to as the "cash register" of the system as it forms the basis for billing the customers. For DCSs under common ownership for the buildings, distribution system, and central plant, metering is also advisable for diagnostic and optimization purposes. The type of meter selected depends on the accuracy required and the expected turndown of flow to meet the low flow and maximum flow conditions. Typically, the higher the accuracy desired, the higher the cost of the meter. Hence, it is important that the meter is sized accurately for the anticipated loads and not oversized since this will lead to inaccuracies. Historical metered or otherwise benchmarked data should be used when available if the actual load is not accurately known.

For DCSs, energy is calculated by measuring the temperature differential between the supply and return lines and the flow rate. Thermal meters (Btu [kWh]) compensate for the actual volume and heat content characteristics of the CHW. Thermal transducers, resistance thermometer elements, or liquid expansion capillaries are usually used to measure the differential temperature of the energy transfer medium in supply and return lines. Figure 5.7 shows a typical district cooling connection with metering.

Water flow can be measured with a variety of meters, usually pressure differential, turbine or propeller, or displacement meters. Chapter 36 of the 2013 edition of *ASHRAE Handbook—Fundamentals* (ASHRAE 2013), Skagestad and Mildenstein (2002), and Pomroy (1994) have more information on measurement. Ultrasonic meters are sometimes used to check performance of installed meters. Various flowmeters are available for district energy billing purposes. Critical characteristics for proper installation include clearances and spatial limitations as well as the attributes presented in Table 5.3. The data in the table only provide general guidance, and the manufacturers of meters should be contacted for data specific to their products.

High-accuracy meters are desired for more accurate billing of the customers, which benefits both the district cooling provider and the customer by reducing the incidences of disputes over billing.

The meter should be located upstream of the heat exchanger and the control valve(s) should be downstream from the heat exchanger. This orientation minimizes the possible

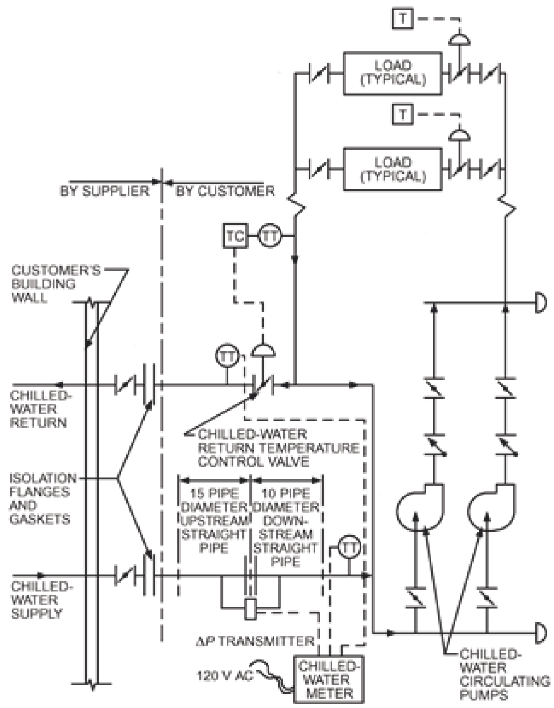



**Figure 5.7** Typical CHW piping and metering diagram.

**Table 5.3** Flowmeter Characteristics

| Meter Type | Accuracy | Range of Control | Pressure Loss | Straight Piping Requirements (Length in Pipe Diameters) |
|---|---|---|---|---|
| Orifice plate | ±1% to 5% full scale | 3:1 to 5:1 | High (>5 psi) | 10 D to 40 D upstream; 2 D to 6 D downstream |
| Electromagnetic | ±0.15% to 1% rate | 30:1 to 100:1 | Low (<3 psi) | 5 D to 10 D upstream; 3 D downstream |
| Vortex | ±0.5% to 1.25% rate | 10:1 to 25:1 | Medium (3 to 5 psi) | 10 D to 40 D upstream; 2 D to 6 D downstream |
| Turbine | ±0.15% to 0.5% rate | 10:1 to 50:1 | Medium (3 to 5 psi) | 10 D to 40 D upstream; 2 D to D downstream |
| Ultrasonic | ±1% to 5% rate | >10:1 to 100:1 | Low (<3 psi) | 10 D to 40 D upstream; 2 D to 6 D downstream |

formation of bubbles in the flow stream and provides a more accurate flow indication. The transmitter should be calibrated for zero and span as recommended by the manufacturer.

Wherever possible, the type and size of meters selected should be standardized to reduce the number of stored spare parts, technician training, etc.

Displacement meters are more accurate than propeller meters, but they are also larger. They can handle flow ranges from less than 2% up to 100% of the maximum rated flow with claimed ±1% accuracy. Turbine-type meters require the smallest physical space for a given maximum flow. However, like many meters, they require at least 10 diameters of straight pipe upstream and downstream of the meter to achieve their claimed accuracy.

The United States has no performance standards for thermal meters, although, efforts are under way to develop an ASTM standard. Where district cooling utilities are regulated by a public utilities commission, many are required to meet an accuracy standard of ±2%

with periodic testing for continued assurance. ASHRAE Standard 125 describes a test method for rating liquid thermal meters (ASHRAE 1992). Several European countries have developed performance standards and/or test methods for thermal meters, and EN 1434 (CEN 2007), developed by the European Community, is a performance and testing standard for heat meters.

District energy plant meters intended for billing or revenue, require means for verifying performance periodically. Major meter manufacturers, some laboratories, and some district energy companies maintain facilities for this purpose. In the absence of a single performance standard, meters are typically tested in accordance with their respective manufacturers' recommendations. Primary measurement elements used in these laboratories frequently obtain calibration traceability to the National Institute of Standards and Technology (NIST).

## REFERENCES

ASHRAE. 1992. ANSI/ASHRAE Standard 125-1992 (RA 2011), *Method of testing thermal energy meters for liquid streams in HVAC systems*. Atlanta: ASHRAE.

ASHRAE. 2013. *ASHRAE handbook—fundamentals*. Atlanta: ASHRAE.

CEN. 2007. EN 1434, *Heat meters—Parts 1–6*. Brussels: European Committee for Standardization.

Hegberg, M.C. 2000. "Control valve selection for hydronic systems." *ASHRAE Journal* (November):33–39.

Hegberg, M.C., and R.A. Hegberg. 2015. *Fundamentals of water system design*, 2d ed. Atlanta: ASHRAE.

IDEA. 2008. *District cooling best practices guide*. Westborough, MA: International District Energy Association.

Perdue, S., and J. Ansbro. 1999. Understanding design implications of constant vs. variable flow pumping on plate heat exchangers. *Proceedings of IDEA District Cooling Conference*, October 7. Atlantic City, NJ.

Pomroy, J. 1994. Selecting flowmeters. *Instrumentation & Control Systems*. Radnor, PA: Chilton Publications.

Rishel, J. 2007. Connecting buildings to central chilled water plants. *ASHRAE Journal* 49(11).

Skagestad, B., and P. Mildenstein. 2002. *District heating and cooling connection handbook*. Sittard, Netherlands: Netherlands Agency for Energy and Environment, an operating agent for the International Energy Agency.

Sperko, W. 2009. Dissimiliar metals in heating and AC piping systems. *ASHRAE Journal* 51(4):28–32.

Tredinnick, S. 2007. Tales of the paranormal: PHE's variable countercurrent flows, and the "X-Files." *District Energy*: Fourth quarter. Westborough, MA: International District Energy Association.

Tredinnick, S. 2008. Opposites attract: The color purple and galvanic corrosion. *District Energy*: Second quarter, Westborough, MA: International District Energy Association.

Tredinnick, S. 2010. Maintaining Plate Heat Exchangers: No need for an idiot light here. *District Energy*: First quarter. Westborough, MA: International District Energy Association.

# 6 Thermal Energy Storage

## OVERVIEW OF TES TECHNOLOGY AND SYSTEMS FOR DISTRICT COOLING

Thermal energy storage (TES) involves the storage of energy over a period of time as heat (or cooling) in a storage medium. Thus, the generation of heat or cooling is separated in time from the use of the heat or cooling. When applied to district cooling systems (DCSs), TES takes the form of cool TES, usually using an insulated tank of water that is cooled or frozen during off-peak periods (i.e., times of low cooling demand), usually nights and/or weekends, and subsequently warmed or melted during on-peak periods (i.e., times of high cooling demand), usually daytime during weekdays. The use of such cool TES in DCSs allows fewer or smaller chillers to operate during on-peak times than would otherwise be necessary to meet peak cooling loads by discharging (heating or melting) TES to meet some or even all of the on-peak district cooling load. Extra chiller capacity operates during the off-peak times to recharge (cool or freeze) the TES, while any off-peak district cooling loads are met simultaneously.

The basic engineering unit of TES capacity is the refrigeration ton-h, equivalent to 12,000 Btu of energy (or the thermal kWh, equivalent to 3413 Btu of energy). One ton-h is equivalent to 3.517 kWh. If used at a constant rate, a 10,000 ton-h TES capacity could, for example, serve a 1000 ton cooling load for 10 h or a 2000 ton cooling load for 5 h, etc. The most common types of TES used in district cooling applications are summarized in Table 6.1, along with their basic performance characteristics.

There are many thousands of successful TES systems operating around the world, including many hundreds in district cooling applications. They include TES capacities ranging from about 200 ton-h to over 700,000 ton-h (with most district cooling applications of TES having TES capacities of 10,000 ton-h to over 100,000 ton-h). TES serves many different types of applications. And TES systems are located in almost all parts of the world and in almost all climate conditions. Finally, they involve the use of several distinct types and subtypes of TES technology.

The types of owners and projects using TES vary widely. They include a variety of applications, each category of which often includes DCSs, i.e., a central cooling system serving multiple buildings or facilities:

- Educational facilities: schools, colleges, and universities
- Healthcare facilities: hospitals, clinics, laboratories, and medical research
- Airports, museums, and sports and entertainment complexes

**Table 6.1** Types of Cool TES and their Basic Characteristics (Typical or Approximate Values)

| | Ice TES | Chilled Water (CHW) TES | Aqueous Low Temp Fluid (LTF) TES |
|---|---|---|---|
| Cooling is stored as | Latent heat (change in phase) | Sensible heat (change in temp) | Sensible heat (change in temp) |
| Cooling storage density | 144 Btu/lb (80 kcal/kg) | 1 Btu/lb·°F (1 kcal/kg·°C) | Near 1 Btu/lb·°F (~1 kcal/kg·°C) |
| Typical TES specific volume | 3 to 4 $ft^3$/ton-h (0.08 to 0.11 $m^3$/ton-h) | 11 to 18 $ft^3$/ton-h (0.3 to 0.5 $m^3$/ton-h) | 7 to 10 $ft^3$/ton-h (0.2 to 0.3 $m^3$/ton-h) |
| Discharge temp from TES | 34°F to 44°F (1°C to 7°C) | 39°F* to 42°F (4°C to 6°C*) | 30°F to 36°F (–1°C to +2°C) |
| Recharge temp to TES | 18°F to 28°F (–8°C to –2 °C) | 39°F* to 42 °F (4°C to 6°C*) | 30°F to 36°F (–1°C to +2°C) |
| Recharge chiller plant power (kW electricity in/kW thermal out) | 0.8 to 1.2 kW/ton (0.23 to 0.34) | 0.6 to 0.7 kW/ton (0.17 to 0.20) | 0.7 to 0.8 kW/ton (0.20 to 0.0.23) |

**The approximate minimum temperature for thermally stratified CHW TES (i.e., the temperature at which maximum density occurs) unless chemical additives are used (as in LTF TES).*

- Government facilities: institutional, military, research, administrative, and correctional facilities, at federal, state, and local levels
- Private industry: commercial and industrial facilities (including aeronautics and aerospace, automotive, computing, data processing, electronics, pharmaceuticals, and telecommunications)
- District cooling utilities: CHW utility systems selling cooling to multiple customers
- Energy services/performance contracting: third-party-financed projects, funded by energy efficiency related cost savings
- Turbine inlet cooling: increasing hot-weather power output and efficiency of combustion turbine power plants

Two recent surveys identified the quantity and demographics of instances of TES use in DCSs—one being a survey of TES use in university or college campus DCSs (Andrepont 2005a) and one being a survey of TES use in thermal utility DCSs (Andrepont 2005b). A summary of the highlights of those surveys is presented in Table 6.2.

The use of district cooling TES systems in hospital/medical, institutional/military, commercial, and other applications has not been quantified, but is known to be extensive and likely similar to the values listed in Table 6.2 for campus and utility DCSs.

Many owners have become repeat customers, using TES at two, three, and even more individual locations. Such repeat customers with multiple TES installations include large, well-known companies and institutions that are household names and respected leaders in their business areas, including the following examples (Andrepont 2018a):

- 3M
- AOL
- Austin Energy
- Bank of America
- Boeing
- California State University (over 260,000 ton-h in 16 TES systems on 14 campuses)
- Dallas/Fort Worth International Airport
- Disney
- District Energy St. Paul
- Dominion Energy

**Table 6.2** Summary of Survey Results of TES Use in Campus and Utility District Cooling Systems

| | TES in Campus District Cooling (Andrepont 2005a) | TES in Utility District Cooling (Andrepont 2005b) |
|---|---|---|
| **Total DC TES** | | |
| Number of TES installations | 159 (at 124 campuses) | 106 |
| Installed TES capacity | 1,808,408 ton-h | 2,610,815 ton-h |
| **Average per DC System** | | |
| Installed TES capacity | 14,584 ton-h | 24,630 ton-h |
| Peak cooling load reduction | 2,083 tons | 3,519 tons |
| Peak electric load reduction | 1.6 MW | 2.6 MW |
| **Chronological Distribution of DC TES** | | |
| During 1981–1985 | 70,000 ton-h | 24,500 ton-h |
| During 1986–1990 | 147,300 ton-h | 102,533 ton-h |
| During 1991–1995 | 571,127 ton-h | 509,373 ton-h |
| During 1996–2000 | 521,024 ton-h | 879,048 ton-h |
| During 2001–2005 | 550,000 ton-h (est.) | 828,169 ton-h |
| **Geographic Distribution of DC TES** | | |
| Inside the USA | 1,677,048 ton-h | 1,883,347 ton-h |
| Outside the USA* | 131,360 ton-h | 727,468 ton-h |
| **Technologic Distribution of DC TES** | | |
| Latent Heat (Ice) TES | 22% of total ton-h | 35% of total ton-h |
| Sensible Heat (CHW and LTF) TES | 78% of total ton-h | 65% of total ton-h |

** Note: Non-US district cooling TES installations may be under reported. But also, there has been rapid growth in non-US installations during 2006–2018, most notably in Middle Eastern and East Asian locales.*

- Florida State University
- Ford
- General Motors
- Honeywell
- IBM
- Los Angeles International Airport
- Lockheed Martin
- Princeton University
- Qatar Cool
- Saudi Electricity Company (SEC) (over 1 million ton-h in turbine cooling systems)
- Siemens
- State Farm
- Tabreed
- Texaco
- Texas Instruments
- Thermal Energy Corporation (Texas Medical Center)
- Toyota
- Trigen Energy
- University of California (nearly 250,000 ton-h in 8 TES systems on 7 campuses)
- University of Maryland
- University of Nebraska
- University of Texas
- UPS
- U.S. Air Force
- U.S. Army
- U.S. Veterans Administration
- Verizon

There are technically and economically successful TES applications in hot and humid climates, such as those found in:

- Rio de Janeiro, Brazil
- Kuala Lumpur, Malaysia
- Doha, Qatar
- Abu Dhabi and Dubai, United Arab Emirates
- Houston, Texas; New Orleans, Louisiana; and Miami and Orlando, Florida, USA

And there are also applications in hot and dry climates, such as those found in:

- Northam and Perth, Western Australia, Australia
- Mexicali, Baja California Norte, Mexico
- Riyadh, Saudi Arabia
- Phoenix, Arizona and El Paso, Texas, USA

There are technically and economically successful TES applications in locales where cooling loads are high all year long, such as Brazil, Malaysia, Mexico, Qatar, Saudi Arabia, the United Arab Emirates, and the southern United States.

But there are also applications where the cooling season lasts only for a portion of the year or even for only a very short portion of the year, such as:

- Edmonton, Alberta, Canada
- Helsinki, Finland
- Tokyo, Japan
- Seoul, South Korea
- Stockholm, Sweden
- Chicago, Illinois and Saint Paul, Minnesota, USA

## TES TECHNOLOGY TYPES

TES technologies for cool storage include two distinct types:

- latent heat storage systems, such as ice TES in which thermal energy is stored as a change of phase of the storage medium, usually between solid and liquid states
- sensible heat storage systems, such as chilled water (CHW) TES and low temperature fluid (LTF) TES, in which thermal energy is stored as a temperature change in the storage medium

### Latent Heat TES

Latent heat TES systems store energy as a change in phase of the storage material. The most common examples are ice storage systems, which use the phase change of water between liquid and solid states. The major benefits of ice storage are its compactness (high energy density per unit volume) and the availability of standardized, modular equipment. The drawbacks of ice storage are that it must be charged using chillers that operate at very low temperatures and thus at high unit-energy consumption; however, a sometimes offsetting benefit is the ability to use ice storage to provide a reduced CHW supply temperature (or even also a reduced air supply temperature) to reduce the size, capital cost, pumping energy, and pumping cost associated with CHW distribution networks (or even to reduce the size, capital cost, fan energy, and fan energy cost associated with air-handlers and air distribution networks). There are many thousands of successful ice TES systems in operation worldwide, primarily in small-to-medium size applications, averaging 2000 to 3000 ton-h (7000 to 111,000 kWh) per installation, but also up to over 200,000 ton-h (700,000 kWh) in a single DCS (for a university in China). See Table 6.3 and Figure 6.1 for some examples of latent heat TES.

**Table 6.3** Some Examples of Latent Heat (Ice) TES in DC Systems (Andrepont 2018a)

| District Cooling System Name—Location | 1st year in Operation | TES Type | Technology Subtype | Capacity (ton-h) |
|---|---|---|---|---|
| Enwave Chicago—Chicago, IL, USA (Plant #2) | 1996 | Ice | Ice-on-coil | 125,000 |
| Enwave Chicago—Chicago, IL, USA (Plant #3) | 1997 | Ice | Ice-on-coil | 97,000 |
| Stanford University—Palo Alto, California, USA | 1997 | Ice | Ice-on-coil | 93,200 |
| Enwave Chicago—Chicago, IL, USA (Plant #1) | 1995 | Ice | Ice-on-coil | 66,000 |
| NRG Energy—Phoenix, Arizona, USA (2) | 2001–2005 | Ice | Ice-on-coil | 56,000 |
| Austin Energy—Austin, Texas, USA | 2005 | Ice | Ice-on-coil | 52,000 |
| Xcel Energy—Denver, Colorado, USA | ca. 1998 | Ice | Ice-on-coil | 37,500 |
| University of Arizona—Tucson, Arizona, USA (2) | 2004–2007 | Ice | Ice-on-coil | 24,000 |
| University of Pennsylvania—Philadelphia, PA, USA | 1993 | Ice | Ice-on-coil | 21,560 |
| Johns Hopkins University—Baltimore, MD, USA | 1993 | Ice | Ice-on-coil | 11,200 |
| George Mason University—Fairfax, Virginia, USA | 1993 | Ice | Ice-on-coil | 7500 |
| Xavier University—Cincinnati, Ohio, USA (2) | 1993–1997 | Ice | Ice-on-coil | 6500 |
| Drexel University—Philadelphia, PA, USA | 2004 | Ice | Ice-on-coil | 4200 |
| Kalamazoo College—Kalamazoo, MI, USA | 1995 | Ice | Ice-on-coil | 4200 |
| University of Miami—Coral Gables, Florida, USA | 1993 | Ice | Ice-on-coil | 3660 |
| Virginia Wesleyan University—Norfolk, VA, USA | 1999 | Ice | Ice-on-coil | 2500 |
| MM21 DC System—Yokohama, Japan | 1994 | Ice | Encapsulated | 35,000 |
| TNEC Bangsar—Kuala Lumpur, Malaysia | 1998 | Ice | Encapsulated | 30,000 |
| Valencia University—Valencia, Spain | 1996 | Ice | Encapsulated | 4300 |
| College of Desert—Palm Desert, CA, USA | 2000 | Ice | Encapsulated | 2500 |
| LES (TIC*)—Lincoln, Nebraska, USA | 1997 | Ice | Ice Harvester | 165,000** |
| SEC (TIC*)—Qaseem, Saudi Arabia | 1998 | Ice | Ice Harvester | 120,000 |
| Nissan—Atsugi-shi, Kanagawa, Japan | 2006 | Ice | Ice slurry | 5000 (est.) |
| Nakanoshima 6 Chome DC—Osaka, Japan | 1980s | Ice | Ice slurry | 3967 |

** Indicates a turbine inlet cooling (TIC) application.*
*** Indicates a weekly (rather than a daily) cycle TES design, resulting in much larger storage.*

## Ice TES Summary

- Energy is stored as latent heat (a change in phase of the storage medium).
- Water is converted to ice during off-peak periods, then ice is melted during on-peak periods.
- Conventional or low temperature supply temperatures are possible, with a range of 34°C to 44°F (1°C to 7°C) being typical.
- Unit storage volumes are relatively compact (typically at approximately 3 to 4 $ft^3$/ton-h [0.024–0.032 $m^3$/kWh]), compared with other TES technologies. However, where a weekly design cycle TES configuration is used (in which the TES is fully recharged each weekend, partially recharged each weeknight, and partially discharged each weekday), the total ice tank volume is typically increased by two to four times versus that of a daily design cycle TES configuration (in which the TES is fully recharged in one night and fully discharged the next day).
- Modular equipment can be a benefit for phased expansions, but it yields relatively little economy-of-scale to benefit the economics of large applications.

Ice TES installations commonly use one of several types of ice TES equipment:

- *Ice-on-coil*. This most prevalent type of ice TES uses heat transfer surfaces, usually coils of pipe or tubing, submerged within a tank of water. To recharge TES, cold fluid (most often water-glycol, but in some cases refrigerant) circulates inside the pipe or tubing, causing ice to form on the outside. To discharge TES, either warm fluid is circulated inside the pipe or tubing, thus melting the ice (known as an internal melt ice-on-coil system), or the warm water flows through the tank in direct contact with the ice thus melting the ice (known as an external melt ice-on-coil system). External melt designs can provide colder discharge temperatures; however internal melt designs can allow ice to be formed completely between adjacent pipes or tubing.
- *Encapsulated ice*. This type of ice TES uses containers (usually small spheres) of water that are stacked within a tank. A heat transfer fluid (usually water-glycol) circulates through the tank and around the water containers. Cold fluid circulates to recharge (freeze) the contained water; subsequently, warm fluid circulates to discharge (melt) the encapsulated ice within the containers. Functionally, the encapsulated ice system behaves similarly to an internal melt ice-on-coil system, as both use a single heat transfer fluid (e.g., water-glycol) to both recharge and discharge TES.
- *Ice harvesters.* This type of ice TES uses vertical heat transfer surfaces (tubes or plates) mounted above an open water tank. Water flows vertically down the heat transfer surfaces, while refrigerant (or coolant) chills the opposite side, causing

**Figure 6.1** The 125,000 ton-h (440,000 kWh) of ice-on-coil TES at Enwave Chicago's District Cooling Plant #2 in Chicago, Illinois, was first in-service in 1996. The 114 ft long × 91 ft wide × 32 ft high (34.7 m × 27.7 m × 9.8 m), 2.5 million gallon (9400 $m^3$) concrete tank, holding 144 ice-on-coil modules, operates at a nominal water-glycol charging (ice building) temperature of 20.5°F (–6.4°C), and a water discharging (ice melting) temperature of 34°F (1.1°C). It can reduce peak demand by up to 14,600 tons (51,000 kW) of cooling and approximately 10 MW of electric power.
*Courtesy of Baltimore Aircoil Company*

ice to form on the water-side surface. Periodically, the cooling is briefly stopped and reversed by flowing hot refrigerant vapor (or warm fluid); this periodic warming causes the ice to release from the heat transfer surface and fall by gravity into the tank below. Subsequently, warm water is returned through the tank of ice and water, thus melting the ice. This technology has seen only limited application in DCSs, and that being primarily in the 1990s for turbine inlet cooling (TIC) applications, which have since shifted largely to CHW TES.

- *Ice slurry*. This type of ice TES produces a (usually pumpable) mixture of ice and water, with or without some additives in the solution. Some systems use falling-film heat exchangers or scraped-surface heat exchangers to create the ice slurry, while another uses a vacuum vessel with direct contact between water vapor, liquid water, and solid ice (all three phases in equilibrium at the triple point of water) to create the ice slurry. In each case the ice-water mixture falls or is pumped into a storage tank for later melting. To date, these technologies have had only limited and relatively small district cooling applications.

For the most prevalent form of ice storage used in DCS, the ice-on-coil (also called external-melt ice storage), the typical operation is as follows:

Normally ethylene-glycol/water chillers are used to charge the TES at night by flowing cold glycol within the serpentine pipes in the TES tank, forming ice on the exterior of the pipes within the water-flooded "shell-side" of the tank outside the piping, see Figure 6.2. The chilled cold water/glycol temperature is normally around 24 to 26 F° (–4.4 to –3.3 °C) at the start of ice-making and must be lowered as ice builds on the coil to account for the thermal resistance of the ice layer; thus when charging ends the glycol/water temperature will be in the 15 to 19 F° (–9.4 to –7.2 °C) range. For discharge (daytime), the water would be drawn from the tank shell-side, pumped to the cooling loads, and returned warm to the tanks to melt the ice. Often, the glycol/water chillers are also operated in the daytime as well (i.e. for a "Partial Shift" TES operation), but at a warmer leaving fluid temperature (to improve efficiency) and then to a glycol-to-water, plate-and-frame heat exchanger to produce the desired supply water temperature, see Figure 6.3. The arrangement can have the TES tank water and the heat exchanger water flowing in parallel, or it can have the heat exchanger upstream of the low temperature water from the ice tanks. This latter arrangement allows the chillers to have their daytime operation at the warmest point, and thus at the best efficiency, while still getting a low supply temperature from the ultimate supply water that leaves the TES tank to the loads.

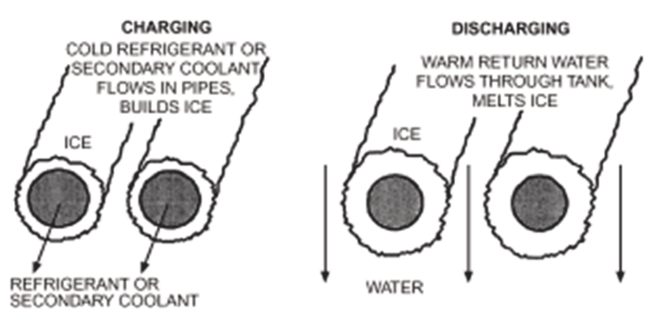



**Figure 6.2** Ice-on-coil TES charging and discharging (ASHRAE 2016).

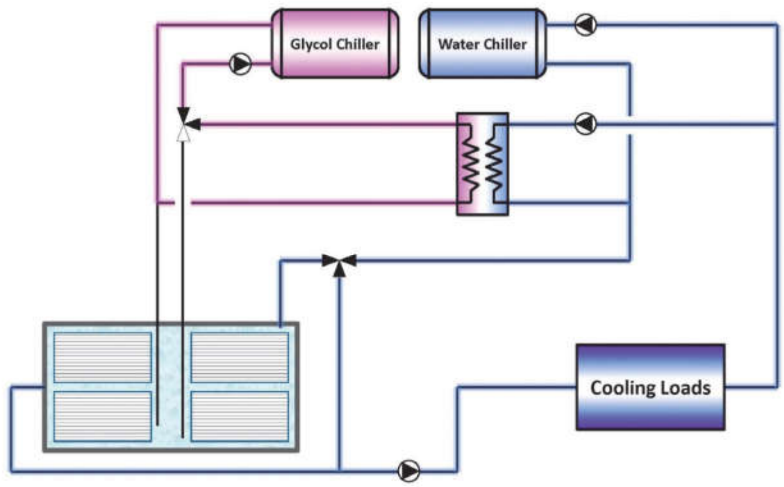



**Figure 6.3** Ice-on-Coil "Partial-Shift" TES with ice melting and water chilling.
*Courtesy EVAPCO, Inc.*

## Sensible Heat TES

These systems store energy as a change in temperature of the storage material. By far the most common examples are thermally stratified CHW storage systems, which use the temperature change of CHW between the cooling system's normal operating supply and return temperatures. The major drawback to CHW storage is the relatively large storage volume required per unit of thermal capacity. The alternative use of stratified aqueous fluid (in lieu of plain water) allows operation at a lower supply temperature (and thus a larger temperature differential between supply and return temperatures) and reduces the necessary storage volume for a given thermal capacity (EPRI 1999). The major benefits of thermally stratified sensible heat storage (whether using CHW or aqueous fluid as the storage medium) include its simplicity of operation and control and its ease of retrofit to existing CHW systems (because CHW TES uses conventional chillers operating efficiently at a constant temperature). Furthermore, sensible heat TES provides a very significant economy-of-scale. Thus, in large capacity applications, such as most DCSs, the capital cost for stratified TES is not only much less than for ice TES, but also much less than equivalent non-TES chiller plant capacity.

## Stratification in CHW TES

*Note: the text of this subsection and its subsections is reproduced nearly verbatim from Chapter 51, "Thermal Storage" in* ASHRAE Handbook—HVAC Systems and Equipment *(ASHRAE 2016), except that the tables and figures have been renumbered and some other minor editorial changes have been made for conformity to the style of this guide.*

Preserving stratification in a CHW TES system is paramount to the most efficient and effective operation of the system. The discussion below (reproduced nearly verbatim from ASHRAE 2016) outlines the characteristics of stratification in CHW TES and outlines the necessary steps to achieve and preserve it.

In stratified cool storage, warmer, less dense return water floats on top of denser CHW. Cool water from storage is supplied and withdrawn at low velocity, in essentially horizontal flow, so buoyancy forces dominate inertial effects. Pure water is most dense at 39.2°F (4.0°C); therefore, colder water introduced into the bottom of a stratified tank

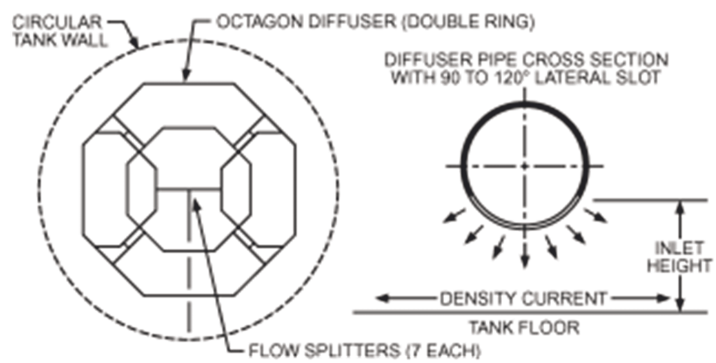
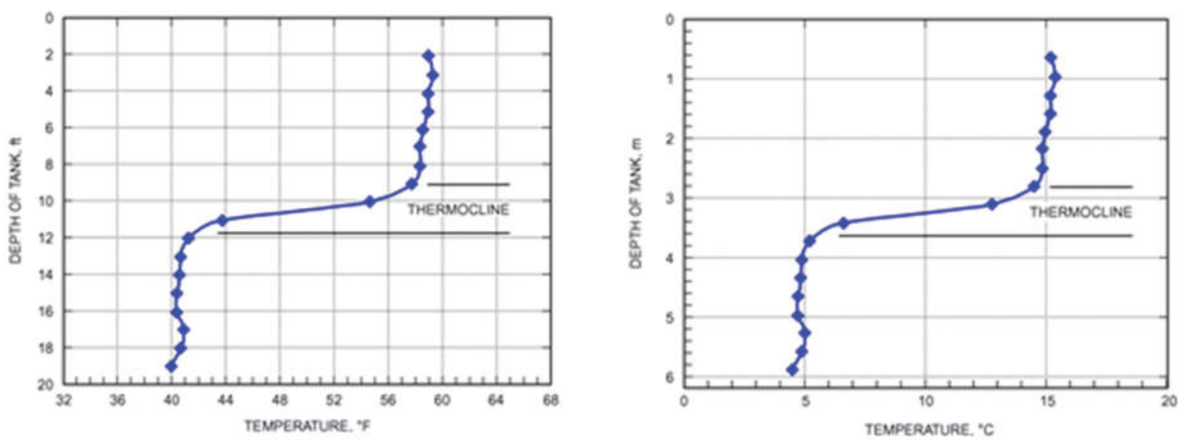



**Figure 6.4** Typical two-ring octagonal slotted pipe diffuser (ASHRAE 2016).




(a) (b)

**Figure 6.5** Typical temperature stratification profile in storage tank, (a) I-P and (b) SI (ASHRAE 2016).

tends to mix to this temperature with warmer water in the tank. However, low-temperature fluids (LTFs), typically water with various admixtures, can be used to achieve lower delivery temperatures, larger temperature differences, and thus smaller storage volumes per ton-hour (megajoule) (Andrepont 2000b; Borer and Schwartz 2005).

When the stratified storage tank is charged, chilled supply water, typically between 38 and 44°F (3 and 6.7°C), enters through the diffuser at the bottom of the tank (Figure 6.4), and the warmer return water exits to the chiller through the diffuser at the top of the tank. Typically, incoming water mixes with water in the tank to form a 1 to 2 ft (0.3 to 0.6 m) thick thermocline, which is a region with sharp vertical temperature and density gradients (Figure 6.5). The thermocline minimizes further mixing of the water above it with that below it. The thermocline rises as charging continues and subsequently falls during discharging. It may thicken somewhat during charging and discharging because of heat conduction through the water and heat transfer to and from the walls of the tank. The initial thermocline formation is the principal determiner for overall tank efficiency: formation of the thinnest possible thermocline provides the greatest usable volume. Therefore, careful consideration of diffuser design is needed. The storage tank may have any cross section, but the walls are usually vertical. Horizontal cylindrical tanks are generally not good candidates for stratified storage, because of the ratio of the volume of water within and outside the thermocline.

## Performance of Chilled-Water Storage Systems

A perfect sensible storage device would deliver water at the same temperature at which it was initially stored. This would also require that water returning to storage neither mix with nor exchange heat with stored water in the tank or the surroundings of the tank. In practice, however, all three types of heat exchange occur.

Typical temperature profiles of water entering and leaving a storage tank are shown in Figure 6.6. Tran et al. (1989) tested several large chilled-water storage systems and developed the *Figure of Merit* (FOM), which is used as a measure of the amount of cooling available from the storage tank. Using the storage profiles of Figure 6.6, the FOM would be:

$$\text{FOM} = \frac{\text{Area between lines A and C}}{\text{Area between lines A and D}} \times 100(\%)$$

Well-designed storage tanks have figures of merit of 90% or higher for daily complete charge/discharge cycles.

## Design of Stratification Diffusers

Diffusers must be designed and constructed to produce and maintain stratification at the maximum expected flow rate through storage. Two main styles are in widespread use: the octagonal pipe diffuser (see Figure 6.4) and the radial disk diffuser (Figure 6.7). Inlet and outlet streams must be kept at sufficiently low velocities, so buoyancy predominates over inertia to produce a gravity current (density current) across the bottom or top of the tank. This relationship between the buoyancy and inertial forces is expressed in a dimensionless form known as the Froude number.

The inlet Froude number $Fr$ is defined as

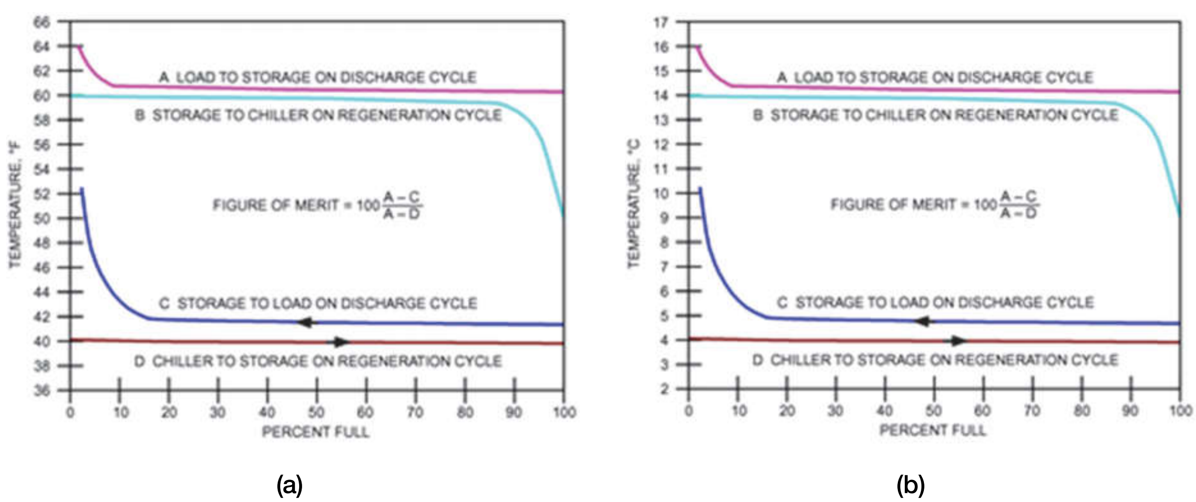



**Figure 6.6** Typical chilled-water storage profiles, (a) I-P and (b) SI (ASHRAE 2016).

$$Fr = \frac{Q}{\sqrt{gh^3(\Delta\rho/\rho)}} \tag{6.1}$$

where

$Q$ = volume flow rate per unit length of diffuser, $ft^3/s \cdot ft$ ($m^3/[s \cdot m]$)

$g$ = gravitational acceleration, $ft/s^2$ ($m/s^2$)

$h$ = inlet opening height, ft (m)

$\rho$ = inlet water density, $lb/ft^3$ ($kg/m^3$)

$\Delta\rho$ = difference in density between stored water and incoming or outflowing water, $lb/ft^3$ ($kg/m^3$)

The density difference $\Delta\rho$ can be determined using the density data contained in Tables 6.4a and 6.4b.

The inlet Reynolds number *Re* is defined as follows:

$$Re = \frac{Q}{\nu} \tag{6.2}$$

where

$\nu$ = kinematic viscosity, $ft^2/s$ ($m^2/s$)

Designers typically select values for *Re*, which determines the diffuser length, then select a value for the diffuser height to create an inlet Froude number of 1.0 or less. However, values up to 2.0 have been successfully applied (Yoo et al. 1986). Some experimental evidence indicates that the intensity of mixing near the inlet diffuser is influenced by the inlet Reynolds number (Equation 6.2).

Wildin and Truman (1989), observing results from a 15 ft (4.6 m) deep, 20 ft (6.1 m) diameter vertical cylindrical tank, found that reducing the inlet Reynolds number from 850 (using a radial disk diffuser) to 240 (using a diffuser comprised of pipes in an octagonal array) reduced mixing to negligible proportions. This is consistent with subsequent results obtained by Wildin (1991, 1996) in a 3 ft (0.9 m) deep scale model tank, which indicated negligible mixing at Reynolds numbers below approximately 450, with the best performance achieved during testing at a Reynolds number of 250.

Bahnfleth and Joyce (1994), Musser and Bahnfleth (1998, 1999), and Stewart (2001) documented successful operation of tanks with water depths greater than 45 ft (13.7 m) for design inlet Reynolds numbers as high as 10,000, although thermal performance of these systems improved at lower inlet Reynolds numbers.

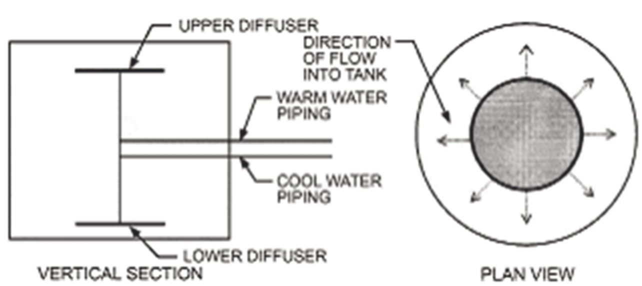



**Figure 6.7** Radial disk diffuser (ASHRAE 2016).

**Table 6.4a** Chilled-Water Density Table (I-P)

| °F | lb/ft$^3$ | °F | lb/ft$^3$ | °F | lb/ft$^3$ |
|---|---|---|---|---|---|
| 32 | 62.419 | 44 | 62.424 | 58 | 62.378 |
| 34 | 62.424 | 46 | 62.421 | 60 | 62.368 |
| 36 | 62.426 | 48 | 62.417 | 62 | 62.357 |
| 38 | 62.427 | 50 | 62.411 | 64 | 62.344 |
| 39 | 62.428 | 52 | 62.404 | 66 | 62.331 |
| 40 | 62.427 | 54 | 62.396 | 68 | 62.316 |
| 42 | 62.426 | 56 | 62.387 | – | — |

**Table 6.4b** Chilled-Water Density Table (SI)

| °C | kg/m$^3$ | °C | kg/m$^3$ | °C | kg/m$^3$ |
|---|---|---|---|---|---|
| 0 | 999.87 | 7 | 999.93 | 14 | 999.27 |
| 1 | 999.93 | 8 | 999.88 | 15 | 999.13 |
| 2 | 999.97 | 9 | 999.81 | 16 | 999.97 |
| 3 | 999.99 | 10 | 999.73 | 17 | 999.80 |
| 4 | 1000.00 | 11 | 999.63 | 18 | 999.62 |
| 5 | 999.99 | 12 | 999.52 | 19 | 999.43 |
| 6 | 999.97 | 13 | 999.40 | 20 | 998.23 |

A parametric study of radial diffusers by Musser and Bahnfleth (2001) found that using Froude numbers less than 1.0 significantly improved performance. Their work, based on field measurements and simulation, confirmed that the Froude number is a parameter of first-order significance for radial diffuser inlet thermal performance, but did not indicate strong effects of varying the *Re*. They found that parameters relating diffuser and tank dimensions (i.e., ratio of diffuser diameter to diffuser height and ratio of diffuser diameter to tank diameter) had a stronger effect on performance and could explain some of the behavior attributed to *Re* by earlier research. A similar study of octagonal diffusers by Bahnfleth et al. (2003a) also indicated that diffuser/tank parameters not previously used in design could be of greater significance than *Re*. They also concluded that diffuser/tank interaction parameters could be important factors in octagonal diffuser performance, and that radial and pipe diffusers are not well described by a single design method.

Because a chilled-water system may experience severe pressure spikes (water hammer), such as during rapid closing of control or isolation valves, the structural design of diffusers should consider this potential event. In addition, the diffusers themselves can also be subjected to buoyancy effects during initial filling or in the rare instance where air entrained in the chilled-water system becomes trapped in the piping and flows to the diffuser. Design and operational considerations should be made to allow any trapped air to escape from the diffuser piping with little or no effect on diffuser performance.

## CHW TES Summary

There are many hundreds of successful stratified TES systems in operation worldwide, primarily in medium to large applications, averaging approximately 20,000 ton-h (70,000 kWh) per installation, and including individual projects with up to 710,000 ton-h (2.5 million kWh) in a single DCSs (for a turbine inlet cooling application in Saudi Arabia). See Table 6.5 and Figure 6.8 for some examples of sensible heat TES. Typical attributes of CHW TES are:

- Energy is stored as sensible heat (a change in temperature of the storage medium).

**Table 6.5** Examples of Sensible Heat (CHW and LTF) TES in DC Systems (Andrepont 2018a)

| District Cooling System Name – Location | First Year in Operation | TES Type | Supply Temp., °F (°C) | Capacity, ton-h |
|---|---|---|---|---|
| SEC PP9 (TIC**)—Riyadh, Saudi Arabia | ca. 2008 | CHW | unidentified | 710,000 |
| Dominion (TIC**)—Emporia, Virginia | 2018 | CHW | 39.3 (4.1) | 268,641 |
| Dominion (TIC**)—Freeman, Virginia | 2016 | CHW | 39.0 (3.9) | 267,800 |
| Dominion (TIC**)—Front Royal, Virginia | 2014 | CHW | unidentified | 232,000 |
| SEC PP8 (TIC**)—Riyadh, Saudi Arabia | 2005 | CHW | 45.5 (7.5) | 192,800 |
| OUCooling—Orlando, Florida | 2003 | CHW* | 40.0 (4.4) | 160,000 |
| Dominion (TIC**)—Pennsylvania | 2009 | CHW | 39.5 (4.2) | 129,000 |
| Princess Noura University—Riyadh, Saudi Arabia | 2010 | CHW | 43.7 (6.5) | 122,000 |
| Calpine (TIC**)—Pasadena, Texas | 1999 | CHW | 39.0 (3.9) | 107,000 |
| Tabreed—Abu Dhabi, UAE (6 systems) | 2001–2004 | CHW | 40.0 (4.4) | 6 × 15,000 |
| State Farm—Bloomington, Illinois (2) | 1994–2000 | CHW | 40.0 (4.4) | 89,600 |
| Dominion (TIC**)—New Canton, Virginia | 2011 | CHW | unidentified | 78,710 |
| Toyota—Georgetown, Kentucky | 1992 | CHW | 44.0 (6.7) | 70,000 |
| Chrysler—Auburn Hills, Michigan | 1990 | CHW | 43.0 (6.1) | 68,000 |
| TECO—Houston, Texas | 2010 | CHW* | 40.0 (4.4) | 64,285 |
| University of Alberta—Edmonton, Alberta, Canada | 2005 | CHW* | 41.0 (5.0) | 60,000 |
| Disney World—Lake Buena Vista, FL | 1998 | CHW | 40.0 (4.4) | 57,000 |
| Florida State U—Tallahassee, FL (2) | 1992–2006 | CHW* | 42.0 (5.6) | 55,209 |
| California DGS—Sacramento, CA | 2010 | CHW | 40.0 (4.4) | 52,000 |
| University of Nebraska—Lincoln City Campus, NE | 2018 | CHW* | 40.0 (4.4) | 52,000 |
| University of Illinois—Champaign, Illinois | 2010 | CHW* | 40.0 (4.4) | 50,000 |
| University of California—Irvine, California | 1996 | CHW | 40.0 (4.4) | 46,150 |
| University of S California—Los Angeles, CA | 2005 | CHW* | 39.0 (3.9) | 45,000 |
| University of N Carolina—Chapel Hill, NC | 2005 | CHW* | 41.0 (5.0) | 40,000 |
| Climaespaco—Lisbon, Portugal | 1997 | CHW | 39.2 (4.0) | 39,807 |
| University of Texas—Austin, Texas (#2 of 2) | 2016 | CHW | 42.0 (5.6) | 39,000 |
| Cornell University—Ithaca, New York | 1991 | CHW | 40.0 (4.4) | 38,000 |
| DESP—St. Paul, Minnesota (#2) | 2003 | CHW* | 40.0 (4.4) | 37,400 |
| California State University—Fullerton, CA | 1993 | CHW | 40.0 (4.4) | 37,000 |
| 3M—Maplewood, Minnesota | 1992 | CHW | 40.0 (4.4) | 32,000 |
| University of Texas—El Paso, Texas | 1999 | CHW | 40.0 (4.4) | 30,000 |
| University of Texas—Austin, Texas (#1 of 2) | 2010 | CHW | 40.0 (4.4) | 30,000 |
| DESP—St. Paul, Minnesota (#1) | 1994 | CHW | 39.0 (3.9) | 28,000 |
| Qatar Cool West Bay—Doha, Qatar | 2006 | CHW* | 40.0 (4.4) | 26,000 |
| University Tenaga—Bangi, Selangor, Malaysia (2) | 1996–1998 | CHW | 40.0 (4.4) | 26,000 |
| Texas A&M University—College Station, TX | 2016 | CHW | 42.0 (5.6) | 24,000 |
| Tabreed—Abu Dhabi, UAE | 2006 | CHW | 40.0 (4.4) | 18,000 |
| University of Nebraska—Lincoln East Campus, NE | 2011 | CHW* | 42.0 (5.6) | 16,326 |
| University of Virginia—Charlottesville, VA | 2001 | CHW* | 42.0 (5.6) | 16,200 |
| MPEA—Chicago, Illinois | 1994 | LTF | 30.0 (–1.1) | 123,000 |
| DFW International Airport—Texas | 2004 | LTF | 36.0 (2.2) | 90,000 |
| Princeton University—Princeton, New Jersey | 2005 | LTF | 32.0 (0.0) | 40,000 |

** Indicates an initial CHW TES, designed for possible future conversion to LTF TES (for achieving a typical 40% to 100% increase in TES capacity, without any increase in tank volume).*
*** Indicates a turbine inlet cooling (TIC) application.*

**Figure 6.8** The 40,000 ton-h (140,000 kWh) of stratified CHW TES at the University of North Carolina in Chapel Hill, North Carolina, was first in service in 2004. The 85 ft diameter × 116.83 ft high (35.6 m × 25.9 m), 5.0 million gallon (18,800 m$^3$) welded-steel tank, operates at CHW supply temperature of 41°F (5°C) and a return temperature of 54°F (12°C) and was designed for possible future conversion to LTF TES service at a larger ΔT and increased thermal capacity. In CHW TES service, it reduces peak demand by up to 10,000 tons (35,000 kW) of cooling and approximately 7 MW of electric power (or approximately 40% higher if converted to LTF TES).
*Courtesy of Affiliated Engineers, Inc.*

- In thermally stratified CHW TES, which is the most commonly used type of CHW TES, an insulated tank with internal flow diffusers maintains the cooler denser CHW supply water beneath the warmer, less dense CHW return water.
- Conventional CHW supply temperatures are employed, with a range of 39°F to 42°F (4°C to 6°C) being typical.
- Storage volumes are large, though less so at larger temperature differentials (typically at approximately 11 to 18 ft$^3$ per ton-h (0.09 to 0.15 m$^3$/kWh), for chilled water supply-to-chilled water return (CHWS-to-CHWR) temperature differentials of 12°F to 20°F (7°C to 11°C). Footprint is often minimized by the use of a tall tank (typically 40 to 80 ft [12 to 24 m]) tall, and sometimes as tall as 100 to 150 ft [30 to 46 m]).
- The dramatic economy-of-scale yields very low unit costs in large applications (e.g., district cooling).

## LTF TES Summary

- Energy is stored as sensible heat (a change in temperature of the storage medium).
- It is similar to stratified CHW TES, but uses fluid supply temperatures below 39°F (4°C).
- Lower supply temperatures are used, with a range of 30°Fto 36°F (–1 to +2°C) being typical.
- Storage volumes are larger than for ice TES, but smaller than for CHW TES (typically at approximately 7 to 10 ft$^3$/ton-h (0.05 to 0.08 m$^3$/kWh), for low temperature fluid supply-to-low temperature fluid return (LTFS-to-LTFR) tempera-

**Table 6.6** Characteristics of Cool TES Technology Types for DC Systems (Generalizations)

| Inherent Characteristics | Latent Heat TES | Sensible Heat TES | |
|---|---|---|---|
| | Ice TES | CHW TES | LTF TES |
| Unit volume (volume/ton-h) | Good | Poor | Fair |
| Footprint (plan area/ton-h) | Good | Fair | Good |
| Modularity | Excellent | Poor | Good |
| Economy-of-scale | Poor | Excellent | Good |
| Energy efficiency | Fair | Excellent | Good |
| Low temperature capability | Good | Poor | Excellent |
| Ease of retrofit | Fair | Excellent | Good |
| Rapid discharge capability | Fair | Good | Good |
| Simplicity and reliability | Fair | Excellent | Good |
| Site remotely from chillers | Poor | Excellent | Excellent |
| Dual-use as fire protection | Poor | Excellent | Poor |

ture differentials of 22°F to 30°F (12°C to 17°C). Again, footprint is minimized by the use of a tall tank, similar to CHW TES tanks.
- The economy-of-scale again yields low unit costs in large applications, such as district cooling.

Although it is possible to separately design and procure the various component elements of a sensible heat storage tank, the almost universally employed approach since the 1980s has been to specify and procure a turnkey (design-build) TES tank, inclusive of all its elements:
- Foundation
- Tank (in accordance with appropriate codes)
- Internal flow diffusers (to provide proper thermal stratification)
- Internal and external coatings
- External thermal insulation system (usually with vapor barrier and architectural finish)
- Standard fittings and appurtenances
- Thermal performance guarantees

In this way, the TES tank is specified and procured in a manner comparable to that used for the other key mechanical equipment components of the DCS (e.g., chillers, cooling towers, pumps, and heat exchangers).

## Comparing TES Technologies

Each TES technology has inherent advantages and limitations. Generalizations can be made and used as approximate rules-of-thumb, as presented in Table 6.6. Of course, any generalizations should be viewed with some caution, as a fuller understanding of the technologies is important to optimally select and employ TES for any specific district cooling application.

In particular, note that each TES technology is rated as excellent and good in some respects, yet only fair and even poor in other respects. Accordingly, it is crucial to focus on those issues that are most important to a particular district cooling application when selecting the TES technology. But also, always remember that Table 6.6 provides only a generalized starting point for evaluation and decision-making.

## DRIVERS FOR AND BENEFITS OF USING TES IN DISTRICT COOLING SYSTEMS

There are often compelling reasons to incorporate TES into DCS design and operation.

### Primary Benefits of Using TES in District Cooling Systems

The traditionally recognized benefit from TES is:

- *Operating energy cost savings*. By shifting cooling loads from peak to off-peak periods, TES will reduce peak electric power demand and electric utility demand charges, and will shift electric energy consumption from high cost on-peak periods to lower cost off-peak periods.

But additional, and sometimes even more important factors, are related to capital cost:

- *Reduction in the required installed capacity of chiller plants*. With TES, it is possible to meet a peak 24 h design-day load profile with an operating chiller plant that has an output that is merely the size of day's 24 h average load. This compares to a conventional (non-TES) situation where the same load profile requires an operating chiller plant with an output at least equal to the day's instantaneous peak load. Accordingly, if TES is installed at a time that would otherwise require an investment in chiller plant capacity, (e.g., at a time of either new construction, retrofit expansion, or chiller plant retirements or rehabilitation), then the necessary conventional chiller plant investment can be avoided or at least reduced. The net result (considering both the cost of the TES system and the avoided cost of the chiller plant capacity) is generally only a small incremental capital cost for the TES option (versus no TES) with a rapid and attractive economic payback. In fact, in many cases (notably, large district cooling applications of CHW or LTF TES) there is an immediate net-capital cost saving versus the non-TES option; there are many examples of such installations, each with multimillion dollar net-capital cost savings, which have been documented (Andrepont 2005c and 2005d).
- *Potential utility incentive payments*. TES capital cost can be further reduced in those situations where the local electric utility provides a one-time cash incentive payment based on the reduction in peak electric demand achieved with the TES system. These payments, where available, can be several hundreds of US dollars per kW of demand reduction.

### Potential Secondary Benefits of Using TES in District Cooling Systems

There are numerous additional potential benefits of TES use, one or more of which may apply in significant ways for particular district cooling project situations:

- *Balancing thermal and electrical load profiles*. TES helps to flatten and balance thermal and electrical loads, which can often aid in the economic deployment of combined heat and power (CHP) as part of a district energy system, improving energy efficiency, fuel use, emissions, economics, and resiliency.
- *Improved on-site energy efficiency*. Although TES does have some inherent inefficiencies (e.g., heat gain into storage and pump energy to charge and discharge TES, as well as increased chiller plant kW/ton when LTF TES or especially ice TES is used); TES also has inherent efficiencies (e.g., lower nighttime condensing temperatures for reduced chiller plant kW/ton and avoidance of inefficient severely low part-load operation of chiller plants, as well as potentially reduced pumping and fan energy if LTF or ice TES is used to reduce supply temperatures). Although some TES systems somewhat increase annual on-site energy

consumption, there are many documented examples of systems with somewhat reduced annual on-site energy consumption, (e.g., Andrepont 1994, 2000a; Fiorino 1992; Potter et al. 1995; Tabors Caramanis & Associates 1995). CHW TES can also allow increased usage of seasonal free cooling from cooling towers.

- *Reduced fuel use and emissions at source power plants*. Independent studies of six large electric utilities in four US states, namely California (Flory 1995; Tabors Caramanis & Associates 1995), Florida (Nix 2008), Texas (Reindl et al. 1994), and Wisconsin (Gansler 1999), have documented the dramatic positive impacts of shifting electric energy consumption from on-peak daytime periods to off-peak nighttime periods. Reductions of 20% to 30% and greater were demonstrated in terms of source utility power plant fuel use and emissions of SOx (sulfur oxide), NOx (nitrogen oxide), particulates, and $CO_2$ (carbon dioxide), due to the different performance of the power plant equipment, which is on the margin during the different times of day. The potential value of energy storage in general, and of TES in particular, is becoming even much greater as increasing amounts of intermittent, variable, and renewable energy (notably wind power, which is often out of phase with peak power demand) is added to the electric power grid.
- *Improved flexibility of operations and maintenance*. TES decouples the production of cooling from the demand for cooling. This provides useful operational flexibility. It also allows chiller plant maintenance to be more easily accomplished during daytime periods.
- *Emergency standby cooling*. TES can be used not only for daily load management of peak cooling loads, but to provide a standby reserve for emergency cooling of mission-critical loads.
- *Emergency condenser water makeup (applicable to CHW TES)*. In some arid climates where freshwater resources are very limited, CHW TES is also used as an emergency reserve for cooling tower or evaporative-condenser makeup water.
- *Fire protection water storage (applicable to CHW TES)*. In many instances, CHW TES serves a dual function as a fire protection reservoir, which reduces risks and can sometimes reduce insurance premiums. The TES tank can be designed and installed to meet NFPA 22 code requirements and even hard-piped through fire water pumps into a fire sprinkler network, or the tank can merely be fitted with a hydrant connection.
- *Skyline advertising (applicable primarily to CHW and LTF TES)*. Tall TES tanks have been used to prominently display corporate or institutional names or logos, and have even been used for commercial signage or advertising at shopping malls.
- *Remote satellite location of TES (applicable to CHW and some LTF TES)*. As described further in the next section, remotely located TES can serve the same function as a satellite chiller plant, feeding the district cooling network from an independent point to overcome peak load bottlenecks in the distribution network.
- *Low temperature distribution (applicable to ice and LTF TES)*. Low temperature supply can be used to increase the $\Delta T$ in the district cooling network, thus reducing the size and cost of piping, pumps, and air-handlers, as well as reducing pump energy and energy costs. And if the low temperature water or fluid supply is used to produce low temperature air, then fans and air ducts can be smaller and less costly, with reduced fan energy and fan energy costs.

- *Improved and stabilized water treatment (applicable to specific LTF TES)*. In the case of at least one commonly used LTF, the chemical additives in the aqueous fluid provide exceptional long-term (essentially permanent) inhibition against corrosion and microbiological activity, with no need for ongoing additions of corrosion inhibitors or biocides.

## SYSTEM INTEGRATION

### Location of TES Equipment

TES equipment in DCSs is often located at or near the central chiller plant or at or near one or more satellite chiller plants. However, in some cases with certain TES technologies, it is sometimes the case (and sometimes advantageous) that the TES equipment is located remotely from the chiller plants.

In cases of ice TES, the ice-generation equipment (which is generally also the ice-storage equipment) must be located at or relatively near to the low temperature ice-making chillers, as there are practical limits to the how far refrigerant or water-glycol can be transported between the chillers and the location where the ice is generated. Some ice TES technologies generate a pumpable ice slurry (water and ice mixture), and in those cases, it can be practical to generate ice in one location (near the low temperature chillers) and pump the ice slurry to one or more remote (distributed) ice-storage locations; however, this has not been done extensively.

In cases of CHW TES, the CHW storage tank can be located either at a chiller plant or remotely at any point along or within a reasonable distance from the CHW district cooling piping network. As a CHW TES tank is recharged using conventional temperature CHWS, that type of TES can be recharged from any CHW plant or directly from the CHWS distribution network. Therefore, the CHW TES location is totally flexible. Of course, if the CHW TES is located remotely from chiller plants, there will need to be dedicated TES pumps located at or near the TES tank for TES recharging and discharging, but no need for local chillers, cooling towers, or other heat rejection equipment, or major electrical service at that TES tank site. If the CHW TES tank is located at a chiller plant, the tank may still employ dedicated TES pumps for TES recharging and discharging, or the tank may be recharged and discharged using the local chiller plant's primary and secondary CHW pumps (as described in the section Hydraulic Integration of TES in this chapter).

In cases of LTF TES, the LTF tank must generally be near the LTF chillers for instances where LTF-to-CHW heat exchangers are used to isolate the LTF only in the low temperature chillers and LTF TES tank, with CHW flowing in the district cooling distribution network. However, there are also systems where the LTF is used as the district cooling medium, directly through the district cooling distribution network (either with LTF-to-CHW heat exchangers at the individual buildings served by the DCS, or with LTF directly through the air-handling units at the cooling loads); in such cases, the LTF TES tank can be located remotely from chiller plants anywhere along or near the district cooling net-work, similar to the case of a CHW TES tank located remotely in a conventional CHW district cooling network.

The use of remotely located TES (e.g., using CHW TES, LTF TES, or even ice slurry TES) can sometimes provide specific benefits versus the use of a tank at a chiller plant. These benefits can include:

- overcoming limited space availability at the chiller plant location(s),
- overcoming aesthetics issues associated with having a tank at the chiller plant location(s),

- allowing the tank's head of water to set the static pressure for the entire district cooling network by locating the tank at an elevated site, and perhaps most commonly and most importantly,
- overcoming bottlenecks in the district cooling network, by locating the TES at a strategic location (similar to the locating of a conventional satellite CHW plant), and in this way allowing the TES to provide peaking capacity not only in terms of CHW generation, but also in terms of CHW distribution.

There have been numerous instances of remotely located CHW TES (particularly on growing university campuses) where the remote location of TES achieved those very benefits and provided a significant reduction in net capital cost versus the cost of the alternatives of increasing network piping or building a satellite chiller plant.

Once a TES tank location has been sited within the plot plan of a DCS (whether at a chiller plant or remotely), there is still the choice of locating the tank aboveground or belowground. In making that design choice, the generalizations presented in Table 6.7 should be considered along with specific requirements or preferences for a particular project and site.

Generally, economic reasons, such as much lower capital cost, as well as lower TES pumping energy cost, have led to the very large majority of TES district cooling installations being installed aboveground.

Furthermore, the preponderance of large aboveground TES tanks are constructed of welded steel, though some have used concrete construction. And, although below-grade tanks always have a substantially higher capital cost, nearly all of those large below-grade (direct-buried) TES tanks are concrete. However, steel tanks have also been constructed at the base of an excavated pit or depression.

## Hydraulic Integration of TES

Capital cost considerations require that virtually all large TES tanks be designed and operated as atmospheric pressure tanks (rather than as pressure vessels).

There are two common methods (and many variations) of hydraulically interconnecting atmospheric pressure, stratified CHW TES tanks to a CHW system. Site-specific considerations dictate the optimum approach for each application. The two most common methods are:

1. Integrating TES with a nearby chiller plant, using the plant's primary-secondary CHW pumping, and
2. Siting TES remotely from (or near to) the chiller plant(s), while using dedicated TES pumps.

**Table 6.7** Issues of Locating TES Tanks Aboveground versus Belowground (Generalizations)

| Issues | Belowground TES | Aboveground TES |
|---|---|---|
| Site space utilization | Can be preferable | Can be less preferable |
| Site aesthetics | Out of sight and mind | Visible |
| Sensitivity to site soil conditions | Very sensitive | Less sensitive |
| TES tank inspection & maintenance | More difficult | Less difficult |
| TES tank capital cost | Much higher | Much less |
| TES-to system hydraulic differential | Higher | Lower |
| TES tank pumping energy | Higher | Lower |
| TES tank pumping energy cost | Higher | Lower |
| Dual-use as a fire protection reservoir | Not as practical | Very practical (for CHW TES) |

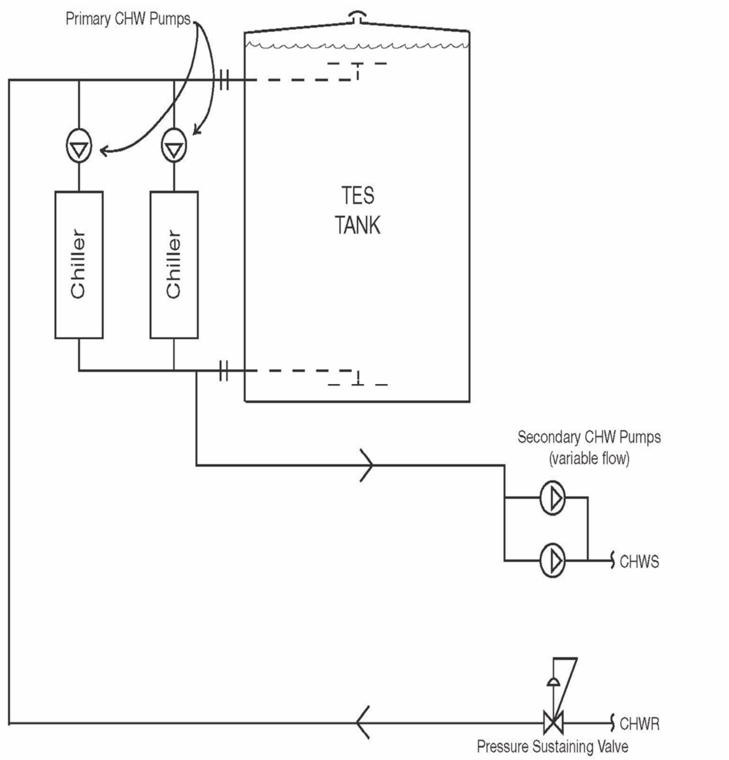



**Figure 6.9** One method of interfacing a CHW tank with the DC network and chiller plant.

In the first method (illustrated in Figure 6.9), the TES tank is piped directly into an open bridge between the chiller plant's CHWR header and the primary CHWS header. The upper region of warm (lower density) return water in the TES tank communicates with the CHWR header. The lower region of cool (higher density) supply water in the TES tank communicates with the primary CHWS header at a point downstream of the chillers but upstream of the secondary CHW pumps. Whenever the plant's primary CHW flow exceeds the secondary CHW flow (which is typically controlled to match the load in the network), as will usually be the case during off-peak periods, the TES tank will automatically recharge at a rate equal to the difference between primary and secondary flows. And whenever the plant's primary CHW flow is less than the secondary CHW flow (again, typically controlled to match the load in the network), as will usually be the case during on-peak periods, the TES tank will automatically discharge to supplement the chillers at a rate equal to the difference between secondary and primary flows. This method is the simplest and least expensive approach, but it requires the TES tank be near to the chiller plant. This method limits TES recharging to only those nearby chillers.

In the second method (illustrated in Figure 6.10), the TES tank again communicates with the CHWS and CHWR headers, with the CHWS communicating with the lower region of the TES tank and the CHWR communicating with the upper region of the TES tank. However, water must always be pumped from the relatively low (atmospheric) pressure of the TES tank into the higher pressure CHW headers; this is the case both during TES discharge when cold water is pumped from the lower region of the tank into the CHWS header and during TES recharge when warm water is pumped from the upper region of the tank into the CHWR header. Generally, the same set of TES tank pumps is used for both recharging and discharging, with interconnecting cross-over

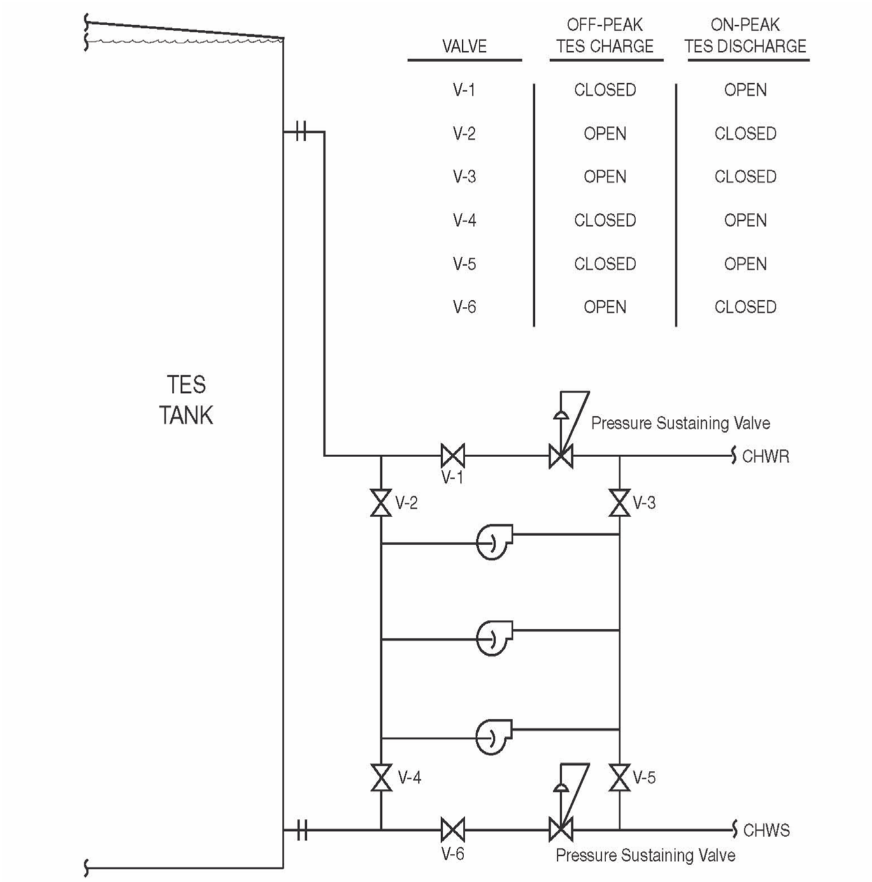



**Figure 6.10** **Alternate method of interfacing the CHW storage tank with the district cooling network and chiller plant.**

piping and six, two-position (fully open or fully closed) valves used to select the available operating mode to suit the time of day. Although a somewhat more complex and costly option, this method is very commonly used as it allows for the TES tank to be sited remotely from the chiller plant. That is sometimes an important benefit to the piping network, because the tank can be strategically located to act as if it was a satellite chiller plant during TES discharge. It may also be important for reasons of land allocation or even aesthetics. Note that this method allows the TES to be charged directly from the campus CHWS header, which is to say that the TES can be recharged by any chillers in one or more chiller plants, whether located nearby or remotely.

*Pressure sustaining valves*. With both of the previously described methods, it is common to use pressure-sustaining valves (PSVs), also known as back-pressure control valves, in the lines flowing back to the TES tank from the higher pressure system. These valves maintain the necessary and appropriate minimum pressure within the network, based on a local pressure set point on the immediate upstream side (i.e., the system or high-pressure side) of the PSV. This prevents the pressurized network water from draining into the atmospheric pressure tank, which could not only overflow the tank but also create a vacuum and draw air into the network at its highest elevation (lowest pressure) points. The PSVs are typically self-contained, pilot-operated devices, with manually adjustable set points.

Other means of addressing the hydraulic pressure differential between the atmospheric pressure tank and the pressurized district cooling CHW system are occasionally used and include the following:

1. *Use of heat exchangers*. Heat exchangers can be used either between the TES tank (or the TES tank and local chiller plant) and the balance of the DCS, or heat exchangers at the buildings connected to the DCS. This simplifies the hydraulic issues but adds substantial capital cost associated with the heat exchangers and an additional pump set, and also reduces the thermal quality of the delivered cooling (in that the supply temperature on the user load or demand side of the heat exchangers will necessarily be higher than that on the chiller plant or supply side of the heat exchangers).
2. *Use of dominantly tall (or elevated) TES tanks*. In some (relatively rare) instances it is possible to have a TES tank that is so tall (or elevated on a hill or a building) that the static head of fluid in the atmospheric pressure TES tank can set the static pressure for the entire DCS, thus avoiding the need for the PSVs.
3. *Use of a somewhat dominant tank in conjunction with PSVs or HXs and tertiary pumps located at the overly tall building loads connected to the DCS*. In this manner, PSVs (or heat exchangers and tertiary pumps) need only be used for that portion of the DCS (e.g., the tallest buildings), that is higher in elevation than the static head of fluid in the atmospheric pressure TES tank.
4. *Use of recovery turbines*. When the differential pressure between the TES tank and the DCS is very large, yet the use of heat exchangers is still not desired (e.g. due to capital costs and/or thermal quality of the supply to the district cooling customers), hydraulic recovery turbines (typically pumps running in reverse) are placed in the return lines to the TES tank, with the output shaft of the recovery turbine driving the input shaft of the outbound pump. In this manner, approximately 60% to 75% of the pump's shaft power comes from the recovery turbine, leaving only the remainder to be met by an undersized motor, thus reducing the electrical input for the pumps. However, capital cost, complexity, and reliability should be considered relative to the savings in pump energy costs.
5. Other alternatives, perhaps uniquely suited to a particular project situation.

Each approach has advantages and disadvantages or limitations, all of which should be explored before selecting the approach to be used for a particular project, after giving consideration to capital costs, operating energy efficiency, operating costs, simplicity, reliability, etc. However, one of the first two options described at the start of this section, using pumps and PSVs (but no heat exchangers), is almost always the choice for CHW TES systems (except where tall or elevated tanks can eliminate the need for the PSVs). The use of heat exchangers, effectively segregating the system into two circuits, is common with LTF TES systems and with many ice TES systems; of course, with most LTF TES and with many ice TES systems (notably the many systems using water-glycol brine to recharge the ice storage), there is already an inherent need for heat exchangers to segregate the two fluids (either LTF and CHW or water-glycol brine and CHW) into two distinct circuits.

Extensive discussion and design details related to the integration of atmospheric TES systems with pressurized CHW systems can be found in the *Cool Storage Open Hydronic Systems Design Guide* (Gatley and Mackie 1995).

## SIZING AND OPERATION OF TES

Selecting the optimum capacity (including total storage capacity as well as maximum recharge and discharge rates) of a TES system is dependent on several application-specific parameters, with key factors usually including:

- Current and future peak design cooling loads
- Current and future 24 h peak design day (or 7-day peak design week) cooling load profiles
- Type, operating condition, temperature capability, and capacity of existing chillers (if any)
- Required or desired level of cooling system redundancy (e.g., none, $N+1$, or $N+2$)
- Current and future electricity tariffs or alternative electricity purchase scenarios
- Secondary usage of TES (if any), e.g., as an emergency cooling reserve or as fire protection.

In addition to the examples below, ASHRAE (2016) provides further detailed examples on sizing of CHW TES for both full-shift and partial-shift systems.

### Full Versus Partial-Shift TES Systems

A TES system can be designed and operated as a full-shift system, in which 100% of the on-peak (high demand) period cooling load is met by the discharge of TES, with no chillers operating at that time. Alternatively, a TES system can be designed and operated as a partial-shift system, in which only a portion of the on-peak (high demand) period cooling load is met by the discharge of TES, while at least some chillers are operating simultaneously to supplement TES and meet the cooling loads at that time.

Full-shift TES systems have the largest TES capacity requirement and also the largest installed chiller plant capacity requirement (as all the chillers are restricted to operate during fewer hours, (i.e. only during non-peak) periods. Full-shift TES systems will maximize the reduction in on-peak power demand and thus generally maximize the operating energy cost saving. However, full-shift TES systems will also generally have the highest TES capital cost due to the larger capacity of both the TES and the chiller plant.

Partial-shift TES systems have a more modest TES capacity requirement and also a more modest installed chiller plant capacity requirement, as at least some of the chillers can operate during both non-peak and on-peak periods. Partial-shift TES systems will provide some reduction in on-peak power demand and the associated operating energy cost saving, but less than for a full-shift system. However, partial-shift TES systems will have a lower capital cost due to the more modest capacity of both the TES and the chiller plant.

One particular case of the partial-shift TES systems is the load-leveling TES system, in which a variable 24 h cooling load profile is met using the operation of a constant (and minimum) capacity chiller plant, with higher (or lower) levels of instantaneous cooling load being met by concurrent discharging (or recharging) of TES. Load-leveling TES systems provide only a relatively modest reduction in on-peak power demand and an associated modest operating energy cost saving. However, load-leveling TES systems will have a minimum capital cost due to having the minimum capacity of both the TES and the chiller plant. As indicated previously, the total capital cost of a partial-shift TES system can be less (even much less) than for an equivalent conventional (non-TES) chiller plant installation (Andrepont 2005c and 2005d).

Chiller plant redundancy can often impact the sizing of TES. For example, a chiller plant may be sized with a firm capacity of $N$ chillers equal to the load-level system capacity (i.e., equal to the average cooling load of the peak 24 h design day), but also including one additional spare chiller (to provide $N+1$ redundancy). Thus, the system uses the minimum possible installed chillers, while still providing the required level of redundancy (in this case $N + 1$). But rather than install the minimum size load-leveling TES capacity, the TES can be sized to accept all of the available off-peak capacity of the full $N + 1$ chiller plant, less the capacity used to meet off-peak cooling loads. This larger TES capacity will provide a larger on-peak load reduction and a larger associated operating energy cost saving. The larger TES has a higher capital cost, but there is no increase in the size and cost of the chiller plant, compared to those of the minimum load-leveling plant. Often the incremental cost for the larger TES tank is fully justified by the larger operating energy cost saving.

Most TES applications for district cooling are designed and operated as partial-shift systems, at least on peak cooling design days. However, on low-load days (e.g., due to reduced user activity and/or cooler ambient conditions), some partial-shift systems can be operated as full-shift systems.

Figure 6.11 illustrates examples of the 24 h design-day chiller load profile for a non-TES base case, as well as for a full-shift TES option and for a load-leveling TES option for a hypothetical DCS with a peak cooling load of 10,000 tons and a design-day load factor (the ratio of average load to peak load) of about 86%. Some items to note include:

1. The total area under each of the three curves (the ton-h of cooling produced by the chiller plant in the 24 h period) is the same in each case.
2. The full-shift TES requires a larger chiller plant (~13,000 tons, plus any spare) versus the non-TES base case (10,000 tons, plus spare), while the load-leveling chiller plant capacity (~8600 tons, plus spare) is smaller than the base case.
3. The full-shift TES reduces on-peak demand by 10,000 tons, for 8 on-peak hours, while the partial-shift TES reduces on-peak demand by up to ~1400 tons over a 13-h period.
4. The area under the no TES curve and above the full-shift curve (or under the no TES curve and above the load-leveling curve) represents the required ton-h capacity of the full-shift TES (or of the load-leveling TES), with the full-shift TES capacity (~78,000 ton-h) being approximately 6.5 times larger than the partial-shift TES (~12,000 ton-h) in this case.
5. The actual results are of course highly impacted by both the cooling load profile and the duration of the on-peak period. Load profiles with more variability (i.e., lower load factors), or shorter duration on-peak periods, will produce even greater value from the use of TES.

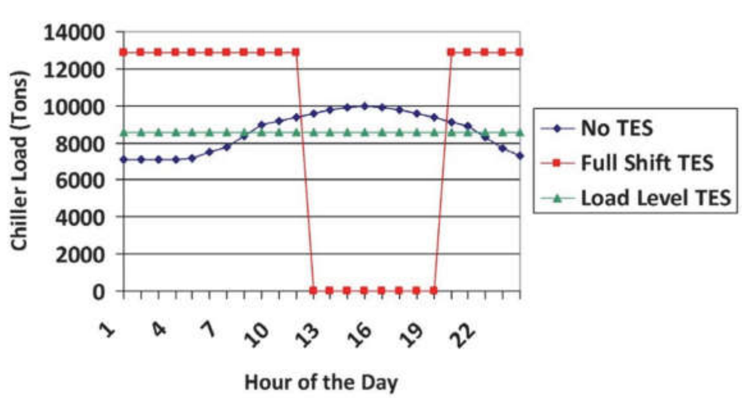



**Figure 6.11** Comparison of TES options for a 24 h design-day load profile.

## Daily Versus Weekly Cycle TES Configurations

A TES system can be designed and operated as a daily cycle system in which a 24 h design-day load profile is met by fully recharging TES during non-peak (low demand) periods and then fully discharging TES during on-peak (high demand) periods. Alternatively, a TES system can be designed as a weekly cycle system, in which a 7-day design week load profile is met by fully recharging a larger TES during a weekend period, then partially discharging TES each weekday during on-peak periods, and partially recharging TES each weeknight during non-peak periods.

Daily cycle TES system designs are by far the most prevalent. Weekly cycle TES system designs are desirable only if weekend hours are entirely (or nearly entirely) off-peak periods, in terms of both low electric rates and low cooling loads. Compared to a daily cycle design, a weekly cycle design will require a smaller and less costly chiller plant, but a larger and more costly storage tank, often two to four times the capacity of that needed for a daily cycle design. Accordingly, the weekly cycle design is almost never found to be economically attractive for any sensible heat (CHW or LTF) TES system or for ice-on-coil or encapsulated ice TES systems (as in all those cases, the increase in storage cost is generally greater than the saving in chiller plant cost). However, for instances of ice harvester or ice slurry TES, which exhibit a high unit capital cost per ton of ice generating capacity, the cost saving associated with the smaller ice generators can more than offset the cost of the larger storage tank. Nevertheless, most district cooling applications of TES utilize daily cycle TES designs and employ sensible heat (CHW or LTF) TES or ice-on-coil (or encapsulated ice) TES.

## TES Control

The control of TES, whether done manually or (more often) by automated means, is essential to maximize the value of TES. In some instances, the use of TES is crucial in meeting the peak cooling loads of the DCS, (e.g., where those peak loads exceed the firm capacity of the installed chiller equipment). In many other instances, TES use is primarily (or additionally) utilized to reduce on-peak electric power and/or to shift electricity usage from high-cost on-peak to low-cost off-peak periods. TES may also be used to avoid ever operating any chillers (and their auxiliary equipment) at severely low (inefficient) part-load conditions.

Specific sequences of operation for TES recharge and discharge should be developed based on the desired and necessary goals of the TES system, (e.g., peak electric-demand reduction and electric-load shifting, part-load avoidance, and/or availability for meeting peak loads).

Some systems (e.g., most CHW TES systems) have little energy penalty when recharging TES versus operating chillers to directly meet the district cooling loads, and as heat gain into storage is generally limited to less than about 1% or less of the TES cooling capacity per day, there is little to be gained by not fully recharging TES during each off-peak (night-time) period. However, some other systems (e.g., many ice TES systems) exhibit substantially higher rates of energy consumption (kW/ton) as the ice TES equipment becomes progressively more recharged; in those cases, it can be preferred to recharge TES each night only to the extent that TES will be discharged the next day.

In all cases, for optimal or near-optimal use of TES, it is important to have a means of monitoring the inventory of cooling available within TES at any time during TES charging or discharging, as well as a means of controlling the rate of TES charging and discharging. This is important in order to ensure that TES can be adequately recharged during a particular off-peak period and TES is fully or nearly fully discharged by the end of a particular on-peak period, but not fully discharged prematurely before the end of that period, (which could result either in insufficient capacity being available to meet peak loads or in excessive chiller loading being needed with resultant excessive on-peak electric power demand).

The inventory of cooling TES is generally monitored:

- In the case of ice TES, by the level of water in the ice storage tank(s), which rises (and falls) as ice is frozen (or melted) due to the higher specific volume (lower density) of ice relative to that of liquid water
- In the case of stratified CHW or LTF TES, by the use of temperature instrumentation located at intervals throughout the depth of the storage tank (either via a bundle of instruments hung through a roof nozzle or via individual instruments in shell-mounted thermowells), used to identify the depth of the zone of cold supply water available in the bottom of the tank

# ECONOMICS OF TES IN DISTRICT COOLING

## Capital Costs

Capital costs of TES installations vary quite widely, based on many factors which include:

- Whether or not TES requires additional new chillers (or can use existing chillers)
- TES system design (e.g., daily versus weekly design cycle, and full versus partial-load shift)
- Type of TES technology
- Capacity of TES and the economy-of-scale of the chosen TES technology
- Supply-to-return temperature difference (for sensible heat TES)
- Maximum charge and discharge rates for TES
- Location of TES (above versus belowground, outside versus in or on a building, and local to versus remote from chillers)
- Site conditions
- Height restrictions and/or requirements
- Geotechnical (soil) conditions
- Hydraulic and pumping interface between TES and the balance of the system
- Unusual TES tank roof loading (if any)
- Unique aesthetic or architectural treatment (if any)
- Quality of equipment and installation
- Material costs (prices and escalation rates vary with time)
- Local labor rates (and whether unionized labor is required)
- Local utility demand-side management (DSM) incentive payments (if any)
- Current economic and market conditions (supply and demand)

Given all the many variables, it is not possible to define specific unit capital costs that could be universally applied for TES. However, the very approximate ranges of typical unit costs presented below (Andrepont 2018a) can be used (with caution) for a preliminary assessment (divide $ per ton values by 3.517 to get cost per kilowatt-hour):

- Chiller plant capacity (if required to be added), including installed chillers, cooling towers or equivalent heat rejection, pumps, valves, instrumentation & controls, electrical service, and building—$2000 to $4000 per ton.
- Ice TES, installed, without chillers, pumps, controls, etc.—$120 to $175 per ton-h (or $960 to $1400 per ton, for an eight-hour discharge of TES)
- Ice TES, installed, with chillers, pumps, controls, etc.—$250 to $560 per ton-h (or $2000 to $4480 per ton, for an eight-hour discharge of TES)

- CHW TES, installed, belowground—\$115 to \$300 per ton-h (or \$920 to \$2400 per ton, for an eight-hour discharge of TES)
- CHW TES, installed, aboveground, large capacity (over 20,000 ton-h)—\$35 to \$100 per ton-h (or \$280 to \$800 per ton, for an eight-hour discharge of TES)
- CHW TES, installed, aboveground, medium capacity (10,000 to 20,000 ton-h)—\$70 to \$200 per ton-h (or \$560 to \$1600 per ton, for an eight-hour discharge of TES)
- CHW TES, installed, aboveground, small capacity (5000 to 10,000 ton-h)—\$95 to \$235 per ton-h (or \$760 to \$1880 per ton, for an eight-hour discharge of TES)
- LTF TES, installed, aboveground—very similar to CHW TES (as the smaller and less expensive tank is roughly offset by the added cost of the chemical additives in the fluid)
- Hydraulic integration of TES to the balance of system – \$150 to \$500 per ton

## An Actual Case Study of TES for District Cooling, with Economics (Andrepont and Kohlenberg 2005)

A major university operates a campus-wide DCS serving the cooling loads on its combined academic and medical, urban campus. In 2002, peak campus cooling loads of 19,500 tons were being met by an existing central CHW plant of 26,000 tons installed capacity (and 22,000 tons firm capacity). However, on-going campus load growth increased the projected peak load to 29,000 tons in 2008, overtaxing the available chiller plant capacity as well as the district cooling piping distribution network capacity. Potential solutions were defined and evaluated, including:

- Adding in-building (non-district cooling) chiller plants to serve the new cooling loads
- Expanding the capacity of the existing central chiller plant and the district cooling network
- Adding a new satellite chiller plant
- Adding TES (by far the most economically attractive alternative)

Accordingly, in 2005, a 60,000 ton-h stratified CHW TES system was installed as a retrofit capacity expansion. Added to address the campus cooling load growth, the 150 ft diameter × 60 ft tall (45.7 m × 18.3 m), nominal 7.9 million gallon (30,000 $m^3$), aboveground welded-steel TES tank was located as remote satellite capacity, at the opposite corner of the campus from the existing central CHW plant. This effectively provides peak shaving not only for the CHW generating capacity, but also for the distribution piping system capacity as peak loads are met by CHW flowing simultaneously from both the chiller plant and the TES tank. The TES tank can meet peak loads of 7215 tons, achieving a peak electric demand reduction of approximately 5.4 MW.

Relative to the otherwise required conventional (non-TES) CHW system capacity addition (a new 7000 ton satellite chiller plant), the use of TES achieved a 30% immediate capital cost savings (approximately \$4.8 million) and was projected to achieve a 12% reduction (approximately \$0.7 million/yr, on average) in campus cooling system annual operating costs. The combined net present value from 20 years of operating savings plus the immediate capital saving was calculated to be approximately \$10.4 million, from the use of TES versus a conventional chiller plant installation.

Furthermore, the TES system (normally operating at a CHW supply temperature of 41.0°F (5.0°C) and a return temperature of 53.6°F (12.0°C) was predesigned to allow for a possible future conversion of the system to LTF service (at a supply temperature of

30.0°F [–1.1°C] and a return of 51.8°F [11.0°C]). This conversion would achieve a dramatic 70% increase in TES capacity (from 60,000 ton-h as CHW TES, up to 102,000 ton-h as LTF TES), with a similar percentage increase in the TES discharge rate (from 7215 tons to 12,265 tons) and in the district cooling network piping distribution capacity. The reduction in peak electric demand would also increase from approximately 5.4 MW to approximately 9.2 MW, with associated increases in operating energy cost savings.

Similarly attractive economic results have been repeatedly seen with the use of TES in many other district cooling applications.

It is recognized that when employing TES systems with smaller chiller plants than would be required without TES, the chillers will generally have more operating hours per year than would non-TES chillers, thus potentially increasing chiller plant equipment maintenance and reducing life expectancy. However, those same TES chillers will generally have less on-off cycling than would non-TES chillers, thus potentially reducing chiller plant equipment maintenance and increasing life expectancy in an off-setting manner.

## COMPARING ENERGY STORAGE TECHNOLOGIES

Reindl et al. (2017) studied available methods of TES to determine how they could help support electric power systems in utilizing renewable energy resources. Figure 6.12 below shows the capital cost for both energy and power capacities for the most promising energy storage technologies. From Figure 6.12 it can be seen that CHW TES (labeled as CTES in Figure 6.12) is very competitive if not the lowest cost alternative for nearly all circumstances. Batteries (Li-ion) are the topic of much discussion at present (Kosowatz, 2017; Winters 2018), but it can be seen their costs are much higher, a detailed discussion of battery advantages and limitation and a comparison to CHW TES is presented below.

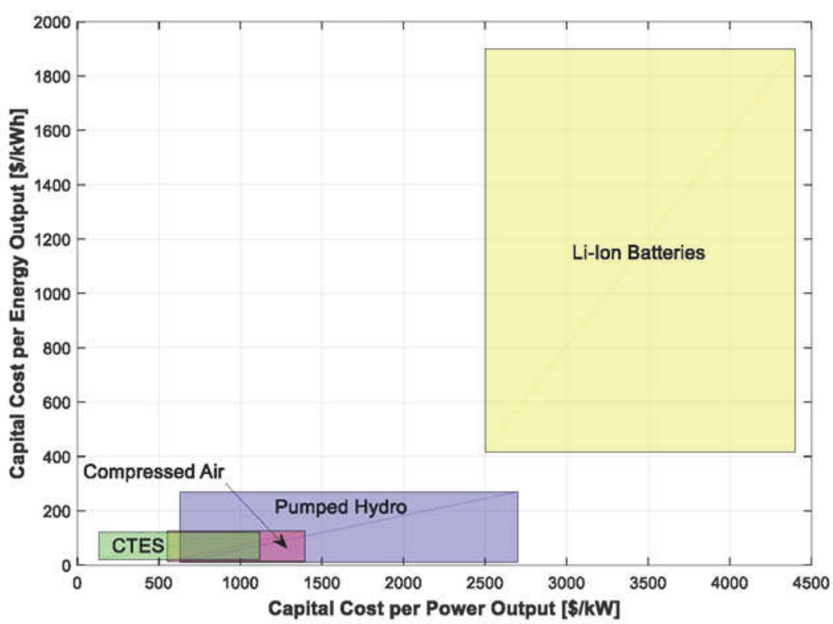



**Figure 6.12** Capital cost comparison of energy storage technologies (Reindl et al. 2017).

## Battery Storage—Advantages and Limitations (Andrepont 2018b)

Batteries provide several characteristics which make them attractive as energy storage, both for the power grid and for microgrids (see Table 6.8a). Accordingly, they are often the chosen storage technology, but unfortunately, this occurs too often without consideration of other options.

After taking a closer look at other characteristics, batteries pose various challenges and serious drawbacks, even for Li-ion batteries, which are the most prevalent type being applied commercially at the present time (as shown in Table 6.8b).

**Table 6.8a** Advantages of Batteries

| | |
|---|---|
| **Energy Density** | Batteries are high in energy density, and thus compact. |
| **Siting Flexibility** | Almost any location is a candidate site for batteries. |
| **Installation Schedule** | Permitting, procurement, and installation schedules are relatively rapid. |
| **Responsiveness to Load** | Batteries can be capable of being discharged and recharged quite rapidly. |
| **Multifunction Capability** | Batteries can serve one or more functions, including: frequency control, load following, peak shaving, and load shifting. |

**Table 6.8b** Limitations of Batteries

| | |
|---|---|
| **Energy Efficiency** | Round-trip energy efficiencies are typically listed in the range of 80% (to 90%), depending on the type of battery and the depth of charge/discharge per cycle. Tesla reported that its highly publicized Li-Ion installation in the South Australia grid had an output of 2.42 GWh from an input of 3.06 GWh for the entire month of December 2017, an efficiency of only 79%. |
| **Material Sourcing** | They require exotic, costly materials, e.g. lithium-graphite anodes and nickel-manganese-cobalt oxide (NMC) cathodes in Tesla's Li-Ion daily-cycle application batteries. And these materials are primarily sourced from potentially unreliable locales, e.g., 70% of graphite comes from China, while 70% of cobalt comes from the Congo, China, and Russia. |
| **Safety and Environmental Issues** | Li-Ion batteries have a cautionary history of explosions and fires in operation, adding regulations, restrictions, and costs to potential application sites. In addition, the mining and refining steps, associated with the sourcing of some of the exotic materials from their diverse locales, pose serious concerns of environmental damage. |
| **Material Disposal** | The compound mix of hazardous, heavy metal components makes frequent end-of-life recycling/disposal a very complex and costly endeavor (or else a long-term hazardous waste issue, if installations are simply abandoned in-place). |
| **Life Expectancy** | Various battery types are listed as having a lifetime of 7 to 15 years (or only 7 to 10 years for some, including Li-Ion); and the number of charge/discharge cycles has an impact on life, such that a battery that cycles on a daily basis (as one typically would for electric load management) will have a shortened life; e.g. Tesla has noted a life of 1,000 to 1,500 cycles for its Li-Ion batteries, which would equate to only 4 to 6 years if cycled for just 250 days/year. Also, batteries have a reduction in effective capacity during their lifetime, exacerbated if the cycling is at or near full charge/full discharge levels. |
| **Initial Cost** | Even though battery costs have been falling, they are still high, and represent only part of the cost of a completed storage installation. Tesla's NMC Li-Ion daily-cycle batteries have been specified to be $3,000 for a rated 7 kWh, or $429/kWh merely for the batteries themselves. A recent project proposal in Texas was priced at $2.3 million for 3.0 MWh of storage, or $767/kWh for a full Li-Ion battery installation. Typical project costs are in the range of $500 to $800/kWh. And these already high initial costs are compounded by the very short life and the end-of-life disposal costs. |

**Table 6.9** Comparison of TES and Li-Ion Battery Electric Energy Storage

| | CHW TES | Li-Ion Batteries |
|---|---|---|
| Installed Unit Capital Cost | < $300/kWh | Approximately double |
| Avoided Future Chiller Plant Cost | $ millions | Zero |
| Extends Use of Winter "Free Cooling" | Yes | No |
| Summer Round-trip Energy Efficiency | 100 to 103% | 80 to 90% |
| Storage Life Expectancy | 40 to 50 years minimum | 7 to 10 yrs |
| Grants or Tax Credits to Justify Economics | None needed | Needed |

## TES versus Batteries

Some direct comparisons between battery storage and TES are illustrative, in fact even quite eye-opening, when considering not only initial capital costs, but also other performance characteristics.

> As Douglas Reindl states in the *ASHRAE Journal,* "Although battery technologies are continuing to evolve and improve, their costs are high [...]. ASHRAE's recently completed research project, RP-1607, found that thermal energy storage is currently the most cost-effective means to enable greater renewable energy generation deployment" (*ASHRAE Journal*, February 2018, p. 20).

In December 2017, the State of Massachusetts' Energy Storage Initiative (ESI) announced an award of $20 million in state grants to 26 energy storage projects, which were also supported by $32 million in private "matching funds." The individual projects had varied storage capacities but average unit capital costs as follows:

- Mechanical flywheel storage $948/kWh
- Battery storage (primarily Li-ion batteries) $656/kWh
- Thermal energy storage (TES) $240/kWh

In late 2017, Harvard University initiated a CHW TES project to be online in 2018 as part of the district energy system serving its new Allston Campus in Boston. The TES capacity is 13,392 ton-h (9 MWh of equivalent electrical storage). Compared to an equivalent Li-ion battery installation, the TES project provides the benefits listed in Table 6.9.

The University of Nebraska-Lincoln installed a CHW TES system, online in 2018, serving its City Campus cooling network. By shifting much of the campus' electric chiller operation from day to night, the TES installation provides the equivalent of 39 MWh of electrical storage, while also providing peak daytime cooling capacity (over 4,000 tons required to meet near-term load growth, plus an additional 4000 tons for future growth). A comparison of the actual TES installation with an equivalent (hypothetical) battery storage installation is shown in Table 6.10.

As George Berbari explained at the 2016 *District Cooling and Trigeneration Summit*, "Chilled-water and hot-water stratified thermal storage is the world's most viable storage technology."

**Table 6.10** Comparison of the Hypothetical Performance of a Li-Ion Energy Storage Installation with the Actual Performance of an Installed CHW TES System

| | (Hypothetical) Lithium-Ion Advanced Batteries | (Actual, 2017–2018) Chilled Water (CHW) Thermal Energy Storage (TES) |
|---|---|---|
| Storage Element | | |
| Peak Cooling Discharge Rate | Not applicable | 8333 tons |
| Peak Electric Discharge Rate | 6.25 MW | 6.25 MW equivalent |
| Duration at Peak Discharge Rate | 6.24 hrs | 6.24 hrs |
| Net Storage (Thermal) | Not applicable | 52,000 ton-h |
| Net Storage (Electrical) | 39.0 MWh | 39.0 MWh equivalent |
| Storage Unit Capital Cost | $350/kWh | $100/ton-h |
| Storage Capital Cost | $13.65 million | $5.20 million (38% of batteries) |
| Full System Capital Cost | $27.3 million | $11.7 million (43% of batteries) |
| **Full System Unit Capital Cost** | **$700/kWh** | **$300/kWh (43% of batteries)** |
| Additional Chiller Plant | | |
| Necessary New Capacity | 4016 tons | Zero (TES already provides 8333 tons) |
| Unit Capital Cost | $2,900/ton | Not applicable |
| Installed Capital Cost | $11.6 million | Zero |
| **Tot Cap Cost (Storage + Chillers)** | **$38.9 million** | **$11.7 million (30% of batteries)** |
| Storage Life Expectancy | 7 to 10 yrs | 40+ yrs |
| **40-Year Life Cycle Capital Cost** | **$132.4 million** | **$11.7 million (9% of batteries)** |
| Round-trip Energy Efficiency | 80 to 90% | Near 100% |

## REFERENCES


Andrepont, J.S. 1994. Capital, operating, and energy savings from the use of chilled-water storage with district cooling. Presented at the ASHRAE Annual Meeting, Orlando, FL.

Andrepont, J.S. 2000a. Analysis and optimization of thermal energy storage for district cooling applications. *Proceedings of IDEA's 15th Annual Cooling Conference*, September.

Andrepont, J.S. 2000b. Long-term performance of a low temperature fluid in thermal storage and distribution applications. *Proceedings of the 13th Annual College/University Conference of the International District Energy Association*.

Andrepont, J.S. 2005a. Cool trends on campus: A survey of thermal energy storage use. *District Energy* 91(1):25–30.

Andrepont, J.S. 2005b. Thermal energy storage (TES) in district cooling utilities—Overview and findings from a survey of applications. Presented as an Introduction to a Panel Discussion at *IDEAS's Annual Conference,* June.

Andrepont, J.S. 2005c. Developments in thermal energy storage: Large applications, low temps, high efficiency, and capital savings. *Proceedings of World Energy Engineering Congress (WEEC)*, *Association of Energy Engineers (AEE)* September.

Andrepont, J.S. 2005d. Thermal energy storage: Capturing both operating and capital cost savings. *District Energy* 91(4):22. Westbourough, MA: International District Energy Association.

Andrepont, J.S. 2018a. Personal data of TES installations in district cooling systems. Lisle, Illinois: The Cool Solutions Company.

Andrepont, J.S. 2018b. *Energy Storage: A clear need for the power grid—But how best to achieve it?* A TICA white paper. Napierville, IL: Turbine Inlet Cooling Association. www.turbineinletcooling.com.

Andrepont, J.S., and M.W. Kohlenberg. 2005. A campus district cooling system expansion: Capturing millions of dollars in net present value using thermal energy storage. *Proceedings of the 18th Annual Campus Energy Conference*, IDEA, March.

ASHRAE. 2016. Chapter 51, "Thermal storage." In *ASHRAE handbook—HVAC systems and equipment*. Chapter 51. Atlanta: ASHRAE.

Bahnfleth, W.P., and W.S. Joyce. 1994. Energy use in a district cooling system with stratified chilled-water storage. *ASHRAE Transactions* 100(1):1767–78.

Bahnfleth, W., J. Song, and J. Climbala. 2003a. Thermal performance of single pipe diffusers in stratified chilled-water storage tanks. ASHRAE Research Project RP-1185 final report. Atlanta: ASHRAE.

Berbari, G. 2016. District cooling sustainable design. Presented at the 2016 *District Cooling and Trigeneration Summit,* Dubai, United Arab Emirates.

Borer, E., and J. Schwartz. 2005. High marks for chilled-water system: Princeton upgrades and expands. *District Energy* 91(1):14–18. Westbourough, MA: International District Energy Association.

EPRI. 1999. *Chilled water thermal energy storage: Emerging chilled water density depressants for capacity enhancement*. Technical Brief TB-114676. Palo Alto, CA: Electric Power Research Institute.

Fiorino, D. 1992. Thermal energy storage program for the 1990s. *Energy Engineering* 89(4).

Flory, J. 1995. TES reduces source energy use and air emissions. *Proceedings of ASHRAE Winter Meeting*.

Gansler, R. 1999. *Energy and environmental impacts of space conditioning systems*. Report on ASHRAE Research Project 991-RP, University of Wisconsin HVAC&R Center, Madison, Wisconsin.

Gatley, D.P. and Mackie, I. 1995. *Cool storage open hydronic systems design guide*. TR-104906, Research Project 3280-03 final report. Palo Alto, CA: Electric Power Research Institute.

Kosowatz, J. 2017. Electric enabler. *Mechanical Engineering.* December 2017139(12). New York: The American Society of Mechanical Engineers.

Musser, A., and W. Bahnfleth. 1998. Evolution of temperature distributions in a full-scale stratified chilled-water storage tank. *ASHRAE Transactions* 104(1):55–67. Atlanta: ASHRAE.

Musser, A., and W. Bahnfleth. 1999. Field-measured performance of four full-scale cylindrical stratified chilled-water thermal storage tanks. *ASHRAE Transactions* 105(2):218–30. Atlanta: ASHRAE.

Musser, A., and W. Bahnfleth. 2001. Parametric study of charging inlet diffuser performance in stratified chilled-water storage tanks with radial diffusers: Part 2—Dimensional analysis, parametric simulations and simplified model development. *ASHRAE Transactions* 107(2):41–58. Atlanta: ASHRAE.

Nix, J. 2008. Thermal energy storage time of day impact on power plant emissions. Presented at the ASHRAE Annual Meeting.

Potter, Jr., R.A., D.D. Boettner, D.J. King, and D.P. Weitzel. 1995. ASHRAE RP-766: Study of operational experience with thermal storage systems. *ASHRAE Transactions* 101(2). Atlanta: ASHRAE.

Reindl, D.T., D.E. Knebel, and R.A. Gansler. 1994. *Characterizing the marginal basis source energy emissions associated with comfort cooling systems*. TSARC 94-1. Madison, WI: University of Wisconsin HVAC&R Center.

Reindl, D., A. Van Asselt, G. Nellis, and S. Klien. 2017. Design and utilization of thermal energy storage to increase the ability of power systems to support renewable energy resources. ASHRAE Research Project 1607 final report. Atlanta: ASHRAE.

Stewart, W.E. 2001. Operating characteristics of five stratified chilled-water thermal storage tanks. *ASHRAE Transactions* 107(2):12–21. Atlanta: ASHRAE.

Tabors Caramanis & Associates. 1995. *Source energy and environmental impacts of thermal energy storage*. Report prepared for the Thermal Energy Storage Systems Collaborative. Sacramento, CA: California Energy Commission (CEC).

Tran, N., J.F. Kreider, and P. Brothers. 1989. Field measurements of chilled-water storage thermal performance. *ASHRAE Transactions*. 95(1):1106–12.

Wildin, M.W. 1991. *Flow near the inlet and design parameters for stratified chilled-water storage*. ASME 91HT27. New York: American Society of Mechanical Engineers.

Wildin, M.W. 1996. Experimental results from single-pipe diffusers for stratified thermal energy storage. *ASHRAE Transactions* 102(2):123–32.

Wildin, M.W., and C.R. Truman. 1989. Performance of stratified vertical cylindrical thermal storage tanks—Part 1: Scale model tank. *ASHRAE Transactions* 95(1):1086–95. Atlanta: ASHRAE.

Winters, J. 2018. By the numbers: Grid energy storage gets cheaper. *Mechanical Engineering.* April 2018, 140(4):28–29. New York: The American Society of Mechanical Engineers.

Yoo, J., M. Wildin, and C.W. Truman. 1986. Initial formation of a thermocline in stratified thermal storage tanks. *ASHRAE Transactions* 92(2A): 280–90. Atlanta: ASRHAE.

## BIBLIOGRAPHY

Dorgan, C.E., and J.S. Elleson. 1993. *Design guide for cool thermal storage*. Atlanta: ASHRAE.

Electrowatt-Ekono Oy 2002. Optimization of cool thermal storage and distribution. Final Report, ISBN 90-5748-025-5, District Heating and Cooling Annex VI, International Energy Agency (IEA).

Elleson, J.S. 1997. *Successful cool storage projects: From planning to operation*. Atlanta: ASHRAE.

Hyman, L.B. 2011. *Sustainable thermal storage systems: Planning, design, and operations*. New York: McGraw Hill.

Vadrot, A., and J. Delbes. 1997. *District cooling handbook*. European Marketing Group—District Heating and Cooling.

# 7 Instrumentation and Controls

## GENERAL

An integrated control and monitoring system for a district cooling plant (DCP) is designed to maintain the complete DCP process operations and functionality. It is also interfaced with the DCP building services controls, and finally the integrated control system is fully mapping/monitoring the serviced energy transfer stations (ETS) provided at each served consumer.

The choice of control system robustness and quality will often be influenced by the relationship between the building owner and the district cooling service provider. Industrial-grade controls with redundancy of sensors are always recommended for the highest level of quality and accuracy in accounting for building cooling use as well as remote monitoring, metering, and control of the operation of the building and the ETS when system components are not owned by a common entity. Higher quality and accurate instrumentation and control is required due to the "custody transfer," and metering of the chilled water and components should be utility/industrial grade. However, a campus DCS, where the buildings, the DCPs, and distribution system have common ownership, a less expensive option such as commercial-grade controls and little or no redundancy may be considered. Also, where buildings are being retrofitted to DC and may also contain an existing building management system (BMS), the owner may opt to use that system interfacing with the existing sensors and controls as a cost-control measure. For a district cooling service provider independent from the building owner, the choice will nearly always be an industrial-grade, microprocessor-based system, and communication should be via high speed/standard communication media (protocol and topology) that can be easily integrated with any other control system in the future.

The ability to deal effectively with the efficiency robing and costly impacts of low $\Delta T$ syndrome (see Chapters 2, 3, and 5) will require high levels of reliability and accuracy in instrumentation and controls. This should be considered as an additional justification for redundancy, and industrial-grade controls and instrumentation regardless of the ownership relationship between the buildings and the DC service provider.

This chapter focuses on the design of a typical DCP process control system. It covers process automatic control, global monitoring, and energy conservation. The system is capable of optimizing all the available mechanical, electrical, and plumbing (MEP) systems and equipment operations. The monitoring and control system also identifies interfaces and relations with other control systems and devices such as building management systems (BMS) and ETS controls.

## BMS OR SCADA?

This is a typical question that arises each time a system control subject is opened. For clarity, let us begin by providing a short description of both systems.

BMS is a system of microprocessor-based controllers (DDC) that is integrated via strong/standard data management and acquisition systems and strengthened by software interfaces and dynamic graphical presentation. Similar to the BMS (building management system), the SCADA (supervisory control and data acquisition) is also made of microprocessor-based controllers (but using PLC controllers) with stronger data management, interfacing capability, and dynamic graphical presentation.

BMS is widely used for human comfort and safety, but for machinery systems efficiency and performance, SCADA is the standard for such industrial application.

### Major Differences

SCADA can maintain all BMS functions and capabilities and exceed BMS performance in the following areas:

- Reliability: because it is created to work in harsh industrial conditions and also because of the capability of having full data management and communication redundancy.
- Accuracy: for its higher processing speed and cache, in addition to the ability to have multiple processor comparison action.
- Power: because of its advanced data error correction tools.
- Security: for its higher access security, in addition to a hardware and connection verification tool.
- Maintainability: for being capable of performing hot-swap modules (in-operation replacement for modules), and the multibus structure for sectional management.

SCADA's higher performance has an impact on the control system initial cost, especially when using industrial standard-field devices that allow a more robust system when compared to a BMS. Industrial-quality field devices are strongly recommended for all control functions on a DCS—at the plant, within the distribution system, and at the consumer interconnection.

### Summary

However, for applications that do not require high accuracy and fast performance, such as MEP (mechanical, electrical, and plumbing) building services controls, using a PLC/SCADA system with industrial-grade field devices will not introduce any additional improvement. Moreover, it will impose additionally unnecessary cost to the control system. In this case, using DDC/BMS with non-industrial-grade field devices is normally an appropriate choice.

In case of a central plant for district cooling, the field devices to be used even for a DDC/BMS system should be industrial grade for their better performance, accuracy and reliability. In this case, the cost difference for using PLC/SCADA rather than DDC/BMS will be reduced to a very reasonable overall cost differential. The modest cost differential has resulted from the current climate of rapid hardware/software development and upgrade that has reduced the huge difference (in both cost and capabilities) between the two systems, especially when using field devices of the same quality and features.

## SYSTEM COMPONENTS

Controllers must be all connected together through a standard, very high quality communication network and that network must also be used for the connection to the central operator workstation (COWS) and servers, located at the control room.

The major components of the system include, but are not limited to, the following:
- Control console
- COWS (Engineering and HMI [human machine interface])
- Data storage unit (database servers)
- Data/Software interface units
- System software utilities
- Alarm and report printers
- Plant data communication network and units
- Network master controllers and zone managers (as applicable)
- PLCs
- I/O (input/output) modules
- Control panels enclosures, including all accessories
- HHOT/POT (hand held operator terminal/portable operator terminal)
- Field instruments

The ICMS (integrated control and monitoring system) is normally divided into four layers:
- Management layer
- Communication layer
- Automation layer
- Field instruments layer

The purpose and function of each of these layers is described in the following sections.

## Management Layer

The management layer consists of the COWS, which includes the HMI workstations, the database/servers, printers, and UPS (uninterruptable power supply). Through this layer, the operator can oversee the complete plant operations. The management layer is also responsible for performing all reporting, trending, alarming, sequence controls, and classification and storage of all events, alarms, and information.

The COWS located at the plant control room will also be responsible for communicating with the remote ETS controllers and any operator-required SCADA systems, as well as the plant BMS and fire alarm system.

## Communication Layer

Inside the DCP, the ICMS controllers will communicate over a redundant network for data transmission security over a peer-to-peer communication structure protocol. The communication layer also includes a simplex ethernet communication media, which is responsible for connecting the controllers with the management layer equipment over the fiber optic communication network.

The integrated communication layer includes all required hardware (such as: a dual-communication controller, routers, tabs, cables, switches, etc.), software (supporting client/server architecture, SCADA/HMI, etc.) and accessories that are required for complete and successful data management through the DCP. Additionally, sensing and measuring devices can transfer their measurements and readings to the controller via standard field communication media.

For communication between the DCP and ETS controllers, a standard industrial-grade fiber optic communication media is normally used. The ETS controller ethernet communication bus is connected to the FO communication media (building connectivity rack/patch panel) in order to transfer the instantaneous measurements and calculations to the DCP application server. It can also receive overrides (to the valve actuators) from the DCP

COWS. A simplex network for ETS controllers communication is used. Additionally, wireless HHOT can be used to access ETS controllers without the need for entering any customer property.

## Automation Layer

The automation layer encompasses the control chassis, which includes the dual microprocessor PLC unit complete with control chassis, communication ports, I/O modules, terminal-blocks, mounting kits, enclosures, power supply, and any required accessories. The automation layer controllers support standalone capability by including the serviced equipment/systems sequence of operation, parameters, and energy conservation techniques. RTUs (remote terminal units) are used for the ETS room's controls.

All controllers normally contain backup batteries required to maintain system parameters and events in case the system goes down.

For DCP, the ICMS will be responsible, at a minimum, for the following functions:

- Control and monitoring of the chillers (enable/disable, status, alarms)
- Control and monitoring of the CHW pumps (primary/secondary, as applicable)
- Interfacing via the data link with the chillers
- Control and monitoring of the cooling towers (CTs) fans and valves
- Control and monitoring of the condenser-water pumps
- Control and monitoring of the makeup water system
- Control of motorized valves and monitoring of valve status
- Monitoring of all temperature measurements (supply header temperature, return header temperature, chiller supply temperature, etc.)
- Monitoring of flow measurements as applicable (common pipe, main CHW return header, etc.)
- Monitoring of all pressure measurements (CHW supply headers, CHW return risers, etc.)
- Monitoring of chiller flow switches
- Monitoring of expansion tank levels
- Monitoring of chemical treatment system
- Monitoring of CTs sump level indicators

The relationship between the ICMS and the DCP mechanical/electrical equipment will normally be as is shown in Table 7.1.

## Field Instruments Layer

This layer includes all the industrial-grade field-installed instruments such as motorized-valves, actuators, temperature transducers, pressure transducers, flowmeters, etc., as well as a connection to any voltage-free-contacts (inputs/outputs) including all installation accessories and connections up to the control chassis. Field instruments can be categorized as follows.

- Industrial-grade sensing and measuring devices such as:

  *Temperature*: Electronic instruments with interchangeable metallic (platinum, nickel, or nickel alloy) RTD (resistance temperature detectors) sensing elements (three or four wire, 1000 ohm). Temperature sensors/transmitters are recommended to be immersion type with stainless steel thermal-wells.

  *Pressure*: Piezoresistance sensors utilizing solid-state circuitry (where resistance changes with pressure) with sensitivity of 6.0 mV/psi (0.41 mV/bar) and a low temperature effect on span (not exceeding ±0.5% full scale).

**Table 7.1** Nature of Device Interfaces with the ICMS

| System Description | Control | Monitoring | Software Interface |
|---|---|---|---|
| Chillers | • | • | • |
| CTs | • | • | • |
| CHW Pumps | • | • | • |
| Condenser-Water Pumps | • | • | • |
| Thermal Storage Tanks | • | • | |
| Filtration Unit | • | | |
| Air Separators | • | | |
| Expansion Tanks | • | • | |
| Makeup Water System | • | • | |
| Chemical Treatment Stations | | • | |
| Water Filtration Systems | | • | |
| Power Distribution | • | • | • |
| Building Management System | | • | • |
| Life Safety and Security Systems | | • | |

*Flowmeter*: Usually velocity meters are used for CHW flow measurements (such as turbine, electromagnetic, or ultrasonic). Electromagnetic induction-type meters are often preferred due to their high accuracy (±0.2% to 0.5% of rate), low pressure loss (≤3 psi or 0.2 bar), and easy installation requirement (upstream straight-piping length of 5 to 10 diameters and downstream straight-piping length of 3 diameters) in addition to their reasonable cost compared to ultrasonic meters. Electromagnetic flowmeter material should be stainless steel tube/stainless steel conical raised electrodes made to produce AC signal proportional to liquid flow.

- Industrial-grade switching devices such as:

Solid-state switches, switching relays, voltage-free contacts, and signal conditioners.

- Energy meters:

A battery operated energy calculator is a microprocessor device that is connected to flowmeter and temperature sensors, in order to calculate the thermal energy based on the measured differential temperature between supply and return. The energy calculator should have a built-in display and keypad for data entry and configuration, it also should provide at least 13 months of data storage capacity.

- Valves and actuators:
  - Butterfly valves for on/off operation (constant flow circuits).
  - Globe valves for two-way modulating action (variable-flow circuits).
  - Electric Actuators (single phase).

## SYSTEM CONFIGURATION

### System Structure

The ICMS operator workstation is normally an open-protocol, high-performance SCADA workstation that is interfacing with the main plant and ETS PLCs through an application server, in addition to any other control panel/system.

The ICMS communication and automation layers can be configured in several architectures, the most popular forms are described as follows:

#### Central Processing Panel and Distributed I/O Modules

This configuration is probably the widest used configuration since it is very efficient, stable, and maintainable, and all this with a reasonable initial cost. For this configuration, the system structure is as follows: the ICMS's central control panel shall be located in the control room in a freestanding cabinet. Remote I/O chassis are normally located on each floor and interconnected with PLCs through a high speed redundant control network. Input/output interfaces normally use industrial standard protocols for processing field input/output signals, excepting for discrete input/outputs. Each I/O chassis is segregated within its operating group from similar equipment to prevent other equipment from simultaneously stopping at one of the I/O card failures. For more details refer to Figure 7.1.

#### Distributed Processing Panels

This configuration is as follows: The ICMS's central control panel is located in the control room within a freestanding cabinet. Remote control panels are located on each floor, each including its own processing unit (PLC control chassis) complete with I/O modules. These locally distributed panels normally communicate with each other with the central control panel. For more details, refer to Figure 7.2.

### Plant Control Room

The ICMS main components must normally be installed in an air-conditioned and temperature-regulated control room and connected to a UPS/emergency power supply.

The control room dimensions and location must be selected to guarantee:

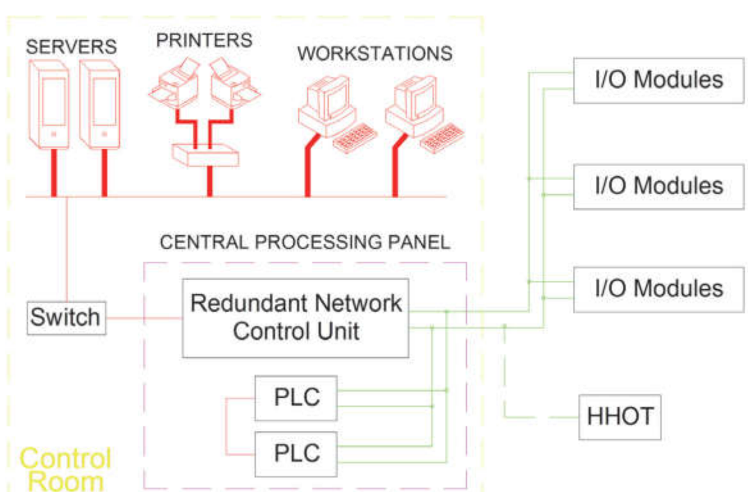



**Figure 7.1** Central processing panel and distributed I/O modules.

- Enough space for the room's furniture including but not limited to console, chairs, storage closet, etc.
- Operators can directly view the plant main hall through a glass-wall
- Housing the main control/communication panel, network/telephone routers, etc.

Audible and visual alarm-annunciation, in addition to a labeled alarm silence switch and a labeled lamp-test push button, should be located in the control room.

A provision for a remote start/stop and an emergency stop from the master control station is recommended for inclusion in the control room.

A suitably sized console/table should be provided in the control room. Figure 7.3 indicates a model of DCP control room layout. All details of the table should be contained in

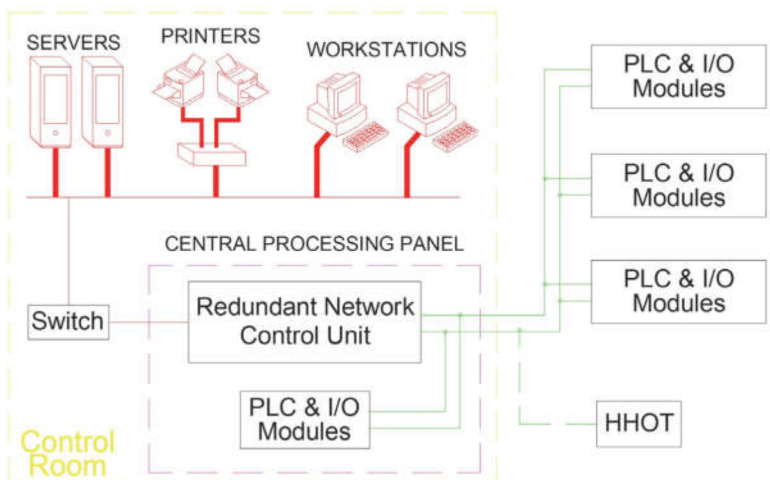



**Figure 7.2** Distributed processing panels.

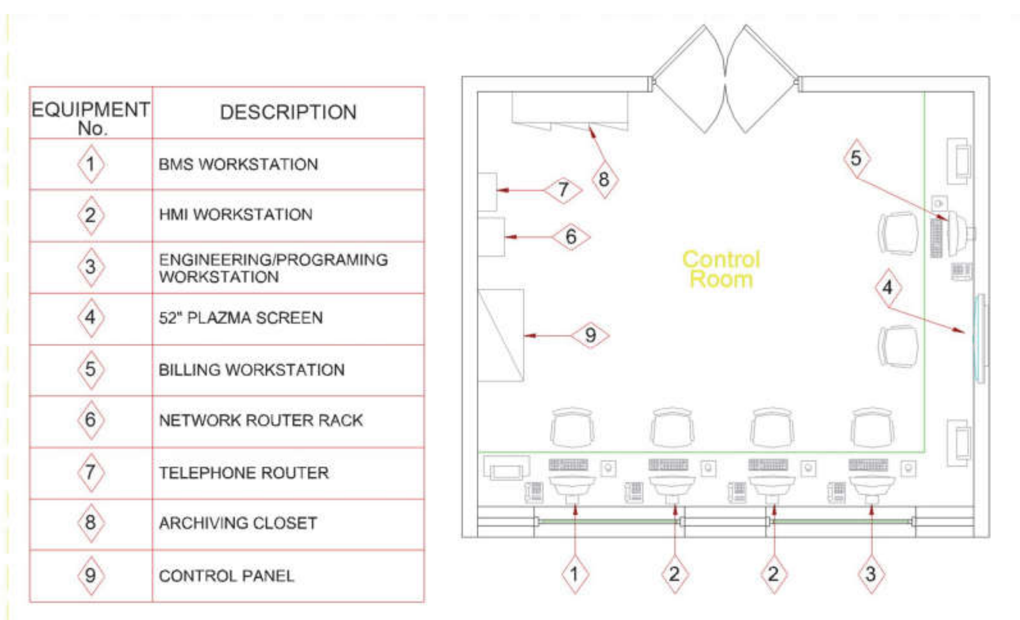



**Figure 7.3** Control room layout.

the control and monitoring shop drawing. The control console/table should be suitable for housing the CPU's, servers, printers, keyboards, flat panel visual display units, a wall-mountable large display unit, and a mouse and telephone. A minimum of 65 $ft^2$ (6 $m^2$) of flat surface work area such that two operators can work concurrently is recommended.

## System Features and Capabilities

The system performance capabilities and basic features should include at a minimum:

- Open system configuration and components
- Standard equipment that has an international recognition for quality
- User-friendly graphical representations for all measurements, events, and alarms
- Generating reports, work orders, and maintenance routines
- Hardware and software with a minimum 20% spare capacity
- Open expandability/upgrading capability
- Secured system access, performance, and data sharing/transfer
- Optimization of energy utilization
- Events, alarms, and information classification and storage
- Stand-alone capability in the event of network failure
- Local and remote access to each controller and COWS
- Communicating with third party vendor equipment for all mechanical and electrical aspects
- Global monitoring for all MEP systems and equipments indications, alarms, and measurements
- Selectively turning on or off all mechanical HVAC equipment and lighting
- Resetting temperatures for occupied or unoccupied conditions allowing automatic ramping of temperature setpoint
- Allowing dynamic analysis to verify efficient equipment operation
- Recording of all systems' alarms, events, operator commands, and power/thermal energy consumption of different zones and spaces, as well as the entire building's domestic water, CHW, and electricity feeders

# OPERATION PHILOSOPHY

## The ICMS for Plant Management

The complete ICMS is a microprocessor-based industrial-grade PLC system, which, through a resident program, (on a completely self-contained stand-alone basis) controls and monitors all the process mechanical systems/devices such as chillers, CHW pumps, CTs, etc.

Standard BACnet® interface modules should be provided for communicating with plant building services control system (BMS) and the fire alarm system (note that other proprietary protocols are also available as an alternative to BACnet®). All building services measurements, alarms, status, operator interaction, etc. should be graphically represented on the BMS workstation and archived at the data servers for the historical backup of all events and actions.

Package equipment control should be performed by the local controller mounted in the local control panel. Main equipment packages such as soft starters, and variable-speed drives should be integrated into the ICMS through BACnet® standard communication protocol for monitoring purposes.

Critical signals, related safety interlock, and control command should be directly connected to the ICMS through hardwired input/output to prevent any time delay of the communication system.

Each chiller should include a local control panel with HMI that shall provide manual and automatic operation of chiller start-up, operation, normal shutdown, and emergency shutdown, as well as the chiller's safety protection interlocks.

Chillers, CTs, CHW pumps, condensate water pumps, makeup water pumps and makeup tanks should be automated to the point that the operator simply has to select the item desired to start, stop, and operate, after which the ICMS provides the necessary logic, safeguards, and permissions. The availability of each device should be tracked and indicated to the operator. The ICMS should be configured to prompt the operator of invalid selections, of the selection of a device already in service, or of a device that is not available (i.e., not in remote, auto, or fault). The opportunity to change the selection should be available before final acceptance keystroke is made.

Integrating the ETS rooms' operating data through ethernet/fiber optic communication media with the data server at the plant control room should be considered.

All auxiliary systems such as filtration, water treatment, blowdown, boosters, and MEP services should be considered for full automation and monitoring by the ICMS.

## Control Philosophy Statement

The ICMS should be configured in such a way as to be able to manage the chillers, CTs, and associated equipment to achieve the following objectives:

- Control the CHW design-supply temperature continuously at a nominal temperature value (setpoint) with an acceptable deviation of a decimal fraction under normal operating conditions. The nominal design-return temperature depends on the ETS condition and functionality as shown in the Chapter 5 discussion on end user interface.
- Manage the operation of chillers, CTs, and associated equipment within their optimum energy-efficient ranges.
- Sequence cycle the chillers, CTs, and all other associated equipment to obtain a balanced capacity condition between the system cooling load and the primary plant-cooling generation capacity.
- Provide the operator with the capability of viewing all trends and values as well as overriding any command, changing setpoints, and all other parameters that may be overridden (such as time schedule, delay-time, etc.) using the HMI application, which is under software access protection (e.g., user ID and password). All operator-definable delays and setpoints should be governed by upper and lower safety limits, regardless of the operator access level authority.
- Limit the plant's demand capacity by receiving external hardwired analog signal input from the local control panel of the equipment. This signal should be prespecified for minimum and maximum running capacity limits, which is applied to chillers, CTs and VFDs.
- Communicate chiller control unit information/parameters with the HMI via standard protocol/gateway; each chiller should be connected with a dedicated gateway translator, which allows a minimum of 50 points per chiller.
- Ensure that certain control-loops concerning equipment safety are uninterruptable by the operator (override), such as stopping of the primary pump, condenser pump, closing valves, or CT, without stopping the associated chiller or overriding delay periods.
- Provide overrides for other control-loops that the operator may interrupt, such as the cycling order for start/stop.

- Limit chiller, CTs, and associated equipment cycling to within recommended manufacturer's cycle times and allow for maximum stability in loop temperatures and water flows.
- Manage the operation of chillers, CTs, and associated equipment within their safe limitations and protected sequence/operation.
- Provide for warning messages/alarms that are initiated after an adjustable time delay as well as high and low limits alarms for each sensor measurement.
- Ensure the required makeup water is provided to the condenser-water circuit.
- Manage condenser-water circuit blowdown and bleed operation from CT basins and/or pipes.
- Provide flexibility in the control system design since not all control loops or motor controls need to reside in the main ICMS environment. Some devices will be controlled by a subsystem that is interfaced with PLC (for monitoring purposes only) via a hard-wire or software connection such as makeup water filtration system, chemical treatment, etc.
- Ensure that the operator is able to override any equipment operation for testing purposes or for manual operation. However, all safeguards and operation precautions must automatically govern such operations in order to avoid human mistakes resulting in equipment damage or operational accidents.
- Provide for monitoring of the plant energy/flowmeters. Input signals into the ICMS to monitor the plant distribution flow via the supply side flowmeter. This includes monitoring the supply flow as well as supplied energy by communicating with the energy meter. The ICMS should integrate the flow signal and display both the flow rate and the totalized volume. Totalized volume should include absolute total (non-resettable) and total volume for the last 24 hours.
- Provide metering and logging of both energy consumption and produced energy of:
  - Each individual chiller.
  - Each individual system comprising a chiller or chiller pair and its ancillaries (chiller water pump(s), condenser-water pump and proportional CT), and the whole plant.
  - The whole DCS (based on refrigeration energy metered at ETSs).

A basic sample of sequence of operation for a chiller plant is provided below (Courtesy of Goss Engineering Inc.).

### Chilled and Condenser-water Systems: Sequence of Operation

The CHW system is a variable-primary system consisting of three chillers with three dedicated condenser-water pumps and three headered CHW pumps. The condsenser-water system consists of three CTs. All pumps and tower fans are equipped with individual VFDs. The system is DDC controlled with electric actuation.

The system operates as follows (all suggested setpoints and settings are adjustable).

### CHW Pump Alternation

CHW pumps alternate to equalize runtime. Selection of the lead, second, and third CHW pump is evaluated on a weekly basis. The pump with the least runtime is the lead pump. The pump with the most runtime is the third pump and the remaining pump is second.

### Chiller Alternation

Chillers alternate to equalize equipment runtime. Selection of the lead, second, and third chiller is evaluated on a weekly basis. The chiller with the least runtime is the lead chiller. The chiller with the most runtime is the third chiller and the remaining chiller is second.

### CT Alternation

The CTs shall alternate to equalize runtime. Selection of the lead, second and third CT is evaluated on a weekly basis. The CT with the least runtime is the lead CT. The CT with the most runtime is the third CT and the remaining CT is second.

### CHW System Enable

The CHW system-enable point is controlled either manually by the operator or by a program function (i.e., time of day). If the CHW system-enable point is on, the lead chiller start sequence is activated.

### Lead Chiller Start

The chillers shall be provided with integral operating and safety controls. A flow switch is mounted in the CHWS and condenser-water supply to each chiller, furnished by the chiller vendor and installed by a mechanical contractor to monitor the status of flow through the chiller.

On system enable, the lead chiller's CHW isolation valve shall open and the lead CT isolation valves shall open. After a delay, the lead CHW pump and the associated condenser-water pump shall start. Both pumps shall ramp to 100% speed. The chiller shall be enabled. If flow is sensed by the chiller flow switches, the chiller shall execute its internal start up sequence and chiller shall start.

The CHWS supply temperature setpoint is set to 42°F (5.6°C) (adjustable) and can be manually adjusted upward by the operator.

### Chiller Lead/Lag

The chillers shall operate in lead/lag mode. If the system demand is greater than 90% of the capacity of the chiller for more than 20 minutes or the supply water temperature exceeds 44°F (6.7°C) for more than 5 minutes, the second CHW pump and the second condenser-water pump shall be enabled at 100% speed. The second chiller's CHW and condenser-water valves shall open, and the second CT's isolation valves shall open. After a delay, the second chiller shall be enabled.

If the system demand is greater than 90% of the capacity of the two chillers for more than 20 minutes or the supply water temperature exceeds 44°F (6.7°C) for more than five minutes, the third CHW pump and the third condenser-water pump shall be enabled at 100% speed. The third chiller's CHW and condenser-water valves shall open, and the third CT's isolation valves shall open. After a delay, the third chiller shall be enabled.

Should the system demand be less than 40% of the current online capacity for more than 10 minutes and the CHWS temperature is less than 46°F (7.8°C), the third system shall be stopped in the following sequence. The third chiller shall be disabled. The third CT shall be disabled after the chiller has stopped. After a 120 second delay, all associated CT isolation valves shall be closed. After they prove closed, the third chiller's isolation valve shall close. The third CHW pump and the associated condenser-water pump shall stop.

Once the third system has been disabled, should the system demand be less than 40% of the current online capacity for more than 10 minutes and the CHWS temperature is less than 46°F (7.8°C), the second system shall be stopped in the following sequence. The second chiller shall be disabled. The second CT shall be disabled after the chiller has stopped. After a 120 second delay, all associated CT isolation valves shall be closed. After they prove closed, the second chiller's isolation valve shall close. The second CHW pump and the associated condenser-water pump shall stop.

If a chiller goes into alarm, the next available chiller is started.

### CHW Pump Control

Once the lead chiller's isolation valve proves open, the lead CHW pump starts at 100%. Once the lead pump is enabled and running at 100% speed, the lead pump's speed shall be allowed to modulate between the maximum speed and minimum speed. The minimum speed shall be hard programmed into the VFD. The minimum speed shall be determined by the water balancer as the minimum speed required to provide minimum flow to the chiller. The VFD modulates pump speed to maintain system differential pressure of 20 psi (138 kPa) (adjustable, to be determined by water balancer) as sensed near the end of the secondary piping run.

The second CHW pump shall start once the second chiller's isolation valve proves open. The third CHW pump shall start once the third chiller's isolation valve proves open. Once the pump is enabled and running at 100% speed, it shall be allowed to modulate as described above. All pumps shall run at the same speed to maintain the differential pressure at setpoint.

The system uses current switches to confirm the pump is in the desired state (i.e., on or off) and generates an alarm if the status deviates from start/stop control. If the pump goes into alarm, the next available pump starts.

### Condenser-Water Pump Control

Once the lead chiller's isolation valve proves open, the associated condenser-water pump starts at 100%. Once the associated pump is enabled and running at 100% speed, the associated pump's speed shall be allowed to modulate between the maximum speed and minimum speed. The minimum speed shall be hard programmed into the VFD. The minimum speed shall be determined by the water balancer as the minimum speed required to provide minimum flow to the chiller. The condenser-water entering and leaving temperature shall be monitored. The condenser-water pump speed shall modulate to maintain a fixed temperature differential across the condenser. If the temperature differential is higher than setpoint, then the pump speed shall ramp up. If the temperature differential is lower than setpoint, then the pump speed shall ramp down. Temperature differential setpoint is to be field-determined.

The second condenser-water pump shall start once the associated chiller's isolation valve proves open. The third condenser-water pump shall start once the third chiller's isolation valve proves open. Once the pump is enabled and running at 100% speed, it shall be allowed to modulate as described above.

The system uses current switches to confirm the pump is in the desired state (i.e., on or off) and generates an alarm if the status deviates from start/stop control. If the pump goes into alarm, the associated chiller system starts its shut down sequence and the next available pump starts.

### CT Control

Any time a specific CT cell isolation valve is opened, its temperature control sequence shall be enabled and the CT fan VFD shall be modulated to control the CT supply water temperature at setpoint. Should the main condenser-water-supply header temperature drop below an adjustable low limit setpoint, the fan will deactivate to maintain CT water-supply temperature. If the main condenser-water-supply header temperature remains below the adjustable low limit setpoint for 15 minutes (adjustable), then the chillers and associated equipment shall go through a normal shutdown process and an alarm shall display on the BMS.

The vibration switches shall be hardwired to the safety input of the fan VFD. In the event of excessive fan vibration, the VFD shall shut down and an alarm shall be displayed on the BMS.

## ICMS Global Monitoring and Alarming Procedure

The ICMS logic will monitor all equipment (pumps, chillers, etc.) for running status, auto status, and all the available alarm indications.

The ICMS will interface with electrical switchgear feeders, VFDs and soft starters as well as interface with utility-level electricity and gas meters as applicable, and the fire alarm system via communication link (software) using standard protocols. The protocols must allow the operator the capability of viewing data and storing data, and providing operational alarms and logs.

The logic transfers to the HMI all field instrument measurements and alarms (temperature, pressure, flow, and level transmitters, as well as flow, level, differential pressure switches, and end switches).

The ICMS should supervise operator access authority as well as operator commands, trends, and parameter adjustments (such as setpoints, high/low limits, etc.).

Each ETS should have a dedicated programmable logic controller (RTU) that will serve as the local control and monitoring system and will transmitting measured values and the control valve position back to the central plant control system.

Appropriate software interface should be provided with the following systems:

- Chiller control panels
- Electric power switchgears
- Soft starters
- VFDs
- Energy meters

The ICMS logic should transfer to the HMI all monitored points such as status, alarms, and measurements for the following subsystems/devices:

- Expansion tanks
- Makeup water and blowdown quantities
- Chemical treatment stations
- Filtration units
- Refrigerant leak detection
- Supply and return header temperatures, pressures, flow, and energy measurements
- Outdoor ambient air conditions (dry-bulb temperature/wet-bulb and relative humidity measurements)

## Interface with BMS

The ICMS should interface with the plant (BMS) via a high speed data link connection (software) that allows the plant operator to monitor all the important building services, operations, and alarms including:

- HVAC equipment (such as fans, air handlers, fan coil units, etc.)
- Fire protection equipment (such as fire fighting pumps, fire water level, waterless fire suppression system, etc.)
- Power systems (such as lighting, central batteries, etc.)
- Low current systems (such as fire alarm, access control, telephone/data, etc.)

Also, ICMS should monitor the following rooms' temperature measurements and high temperature level alarms:

- Control room
- Switchgear room
- Transformer rooms
- Motor control center (MCC) room

This data link should also guarantee that the plant operator is aware of the plant building services operation and instantaneous conditions, in addition to sharing some important hardwired signals such as refrigerant leak indication to the fire alarm system/BMS.

### Rotation Sequence

Any equipment to be considered in the rotation sequence (lead/lag sequence) should be in auto mode and non-alarm status.

Lead/lag sequence should be applied to each type of equipment such as chiller modules (chillers and associated pumps and valves), CTs, makeup pumps, etc. Lead/lag sequence should define the following:

- The next lead-to-start equipment.
- The next lead-to-stop equipment.
- Ignoring any failed devices on the previous selection.
- Any replacement to the next lead-to-start equipment, in case of starting equipment to replace any failed or failed-to-start equipment. (Lead-to-start equipment is considered the standby for the already operating/commanded to start equipment).
- The maximum permissible delays, and number of starts/stops.
- Any other factors that might be requested by the device manufacturer.

The lead/lag sequence should be defined based on the total runtime of each piece of equipment in order to guarantee equal total runtimes for all devices, as follows:

- Lead-to-start equipment is the one with the lowest total runtime from the off-duty devices.
- Lead-to-stop equipment is the one with the highest total run-time from the on-duty devices.

## ENERGY AND OPERATIONAL CONSIDERATIONS

### Condenser-Water Return Temperature Setpoint Reset

A lower controlled common condenser-water temperature has the following qualities:

- *Advantages*: a higher chiller efficiency (within chiller-allowable operating range), and it also results in increased chiller life and a reduction in condenser tube scaling.
- *Disadvantages*: this will lead to an increase in CT fan energy and increased airflow volumes (which increases the condenser-water contamination, and accordingly increases maintenance/water treatment works as well as water consumption).

Thus, optimum operating condenser-water temperature must be identified by considering each of these factors. This can be achieved either by using the chiller proprietary control schemes for dynamically optimizing reset set point (provided by the chiller manufacturer) or by power-ratio calculation of the optimized power consumption of all operating CT fans and chillers.

It may also be proposed to control the common condenser-water leaving temperature over a wide temperature range with fan-speed control that guarantees satisfying the condenser-water temperature at the reset value by limiting maximum speed from 50%—when the water temperature is at or below setpoint—proportionally up to 100% when temperature reaches the design wet-bulb temperature. This temperature-controlled cycle in addition to time delay should limit the cycling. With a common condenser-water leaving temperature above 95°F, all available CT fans will therefore remain in operation at full speed.

Where a reduced noise level is a consideration, staging all fans on should be considered. One CT fan (according to rotation sequence) will be executed to operate at low speed. On a demand increase, the next-to-start fan shall also be programmed to operate at low speed and so on until bringing on all available fans at minimum speed, according to condenser-water return temperature (increasing the number of operating CTs at low speed, instead of increasing fan the speed of the actual required number of CTs). Once all available fans are operational, the fans' speed modulates up in order to maintain the return condenser-water temperature setpoint. On a demand decrease, all the fans in operation will modulate to reduce speeds until reaching minimum speed and prior to activating the fan stopping sequence.

The maximum stopping time should be considered for condenser-water pumps and CTs, in order to prevent condenser-water stagnancy and avoid bacterial formation (such as *Legionella*). The ICMS should include the maximum stop time for each condenser pump/CT within the rotation sequence of the equipment.

## CHWS Temperature Setpoint Reset

CHWS temperature setpoint can be reset to a higher value based on the signal provided by the operator (via 4–20 mA external hardwired signal from the SCADA/HMI to reset CHWS temperature setpoint). This signal should be between prespecified minimum and maximum CHW leaving temperature setpoints. Automatic control mat be configured to suggest temperature reset and the action may require confirmation by the operator. A higher supply CHW temperature (within the applied ETS capacity limitations as dictated by seasonal load requirements) will result in the following energy considerations:

- *Positive*: an increase in chiller efficiency, as well as a reduction in site-wide cooling demand as a result of the reduced dehumidification capability of ETS (AHU/FCU [air handling unit/fan coil unit.) operation at an increased CHW temperature. This will in turn result in reductions in chiller, CT, and pump energy.
- *Negative*: an increase in secondary CHW flow rates with associated increase in pump horsepower. Also, a possible increase in site-wide fan energy on applied variable air volume systems may be expected.

It is normally accepted that the energy savings listed under positive considerations exceed the energy increases listed under negative ones. The supply CHW temperature should therefore be kept as high as possible, limited only by the users' AHU/FCU capacity limitations to achieve the required temperature and humidity comfort levels (see Chapter 5 for a discussion of constraints imposed by the end user equipment).

## TES Tanks

TES will help ensure that chillers operate near optimum efficiency (usually close to 95% of load) by storing the excess flow of the primary circuit over the secondary in part-load condition, the energy stored inside the TESs will be used to satisfy the load whenever the operation is out of the peak expected period, and under one condition, which is that the thermal energy stored in TES is above the minimum level (see Chapter 6 for a complete discussion of TES). Control logic must take into account the minimum energy consumption and costs based on load leveling and/or peak shaving at minimum outdoor wet-bulb temperature. Also, control logic should take into account the availability of the full contract capacity at the peak load.

# 8 Operations and Maintenance

## INTRODUCTION

The importance of operations and maintenance (O&M) practices to the safety and success of a district cooling system (DCS) cannot be overemphasized. While attention is necessarily focused on the conception, design, and construction of district energy systems, it's worth noting that many multiples of years in a DCS project's life cycle elapse after project delivery and are concerned with its continuing care and upkeep. Relative to even a protracted development/design/construction timeline, continuing O&M will span decades, given the type of infrastructure investment a DCS embodies. ASHRAE (2015) projects primary chiller production equipment median life expectancy at 23 to more than 25 years, at the longest. However, real-world economics and an owner's imperatives based on site-specific circumstances often mean that this equipment will be run for years, if not decades, in excess of such forecasts. Owners and operators are acutely aware of this. After the project, when designers and builders have left the building and moved on to other projects, it is incumbent upon facility staff to optimize the system they inherit and to do so over a long lifetime.

Even systems of superior quality in design, component selection, and construction can become inefficient, unreliable, and have their lifetimes drastically shortened by poor O&M practices. In addition, and more importantly, poor O&M practices can lead to unsafe facilities that can endanger not only the district cooling workforce but also the public at large.

To achieve a successful O&M program, the management of the DCSs must be committed to success. This is a necessary condition, but not sufficient in itself. Programs must be organized such that execution becomes part of the everyday routine.

Trained and qualified personnel having the requisite skillset are of course the cornerstone of chilled-water (CHW) plant and distribution operations, but it is also of utmost importance for an enterprise to establish documented procedures and associated guidance materials to ensure, at a minimum, continuity with personnel changes, let alone basic due diligence in operation by the owner. Product manuals for primary equipment components can form the basis for a given DCSs O&M program, along with establishment of additional work practices tailored to the particular site and system. At minimum, equipment checklists that cover startup, shutdown, and normal and emergency operating procedures need to be developed. Being by nature a 24/7/365 enterprise that delivers vital utility service, DCS system knowledge needs to be documented and readily accessible, so that operations team members charged with responsibility to deliver service have the means to do so. Any given operations team will have a range of expertise depending upon their individual tenures, so it is important that hoarding knowledge is avoided, and senior personnel should be actively encouraged to share their experience with less-seasoned people. For instances in which mentoring can make an enormous positive difference, it would be hard to find a better example than DCS utility O&M.

For purposes of this chapter, operations as well as maintenance are treated as integrated functions. However, operations as used here can be simplified to describe those activities related to the continuing production and distribution of chilled water, e.g., process-related monitoring and supervisory tasks. Maintenance, on the other hand, as used here, refers to the periodic upkeep of the system, which can range from minor tasks performed on in-service equipment, to phased/programmed activities that require scheduled equipment outages, to reactive repairs done to address an equipment breakdown or other upset condition.

As with any field of endeavor, performance measurements are important, and for O&M, perhaps even more so. This is because of its continuing and, in many ways, repeatable nature. The desire for consistency makes it susceptible to establishing benchmarks or "bogeys" and the monitoring of improvement (or degradation) trends over time. There are many types of analytics and reports that operators can produce to aid them in their work, and many are tailored to site needs, but a few are fundamental to DCSs in general. These, of course, include plant efficiency, typically expressed as kW/ton refrigeration but also, as applicable, heat input per ton. More information on such metrics is provided later in this chapter and also in Chapter 3. Metrics can be further subdivided to measure performance of individual production units in order to assist operators in dispatching units. Other standard metrics include various measures of equipment availability, such as percentage of time delivery of CHW is on specification and forced downtime. Still more practices that can evaluate maintenance condition are in common use, such as vibration measurement for major pieces of equipment, use of corrosion coupons, and infrared thermography for example.

Space considerations will limit the level of detail into which this chapter can cover DCS O&M. Furthermore, the layout, equipment configuration, commodity cost structure, load characteristics, weather and geography, customer profile, and a host of other considerations make each and every DCS an enterprise that is unique to itself. However certain principles are fairly universal. Original equipment manufacturer (OEM) manuals are indispensable source documents for O&M and not only for the chillers, cooling towers, and pumps; the balance-of-plant consists of scores of ancillary assets that will have guidance literature available for them. One of the main functions of the operating staff is to assimilate this information, organizing it into their work practices, and administering its execution.

Furthermore, a number of codes and standards, some of which mandate specific practices and/or equipment considerations, provide guidance and standardization. The applicability of these codes and standards may vary from one jurisdiction to another. ASHRAE (2016) provides a comprehensive overview of codes and standards for HVAC systems with a number of these being applicable to DCSs. Other sources of guidance/standards/regulatory requirements are cited where appropriate.

## WORKPLACE SAFETY

In most jurisdictions, agencies exist that are responsible by law for ensuring workplace safety. Within the US, the most often-cited agency is the Federal Occupational Safety and Health Administration (OSHA), which is part of the US Department of

Labor. The Occupational Safety and Health (OSH) Act of 1970 created OSHA to help employers and employees reduce injuries, illnesses, and deaths on the job. Since then, workplace fatalities have been cut by more than 60% and occupational injury and illness rates have declined 40%, despite the fact that U.S. employment has more than doubled during that same period. However, significant hazards and unsafe conditions still exist in US workplaces. Each year, almost 5200 Americans die from workplace injuries in the private sector. Perhaps as many as 50,000 employees die from illnesses in which workplace exposures were a contributing factor. And nearly 4.3 million people suffer nonfatal workplace injuries and illnesses. The cost of occupational injuries and illnesses totals more than $156 billion per year.

Thus, district cooling providers like all businesses that operate in hazardous environments, must have workplace safety programs in place. The experience of one district heating and cooling provider in developing an effective workplace safety program is described by Toth and Merrill (2009).

Many of the elements of safe workplace practices for DCSs are shared with other utility systems. Familiar categories common to most or all are personal protective equipment (PPE); energy isolation (i.e. lockout/tagout), electrical safety, chemical safety. fall protection, and confined space. Each of these areas, and others as well, require the full attention and commitment of facility management and staff.

Certain safety-related aspects are of particular importance in DCS systems, and to some extent, unique to the industry. Refrigerant safety is one example (see for example ASHRAE Standards 15 [ASHRAE 2013a] and 34 [ASHRAE 2013b]). At present there are a wide variety of refrigerants utilized in DCS systems. These substances have a range of characteristics depending upon the specific chemistry of the refrigerant and specialized attention is required for each (see Chapter 3). Generally speaking, refrigerants of all types carry some inherent risk and as such need to be handled and utilized with care. Toxicity and flammability as well as asphyxiation potential are of concerns to varying degrees for refrigerant formulations currently in use. It is incumbent on the O&M staff to become intimately familiar with the characteristics of its particular refrigerant working fluid and to remain vigilant that all required safeguards and procedures are in place and complied with.

DCS systems are also typically situated in high-density areas, and as such, safety awareness must extend in many cases to interaction with customers as well as the general public. The distribution network is often routed through rights-of-way with heavy vehicle and/or pedestrian traffic. This means that maintenance of facilities involving excavations, manholes, and vaults brings the work into close contact to the public and an extra level of vigilance and protection is required to keep passersby and others safe in addition to personnel.

Furthermore, as cooling tower installations of significant size are typical of DCS production plants, O&M personnel must pay attention to the potential risks associated with these components, most notably as relates to *Legionella* bacteria, a disease pathogen that is found in many types of water systems, including evaporative cooling systems. DCS best practices incorporate a number of measures to monitor and control water quality and maintain equipment components to mitigate these risks.

*Requirements*. For example, in the case of systems in the US, OSHA standards require that employers maintain conditions or adopt practices that are reasonably necessary and appropriate to protect workers on the job, be familiar with and comply with standards applicable to their establishments, and ensure that employees have and use personal protective equipment when required for safety and health.

*Hazards*. Aside from the expected hazards on DCS construction/repair sites, many workplace hazards are found in the central plants of DCSs. Workplace hazards may also exist in distribution system valve vaults and even within the end user building where the consumer interface is located. OSHA, for example, issues standards for a wide variety of workplace hazards including toxic substances, harmful physical agents, noise, electrical hazards, fall hazards, trenching hazards, hazardous waste, infectious diseases, fire and explosion hazards, dangerous atmospheres, machine hazards, and confined spaces. In addition, where there are no specific OSHA standards, employers must comply with the OSH Act's "General Duty Clause," Section 5(a)(1), requires that each employer "furnish...a place of employment which [is] free from recognized hazards that are causing or are likely to cause death or serious physical harm to his employees." For more information on OSHA and their guidelines and standards refer to their comprehensive website at www.osha.gov.

## SECURITY

Beyond the normal security that is intended to protect property from loss due to theft or vandalism, DCSs need to consider instances where their facilities might put the public at risk and provide appropriate security. Examples might include the protection of fuel storage facilities, refrigerant storage, water treatment chemicals, or large CHW storage tanks. Increased physical security of fixed facilities has received much attention after the terrorist attacks of September 11, 2001 in the US. A number of sources of information are available for both the design of new facilities and the retrofit of existing facilities; for example, see www.wbdg.org/design/secure_safe.php (WBDG 2012). In addition to physical security, designers must also consider the potential impacts of natural disasters such as earthquakes, tsunamis, tornadoes, hurricanes, typhoons, and wildfires. As appropriate to the location of the district cooling utility, designs should consider these events and owners should be prepared with disaster plans.

## DISTRICT COOLING SYSTEM OPERATIONS AND MAINTENANCE

### Organization and Structure

DCS systems, although varying significantly in size and scale, production equipment and methods, as well as the commercial relationship with their end users, nonetheless have in common that they essentially perform the function of an energy utility. In fact, more often than not, they are structured in many ways as a vertically integrated utility, albeit a relatively localized one, in which the same entity produces, distributes transfers, and transacts for the chilled-water energy delivery to the customer.

There are of course a number of variations on this theme, including but not limited to commercial systems in which chilled water is marketed and sold to third parties; divisions of public investor-owned utilities with a chilled water customer base; municipalities operating a chilled water system for other municipal host buildings, (and in some cases to nongovernmental customers as well); and campus-type systems in which the chilled-water function is a department or unit of the institution, and service is delivered to the institutional owner's user buildings that comprise the campus.

From an O&M standpoint, however, these varying enterprises have much more in common than they do differences. Regardless of commercial structure, chilled water must be delivered reliably and efficiently on a continuous basis and within required service specifications. A worthy ideal for a DCS, even those of modest size, is to structure, organize, staff, and administer the system according to a best-in-class utility model, because, apart from differences in scale, many of the challenges and risks, as well as opportunities and objectives, are very similar.

The importance of establishing a high-performance O&M function is not always generally understood. Not surprisingly, the development, design, and construction of new capital equipment, hardware, and infrastructure gets almost all of the attention when a DCS comes together. However, once a system has been designed, constructed, commissioned, and turned over to an owner, it is up to the managers and supervisors, operators, maintenance mechanics, and technicians to provide stewardship for the system. This of course constitutes the lion's share of a DCS project's life-cycle. Looked at from that perspective, owners and developers of DCS systems would be wise to place a high priority on establishment and continuation of O&M best practices. The best intentions, and even execution, of first-rate design and construction can only be realized and perpetuated by a comparably qualified O&M organization staffed by professionals.

The following is a discussion of several primary considerations when establishing an O&M organization for DCS. These are necessarily very general and high-level, but they are meant to be guideposts for owners and operators to frame the fundamental elements for management and upkeep of the DCS property, equipment, and systems under their care. It is by no means an exhaustive list, and the specifics of a given DCS will likely add many other equally important considerations.

### Personnel

Nothing could be more obvious, and it borders on being a cliché, that the people operating, maintaining, and supervising the performance of a plant and systems should be highly experienced, appropriately qualified, and dedicated O&M staff. By and large this is the case in the DCS industry, and you can find countless examples of O&M professionals delivering reliable service to critical needs in places around the world, day-in, day-out, and often under challenging conditions.

It is important to note, however, that the scope and complexity of DCS O&M service delivery is not universally recognized by owners and managers that are reliant on the service. Occasionally in some settings, often in which DCS is a departmental function, i.e., a "back-of-house" service/cost center of a larger enterprise, it can be susceptible to getting short shrift in terms of attention and resources. The DCS's operation and maintenance can be relegated to that of a quasicustodial role, along with other facility-related services. This can be a recipe for trouble as a DCS is anything but "plug-and-play," and requires specific technical sophistication. It goes without saying that any similarity between a window air-conditioner and a DCS ends definitively with the basics of the thermodynamic refrigeration cycle. As this district cooling guide lays out in some level of detail, a DCS is a complex system that requires a staff with a strong grasp of technical principles augmented by the right mix of direct, relevant experience. While this experience can be cultivated from within an organization or recruited from outside of the organization (or for that matter outsourced to qualified third parties under a term contract), it should be viewed as nonnegotiable to entrust the very substantial, complex, and often mission-critical nature of a DCS to anything other than a well-qualified team.

Finding and recruiting talent can be challenging, as is the case for many highly skilled, vocational trades is in this day and age. This is not isolated to any one country or region, and in some areas, the availability of talent can be even more acute. It goes without saying that attracting talent with directly relevant experience in similar installations and with comparable equipment is a first choice. But there are naturally limits to those opportunities. Certain skill sets can often lend themselves well to adaptation to DCS plants. Power plant and central heating plant operators will often have complementary skills, knowledge (e.g. rotating machinery), work habits and familiarity with the employment demands of operating DCS system. On the maintenance side, specialized technical skills from other utility and/or industrial settings are often transferable to DCS needs. Millwrights, pipefitters, welders, electricians, instrumentation specialists, etc. are all trades that are prized when rounding out a DCS O&M staff. Once again, attracting such qualified personnel is competitive. In the US, some places to prospect for talent searches include retiring military, and in the past, the US Navy in particular for several of the base requisite skill sets. In recent years, as awareness has grown of technical skill shortages, and likewise, the fact that DCS and many other industries provide opportunity for young people to commence a rewarding and fulfilling career, some community colleges and vocational schools have geared up to train and develop their students to fit the need. These programs are worth seeking out. In summary, in terms of O&M, nothing is more important than the quality of the staff that runs the DCS, thus that is the first order of business.

### Procedure and Documentation

Going hand-in-hand with ensuring that the right personnel are in place is establishing consistent O&M protocols that can be readily accessed and communicated among personnel. Many DCSs are fortunate to have senior personnel with extensive site-specific experience, but a concentration of expertise (and consequentially, very heavy reliance) on one or a handful of individuals can become a liability in the inevitable event of employee turnover. This becomes a real concern, particularly if important know-how is not widely and routinely shared. For this reason, the time and effort the O&M organization spends to produce standard operating procedures (SOPs) is worthwhile. As noted, OEM documentation covering major equipment components certainly provides the foundation for developing these SOPs, but they must be tailored to describe the unique characteristics of the given system. Site-specific plant start-up, shutdown, equipment sequencing, part-load operation, as well as procedures to address system contingencies should be defined and reduced to written form. In conjunction with this, operator checklists help ensure that steps are followed in the correct sequence. Similarly, good DCS O&M practice requires a maintenance documentation approach that is aligned to the needs of the plant. Largely, in view of the scale of most DCS facilities, this presupposes a computerized maintenance management system (CMMS) that utilizes a software program to ensure required maintenance is scheduled and performed according to the appropriate intervals. Such a system also functions to provide a readily accessible history and database that grows in value to the O&M team as plant equipment progresses through its life-cycle. Less-sophisticated means of scheduling and tracking maintenance (i.e., a spreadsheet) may have the virtue of simplicity, but little beyond that to recommend it. Often such schedules and records are static and are not readily nor easily updated, and further subject to not being shared widely enough and/or kept in multiple versions of varying accuracy. A wide variety of CMMS systems have been a fixture in DCSs for a number of years and are readily available and economical for even the most modest of enterprises, so their use is highly recommended. When CMMS systems are deployed, management must commit to fully using them. A software system's effectiveness is a function of the completeness, validity and accuracy of the data that is put into it, and likewise, attention to execution and closeout of the work in the system as well. On occasion, maintenance routines for certain equipment will be set up outside the CMMS for one reason or another; however, absent a compelling justification for the practice, such "shadow" systems should be discouraged. The right CMMS system will generally be versatile and powerful enough to capably assist in maintenance management for all aspects of the most sophisticated of DCSs. Further, because it functions as the primary repository of plant history and condition status, it is recommended to consolidate all maintenance activity into one CMMS for the enterprise. Further discussion of the overarching approaches to maintenance will take place later on in this chapter.

## Training

As DCS facility managers understand all too well, there is no readily available pool of talent that can come in off the street and advance quickly to attain qualification to operate and maintain a DCS. From time-to-time an experienced hand will come available, but by and large, demand is high and competition fierce for credentialed staff. Add to this the fact that in the US and developed world generally, vocational training that concentrates on the necessary technical disciplines relevant to DCS often goes begging for people willing to take it on. That puts the onus on in-house staff to develop and follow through with a program to train O&M personnel, many of whom may enter the workforce with a minimal or elementary baseline of experience. Personnel training in a DCS will typically be a mix of informal, on-the-job instruction, along with classroom training (which itself can be developed internally) as well as on-premises and/or online commercial training to supplement the effort. Mentoring, or perhaps more in keeping with the skilled-trade nature of DCS, an apprenticeship-type approach can be the most time-tested and proven way to cultivate a strong DCS O&M team. Foremen/lead personnel, previously qualified and well-seasoned on the equipment in their care, can observe, instruct and evaluate the trainees under their supervision, and upon satisfactory demonstration of skill, sign off on advancement according to prescribed levels of understanding and competence. As with many aspects of DCS systems, no two examples of training progressions will be identical.

Each facility's unique staff composition, plant layout, technology, and circumstances, along with the base qualifications the new hire starts with, will dictate the rate of progress to qualification. A journeyman-level operator/technician attracted from another DCS plant may rapidly qualify to take over shift responsibility in three months or less. A green hire with the basic raw skills may take upwards of two years or more to gain the confidence and competence to reach a passable level of competence—even when closely guided and instructed by experienced staff.

Another aspect with regard to O&M qualification is that in recent decades the prevalence of cross-trained personnel has become fairly widespread. Save for some situations, for example, collective bargaining labor agreements that specify certain work rules, and/or a plant/systems particular configuration that can justify a specialized skill/certification, DCS personnel will quite often wear the hat of both operator and technician/mechanic. This trend was naturally accelerated by advances in automation and monitoring/supervisory/control systems that centralized these functions to a central platform. This is by no means to suggest that regular operator rounds aren't essential, but it has to a great extent consolidated what used to be the role of several operators spread out among several operating platforms to a single central operator will often also monitor the balance of the system including delivery of service to the users. On the ground, the practical impact of this change means that apart from a single-shift operator, most plant personnel are assigned to equipment upkeep during business hours, though available and subject to augmenting operational duties as needed. In summary, DCS O&M personnel need to acquire significant breadth as well as depth of skills and qualifications. DCS training programs therefore need to be designed to take this into consideration.

## Performance and Metrics

As a utility service that provides a vital, primary service to its customer base, reliability of service is paramount when gauging the performance of a district cooling system. It is commonly expressed as a percentage describing the hours that system capacity was available versus total hours in the reporting period. In DCS, system reliability is generally expected to be in the high 90s, if not well in excess of 99%. It may be reasonable to record somewhat lower availability figures for individual chillers or other primary equipment, but this is provided that the DCS has the requisite redundant capacity to ensure the highest value for overall system delivery. Other measures, some of which are adapted from use in electric utility industry, can further refine measurement of reliability, e.g., equivalent forced outage rate, defined as the fraction of time in which the system/plant is unavailable due to unplanned or forced equipment downtime. Reliability is perhaps the most visible and crucial statistic of a DCS, but there are many other metrics that are used to gauge O&M performance which are important tools as well. A general non-DCS specific sampling of frequently tracked metrics is illustrated in Table 8.1, but most DCS facilities will track a number of others as well. IDEA (2008) also provides a compilation of metrics for DCS operators.

DCS managers may find certain of these measures useful and others less so, and/or the targets may differ for a specific site. For example, maintenance labor productivity might be substituted for Metric #2 in Table 8.1, schedule compliance, where maintenance time allowances are not well known. In addition, as a critical utility DCS will likely aim for more stringent metrics than those suggested here. An example would be a DCS with a highly proactive maintenance program targeting a 97%+ rate of preventive maintenance completion (Metric #5 in Table 8.1). Similarly, with respect to overtime labor (Metric #4), it may be warranted to operate with a higher proportion of overtime depending upon the staffing situation for a given facility.

When considering metrics that involve measuring system parameters, IDEA (2008) provides some fundamental advice:

- "Essence of measurement: If you cannot measure it, you cannot manage it."
- "Accuracy is key: If you cannot measure it accurately, you had better not measure it."
- "Meaningful reporting: Data must be assembled and reported in structured automated reports."

Given that a DCSs lifetime will span decades, operational benchmarks/targets coupled with continual monitoring/measurement of current performance, and comparison against those target objectives, is a must. Apart from reliability, energy efficiency is the most significant driver that will define successful DCS operations.

Familiar to all, energy efficiency is typically expressed (depending upon input motive energy and geographic location) in DCS systems as kW/ton or $kW_e/kW_t$. It can be viewed as the consummate measure of O&M effectiveness, as nearly every plant occurrence, practice, and intervention will likely have a direct influence on it to greater or lesser degree. Water efficiency in most if not all cases is of similar importance, and nowhere is

**Table 8.1** Industry O&M Metrics and Benchmarks (NASA 2000)

| | Metric | Variables and Equation | Benchmark |
|---|---|---|---|
| 1 | Equipment Availability | Hours each unit is available to run at capacity/Total hours during reporting time period | > 95% |
| 2 | Schedule Compliance | Total hours worked on scheduled jobs/Total hours scheduled | > 90% |
| 3 | Emergency Maintenance | Total hours worked on emergency jobs/Total hours worked | < 10% |
| 4 | Maintenance Overtime | Total maintenance overtime during period/Total regular maintenance hours during period | < 5% |
| 5 | Preventive Maintenance Completion Percentage | Preventive maintenance actions completed/Preventive maintenance actions scheduled | > 90% |
| 6 | Preventive Maintenance Budget/Cost | Preventive maintenance cost/Total maintenance cost | 15%–18% |
| 7 | Predictive Maintenance Budget/Cost | Predictive maintenance cost/Total maintenance cost | 10%–12% |

this more relevant than in the arid regions of the world in which DCS is experiencing its most rapid growth. The metric used in this case is typically gallons/ton, or some variant; see Chapter 3 for more information on specific performance metrics for the DCS plant.

Many other metrics also serve as guideposts in an O&M organization. O&M vigilance and a proactive approach to running and caring for systems in a first-rate manner is a proven means to mitigate the inevitable degradation in a given system's performance that will otherwise occur as a function of time in service. Some examples include the following:

- Program/preventive versus corrective (reactive maintenance)
- Preventive maintenance compliance
- Maintenance work order backlog/aging
- Maintenance cost per CHW ton-h
- Staff retention/turnover
- Absenteeism
- Customer interruption rates
- Customer complaints
- Inventory/spares
- Transmission and distribution losses
- Corrosion rates for steel and copper components

Although discussed last here only in terms of metrics, adherence to safety and environmental requirements must be the foremost goal and an underlying premise. Some standard measurements, such as lost-time accident rates, compilation of employee training progress, and recordkeeping of environmental compliance performance should be considered as minimum measures. In many jurisdictions, minimums are likely to be required by law where employee and/or public safety or environmental protection is at issue. DCS owners and operators are encouraged to institute programs that equal or exceed the industry standard in this regard

## Administration and Support

As noted previously, DCSs are typically structured as vertically integrated utilities that serve the entire value chain from production through end-user delivery/transaction. This basic structure remains valid for the most part, even for quasi- or noncommercial, institutional facilities in which DCS is a departmental function. As such, DCSs require support for business functions outside of the realm of but nonetheless supporting plant operation and maintenance. Procurement of fuel, energy, materials, spares, and supplies; storeroom operations; engineering support; vendor administration; IT and technical support; regulatory compliance and governmental affairs interaction; metering and billing; finance, budgeting and accounting; marketing and business development; customer relations (these latter two items as applicable); and other peripheral functions are all, to varying degrees, characteristics that are part and parcel of the ongoing operation of a DCS enterprise. The technical personnel that specialize in DCS O&M and actually run the plant are often hard-pressed (and, quite understandably, not necessarily possessing the skill sets) to also deal with the many business-related functions attendant to running the enterprise. This can be a particular concern in smaller operations in which personnel often "wear many hats." While many O&M teams can manage diverse duties quite capably, structuring the organization such that sufficient administrative support is in some way available to the O&M team can help these specialists focus on what they do best.

## O&M Outsourcing

As previously alluded, an owner's responsibility for staffing, managing, and administering a DCS can be daunting, and especially when that owner's primary expertise and resources are in a field of endeavor far removed from DCS. Some owners may therefore consider contracting some or part of their delivery of service to a third-party firm. Before going any further, very many institutions, such as some universities, health care facilities, corporate campuses, and other enterprises manage and run DCS facilities entirely in-house, all the while setting a high industry standard. However, the fact remains that delivery of service at the scale of DCS is a complex activity that some owners may, for a number of reasons, opt to outsource O&M to a specialist, just as they may do so with other cost centers.

This option can be approached on a more limited basis, such as the common practice of contracting out the provision of water treatment services and/or taking advantage of long-term service agreements for major equipment, wherein periodic maintenance is provided on a turn-key basis by the OEM or a similarly qualified provider. Even for enterprises that pride themselves on providing service more or less entirely in-house, the opportunities to parcel out certain specialized areas to third parties is often a compelling option from the standpoint of economics and simple practicality.

Owners that opt to contract out some or all of their O&M requirements do so for a variety of reasons. For example, the DCS endeavor may be far removed from their core competencies and/or present obstacles to staffing, recruiting, and retaining personnel with the required skills. Other owners may seek to leverage DCS assets by negotiating an energy savings performance contract (ESPC) or similar contracting mechanism as a means to finance upkeep and maintenance of the DCS as well as, ideally, yield operational savings.

The decision to contract O&M out is a significant one that requires an owner's diligent research and deliberation. An arrangement that turns over O&M responsibility and can span a long period of time (one to five years typically or much longer terms, some even encompassing third-party plant equity stakes or asset sales/transfers as part of an agreement) is one not to be taken lightly. If considering outsourcing, owners should consider a number of common-sense principles. A few are listed as follows:

- Because O&M contracts do not typically require the outside vendor to bring equity, a broad range of potential providers exists, from relatively small energy service contractors/providers, to product manufacturers, to utility subsidiaries, among other entities.
- Given the above, owners' can enjoy somewhat of a "buyer's market," as providers will compete fiercely for O&M business. That said, track record, reputation testimonials, and general past performance, not to mention appropriate scale and resources of potential providers, must be thoroughly vetted.
- Contract details are very important. Setting baselines from which to evaluate the quality of vendor performance is key. Performance incentives and penalties, as well as "shared savings" provisions can be included. Owners are well within their rights to put the onus of risk on the contractor for plant reliability, availability, safety, environmental performance, and other important aspects. Similarly, care should be taken to ensure that uncontrollable factors such as commodity price risk are equitably shared between the parties. Provisions such as termination for convenience should be considered by owners to protect their interests in a relationship that can sour.

These points are but a few basic aspects to consider when outsourcing, and much more must be evaluated for such a significant alternative. Still, turning to and partnering with the right provider to take care of O&M can be a viable option for owners under the right circumstances. However, DCS owners should take advantage of their desirability as a potential client and work diligently to select the right pool of providers and negotiate the most cost-effective deal for their customer base.

## DCS CENTRAL PLANT OPERATIONS AND MAINTENANCE

District cooling production is detailed to a considerable degree throughout this guide, from the basics of the refrigeration cycle, to the chilling technologies used, to plant and pumping configurations. Furthermore, the primary and most pertinent written source for O&M guidance is that provided by the original equipment manufacturers of the primary equipment used, augmented to an equal or greater degree by the institutional knowledge and experience of O&M staff. In light of that, this discussion will be limited to general observations regarding DCS central plant O&M, and is by no means a comprehensive guide.

### Chilled-Water Production

Given that a DCS by definition serves an assembled collection of cooling loads, typically larger structures in a relatively dense urban or campus area, plant capacities are relatively large, at least by a layman's standard of the provision of air conditioning. But even that classification covers a great deal of ground. A small college facility could be composed of a chiller(s) serving a few hundred tons of connected load, and at the other end of the spectrum, a DCS can also serve a dense urban metropolis totaling perhaps 100,000 tons or more of cooling load. Nonetheless, in several important respects, fundamental O&M principles will be remarkably similar for each, even though these two extremes couldn't be more different concerning scale.

Generally speaking, the primary production equipment found throughout the range of existing DCS systems is mechanical vapor compression chillers. More specifically, these are most commonly water-cooled centrifugal machines. The condensing water for heat rejection is typically provided by coupling the system with cooling towers using evaporative heat rejection.

Once again, the foregoing is very much a generalization. Other production machine variations have and continue to be used in certain cases. Examples are the use of absorption refrigeration chillers where a viable waste heat or other heating source exists, either solely or in conjunction with mechanical compression chilling. Deep lake cooling or other natural chilling energy sources are utilized for DCS where the opportunity is present. Surface water diversion in lieu of condensing water by means of evaporative cooling, also where local circumstances permit, is sometimes used in specialized instances. In arid, landlocked regions of the world, large scale air cooling/radiator installations can in certain applications be a viable substitute for the heavy water use required by evaporative heat rejection.

#### Discussion of Chilled-Water Production and Distribution Operations.

**Chiller Operations.** As noted, DCS production is typically anchored by mechanical vapor compression chillers, and these are predominantly centrifugal machines, but with screw-type and reciprocating positive-displacement machines also well represented. Despite the different means to compress the refrigerant gas, each type employs an evaporator section in which heat is absorbed into the refrigerant from the chilled water circuit; a condenser in which heat is rejected to circulating condenser water; and the thermal expansion valve which expands the refrigerant into the evaporator section.

Absorption chillers, sometimes referred to as absorbers, are used with some frequency in DCSs, particularly in association with combined heat and power applications or other settings in which there is a readily available heat source (including otherwise wasted heat) that can be produced or recovered to produce chilling. Absorbers differ substantially in concept and design from vapor compression chillers in that it uses a thermal rather than mechanical process to produce chilling. Absorbers are ultimately capable of producing chilled water at similar specifications, but their operating principles are quite different. Chapter 3 of this guide provides further information on their theory and operating principles. From an O&M perspective, there are a few broad generalizations that can be made concerning absorbers in a DCS setting:

- When configured in hybrid plants that also include mechanical chillers, operators should seek to base load absorbers when possible. This is often desirable in any case, as absorbers are frequently the most economical production unit when using low-grade and/or waste heat. Further, they are often less amenable to responding to load swings than mechanical equipment.
- Very generally speaking, the relative scarcity of these machines means that the O&M knowledge base is not extensive. Readily available manufacturer support to operators is therefore highly desirable when absorption chilling is present. In its absence, O&M staff having responsibility to operate and maintain absorption chillers should receive unit-specific training and qualification from an OEM representative or other authorized person. CHW plant operator skill sets tend to be geared towards experience with rotating machinery and ancillary equipment. Supervision of an absorber requires an understanding and competence with the very different pressure, temperature, and flow relationships in a thermal process.
- Certain equipment safeguards must also be maintained when using absorbers. Low condensing water temperature excursions can lead to crystallization, or “rocking,” on tube and vessel surfaces, and this can be difficult and costly to reverse. Similarly, conditions that lead to air-in leakage can lead to a reaction with the Li-Br salts and accelerate corrosion, which in turn exacerbates the problem and degrades performance.

Caveats aside, absorption chillers can be integrated very well into DCS plant operation in many circumstances. Besides the optionality they provide to O&M teams, their utilization can be the difference maker in the make-or-break economics of a DCS project. In DCS operations, chilled water is typically the working fluid, distributed in most cases through a closed-loop network in a more or less unadulterated form, but in some installations as a dilute solution (brine)—often in the latter case to facilitate incorporation of thermal energy storage (TES).

The operator’s responsibility can be stated in fairly simple terms: to reliably and efficiently deliver the chilled-water product to the end-users. However, a glib understatement such as this belies the many variables and circumstances inherent to DCS O&M, and more so, the skills and experience of the operators, technicians, and mechanics that actually make it happen day-in and day-out.

Documentation produced in the course of actual operating experience on the ground, supported by OEM handbooks, design and construction records, commissioning data, and engineering reports, etc. serves as primary source material for any given plant. Nonetheless, a number of basic requirements regarding the performance of DCS O&M in the form of some high-level generalizations can be made for purposes of this guide, even though each DCS facility will be unique.

DCS plant upkeep and maintenance can be divided into activities that are wholly integrated into the ongoing functioning of the process, i.e., maintenance that is of a nature that the activity is more or less seamless with the production cycle of the equipment. It is performed either when the equipment is offline (but available), or if practical, on operating machinery. In a sense, even the monitoring and inspection that make up the standing duties of the operators is in fact itself the baseline maintenance of the system.

On a per-shift or daily basis, typical operator activities will include the monitoring of process temperature/pressure/flow and other parameters to ensure they are within specified set points. Hotwells, sumps, lube oil reservoirs, sight glasses, motor bearings, etc., are to be monitored regularly for anomalous conditions. Depending on a given plant's level of automation, a varying proportion of these values may be monitored, recorded, and/or trended with a plant supervisory system stationed at a central operator's platform. This instrumentation greatly enhances an operator's ability to manage DCS production, but there is still no substitute for equipment inspection rounds performed by technicians on a regular basis to confirm the remote data based on actual observation, inspect equipment visually, and to intervene directly in the event of unusual or upset conditions. Certain metrics, such as tons refrigeration produced; chilled-water delivery; electricity, fuel, and/or steam consumed; condensing makeup water consumption; and other variables are commonly recorded by operators, typically on at least a per-shift basis if not more frequently. Also, of unique importance in routine DCS operations is the monitoring of CHW delivery conditions among the assembled customers. Each customer's particular demand and usage characteristics; and the configuration of their terminal equipment (as well as its condition) in addition to geographic location on the network all create an operating dynamic that operators must adapt to in real time. Inherent in this is the ability of operators to anticipate changes in cooling requirements to the extent it is possible, and dispatching and sequencing the plant to react and respond as needed.

Other routine operational activities are performed frequently as well, but possibly on a somewhat less-frequent interval than a per-shift basis. Water treatment, whether serviced in-house or by an outside specialty contractor or service provider, must nonetheless be monitored regularly by operators. Systems using open cooling towers in particular need to be tested for proper dosing and concentrations of chemical agents to mitigate scaling and corrosion as well as biological growth. Regular rounds are also made of the towers and related auxiliaries themselves, inspecting for proper lubrication; vibration in pump and fan drives; excessive accumulation of debris, slimes, or other contaminants in decks, sumps, and basins; proper operation of float-operated devices/electronic sensing units; damage or fouling in tower fill; excessive fogging; and general condition of the system. Blowdown is also typically managed by operators to maintain the proper treatment concentration, with makeup water tuned to compensate for that removal. In short, operators monitor the tower/condensing water components in a manner similar to the rounding that takes place for the chillers and balance-of-plant production equipment.

While the preceding describes a very high-level overview of the "O" in O&M, DCS best practices also presupposes a rigorous and consistent periodic maintenance program for all plant equipment. As will be discussed later in this chapter, there are competing philosophies in terms of things like the triggering of a maintenance interval, and each has its merits. However, when it comes to actual execution of specified steps, there is little difference among them as to the "what" and the "how." In the end, maintenance is nonnegotiable. The cost of maintaining DCS assets will be borne one way or the other; it is much more economical, reliable and efficient to acknowledge, plan for, and carry out these necessary maintenance events than to neglect them and react to inevitable upsets and untimely failures.

The life cycle of DCS plant capital equipment as well as continuing utility-grade, reliable service can be directly traced to the establishment of a periodic maintenance program that is executed consistently and according to established intervals.

Broadly, preventive maintenance is the antithesis of breakdown (reactive) maintenance. As discussed briefly later in this chapter, there are several approaches that each have their merits, but all are designed to minimize equipment downtime, preserve the useful life of CHW equipment, and avoid equipment and system failure. Intervals will, of course, vary depending upon materiality of the work needing to be done. Several of the previously discussed operations-related tasks in essence qualify as preventive in nature. The other extreme might be a condenser retubing or prime mover replacement following 5-years/40,000 operating hours or, more likely, a much longer time in service. However, more familiar and frequent in nature is the significant maintenance activity performed under the direct supervision of DCS O&M staff on weekly, monthly, quarterly, semiannual, and annual frequencies. Refrigerant leak checks of operating equipment are customary of fairly frequent intervals. Condenser surfaces are one area in particular that require regular monitoring and maintenance. Scale buildup on water sides will impede heat transfer, and annual cleaning of tubes during an outage period will control this. As condenser tube bundles age, eddy current testing of tubes is often recommended in order to assess wall loss. In those fairly infrequent instances in which open-loop surface water cooling is utilized, gross fouling as well as abrasive siltation can greatly affect the condition and service life of condensers. In these cases, the advantages of elimination of cooling towers and extensive makeup water use is somewhat tempered by the need to maintain a large strainer and/or centrifugal separator. The condenser also may require special metallurgy and yet still be subject to a shorter service life.

According to much longer intervals, major maintenance that begins to enter the realm of capital replacement takes place. Compressors and their prime movers will undergo overhauls and/or be replaced in kind. Entire tube bundles may need replacing, and the list can go on. A sample maintenance schedule from DOE (2010) for a large chiller is shown in Table 8.2.

Intervals should be viewed as guidance only. Original equipment manufacturers may specify more or less frequent maintenance, and additional requirements are likely to be appropriate for differing equipment and facilities. In addition, frequencies may also be prescribed according to a run-hour approach rather than a calendar, particularly in situations where some units see proportionally heavier or lighter duty relative to one another, or if the plant's operating mode involves cycling of equipment more so than steady-state operation.

### Thermal Energy Storage (TES) Operations

The demonstrated advantages associated with TES, particularly in locales in which there are time-of-day energy cost disparities that an owner has an opportunity to take advantage of, have resulted in significant adoption of the technology in DCS systems in a number of diverse geographic locations. As is ever the case with such relatively sophisticated and capital-intensive design enhancements to DCS delivery, it is essential to create a workable O&M strategy that is followed up by consistent, day-in, day-out execution, properly integrating TES into the chilling production scheme. Effective deployment of TES is highly dependent on devising and adhering to operational sequencing that itself can be fluid and variable, as it is contingent on a number of factors. For example, changes in weather forecasts, load-side behavior, energy commodity prices, and equipment status can all influence performance of TES. The optionality that TES affords also carries with it the responsibility for operators to go beyond matching chilling production with cus-

**Table 8.2** Maintenance Frequencies for a Chiller as Recommended by DOE (2010)

| Description | Comments | Maintenance Frequency | | | |
|---|---|---|---|---|---|
| | | Daily | Weekly | Semiannually | Annually |
| Chiller use/sequencing | Turn off/sequence unnecessary chillers | X | | | |
| Overall visual inspection | Complete overall visual inspection to be sure all equipment is operating and safety systems are in place | X | | | |
| Check setpoints | Check all setpoints for proper setting and function | X | | | |
| Evaporator and condenser | Assess evaporator and condenser coil fouling as required | | X | | |
| Compressor motor temperature | Check temperature per manufacturer's specifications | | X | | |
| Perform water quality test | Check water quality for proper chemical balance | | X | | |
| Leak testing | Conduct leak testing on all compressor fittings, oil pump joints and fittings, and relief valves | | X | | |
| Check all insulation | Check insulation for condition and appropriateness | | X | | |
| Control operation | Verify proper control function including:<br>• Hot gas bypass<br>• Liquid injection | | X | | |
| Check vane control settings | Checking settings per manufacturer's specification | | | X | |
| Verify motor load limit control | Checking settings per manufacturer's specification | | | X | |
| Verify load balance operation | Checking settings per manufacturer's specification | | | X | |
| Check chilled-water reset settings and function | Checking settings per manufacturer's specification | | | X | |
| Check chiller lockout setpoint | Checking settings per manufacturer's specification | | | | X |
| Clean condenser tubes | Clean tubes at least annually as part of shutdown procedure | | | | X |
| Eddy current test condenser tubes | As required, conduct eddy current test to access tube wall thickness | | | | X |
| Clean evaporator tubes | Clean tubes at least annually as part of shutdown procedure | | | | X |
| Eddy current test evaporator tubes | As required, conduct eddy current test to assess tub wall thickness | | | | X |
| Compressor motor and assembly | • Check all alignments to specification<br>• Check all seals, provide lubrication where necessary | | | | X |
| Compressor oil system | • Conduct analysis on oil and filter<br>• Change as required<br>• Check oil pump and seals<br>• Check oil heater and thermostat<br>• Check all strainers, valves, etc. | | | | X |
| Electrical connections | Check all electrical connections/terminals for contact and tightness | | | | X |
| Water flows | Assess proper water flow in evaporator and condenser | | | | X |
| Check refrigerant level and condition | Add refrigerant as required. Record amounts and address leakage issues. | | | | X |

tomer demand more or less concurrently. The dynamics involved with dispatching the TES relative to that of the output of the mechanical chillers—sometimes also complicated by weather, load disruptions, and/or electricity prices that can change in real time—adds another dimension. All of this necessarily raises the bar for operators, requiring adherence to a disciplined approach to optimize production along with storage in order to achieve design intent and realize investment returns.

Chapter 6 of this guide details the design elements and service applications of TES relative to DCS facilities. Once again, specific circumstances will dictate operating strategy. In a facility in which the intent is to primarily offset electric demand during peak periods, mechanical chilling will often be cycled to produce chilling in tank charging mode during the off-shifts. Then, depending on the discharge rate dictated by load at a given point in time and for a given storage capacity, the production machines are typically cycled to minimal output, if not cycled off altogether. This puts the onus on operators to effectively manage the cycle, as the consequences of demand spikes at inopportune times can be costly with the potential for even one shortfall potentially having far-reaching impacts, e.g., in the case of a purchased electric tariff that applies a demand ratchet. In such demand-limiting scenarios then, the production buffer and storage resource that TES provides can nonetheless be unforgiving if it is not skillfully managed and deployed by plant staff. At no time is such operator competence and foresight more relevant than on the "design day," on which the intricate combination and timing of energy production, storage, and discharge is critical.

In a load-leveling scenario, TES functions to essentially boost CHW production with less mechanical hardware. Chilled-water storage accumulated during the off-peak period is paralleled with mechanical production during the peak. In an ideal scenario, mechanical chillers run more or less steady-state near their rated capacity.

From the operator's standpoint, throughout the cooling season, attention is routinely focused on configuration and dispatch of CHW pumping to meet the charge/discharge cycle more so than the sequencing of production machines. The mixing and matching of storage/simultaneous mechanical production must of course be managed by operators in tune with seasonal load variations as well.

When integrated into a DCS, TES is a powerful advantage that serves to enhance facility reliability and flexibility to a great degree. With that, TES provides operators with the optionality and means to provide service more economically. By the same token, it adds another level of complexity requiring a more sophisticated approach in CHW operations in order to attain those benefits that it is designed to yield.

### Refrigerant Management

In the US, environmental regulations have been instituted to require owners and operators to follow formal management guidelines to limit release of ozone-depleting gases. These requirements prescribe certification, work practices, and related reporting and recordkeeping standards aimed at mitigation of losses to atmosphere of CFC, HCFC, and, more recently, HFC refrigerants. Given the breadth of the covered substances under the most recent revision, it follows that most DCS operations in the US are subject to its provisions.

Beyond jurisdictional mandates relating to environmental effects of refrigerants, effective management of refrigerants is important. This is true primarily for personnel safety and also simply because of its criticality to the DCS production process, as well as commodity availability and cost concerns. Establishment and documentation of procedures for storage, handling, transferring, and disposing of refrigerants are important. So too are the means and equipment to monitor the production process and provide early detection of leaks. For the these reasons and more, refrigerant management policies and procedures, whether mandated by code, should nonetheless be part of every DCSs O&M strategy.

## WATER TREATMENT

A proper water treatment program is paramount to ensure DCS component life, operating efficiency, and control of potential public health threats. Chemical treatment is used to control corrosion, scaling, biofouling, and biothreats. Mechanical filtration is used to remove suspended particles.

The following discussion in this section of water treatment issues for DCSs is extracted largely from Chapter 49, "Water Treatment," of the 2007 edition of *ASHRAE Handbook—HVAC Applications*, and the reader is referred there and the updated version (ASHRAE 2015) for additional details.

### Corrosion

Corrosion is the destruction of a metal or alloy by chemical or electrochemical reaction with its environment. In most instances, this reaction is electrochemical in nature, much like that in an electric battery. For corrosion to occur, a corrosion cell consisting of an anode, a cathode, an electrolyte, and an electrical connection must exist. Metal ions dissolve into the electrolyte (water) at the anode. Electrically charged particles (electrons) are left behind. These electrons flow through the metal to other points (cathodes) where electron-consuming reactions occur. The result of this activity is the loss of metal and often the formation of a deposit. Various types of corrosion exist: general, localized or pitting, galvanic, etc. The reader is referred to *ASHRAE Handbook—HVAC Applications* for additional details on the types of corrosion and the factors that contribute to corrosion (ASHRAE 2015). ASHRAE (2015) also contains discussion of the various factors that contribute to corrosion. For corrosion issues specific to dissimilar metals in piping systems, the reader is referred to Sperko (2009).

### Corrosion Protection and Preventive Measures

*Materials Selection.* Any piece of equipment can be made of metals that are virtually corrosion-proof under normal and typical operating conditions. However, economics usually dictates material choices. When selecting construction materials, the following factors should be considered:

- Corrosion resistance of the metal in the operating environment
- Corrosion products that may be formed and their effects on equipment operation
- Ease of construction using a particular material
- Design and fabrication limitations on corrosion potential
- Economics of construction, operation, and maintenance during the projected life of the equipment, i.e., expenses may be minimized in the long run by paying more for a corrosion-resistant material and avoiding regular maintenance
- Use of dissimilar metals should be avoided. Where dissimilar materials are used, insulating gaskets and/or organic coatings must be used to prevent galvanic corrosion
- Compatibility of chemical additives with materials in the system

*Protective Coatings*. The operating environment has a significant role in the selection of protective coatings. Even with a coating suited for that environment, the protective material depends on the adhesion of the coating to the base material, which itself depends on the surface preparation and application technique.

*Maintenance of Protective Coating.* Defects in a coating are difficult to prevent. These defects can be either flaws introduced into the coating during application or mechanical damage sustained after application. In order to maintain corrosion protection, defects must be repaired.

*Cycles of Concentration.* Some corrosion control may be achieved by optimizing the cycles of concentration (the degree to which soluble mineral solids in the makeup water have increased in the circulating water due to evaporation). Generally, adjustment of the blowdown rate and pH to produce a slightly scale-forming condition (see section on Scale Control) will result in an optimum condition between excess corrosion and excess scale.

*Chemical Methods.* Protective film-forming chemical inhibitors reduce or stop corrosion by interfering with the corrosion mechanism. Inhibitors usually affect either the anode or the cathode.

Anodic corrosion inhibitors establish a protective film on the anode. Though these inhibitors can be effective, they can be dangerous if insufficient anodic inhibitor is present, the entire corrosion potential occurs at the unprotected anode sites. This causes severe localized (or pitting) attack.

Cathodic corrosion inhibitors form a protective film on the cathode. These inhibitors reduce the corrosion rate in direct proportion to the reduction of the cathodic area.

Table 8.3 lists typical corrosion inhibitors of these three types. The most important factor in an effective corrosion inhibition program is the consistent control of both the corrosion inhibition chemicals and the key water characteristics. No program will work without controlling these factors.

*Cathodic Protection.* Sacrificial anodes reduce galvanic attack by providing a metal (usually zinc, but sometimes magnesium) that is higher on the galvanic series (see ASHRAE 2015) than either of the two metals that are coupled together. The sacrificial anode thereby becomes anodic to both metals and supplies electrons to these cathodic surfaces. Proper design and placement of these anodes are important and usually require the services of a National Association of Corrosion Engineers (NACE) registered engineer. When properly used, they can reduce loss of steel from the tube sheet of exchangers using copper tubes. Sacrificial anodes have helped supplement chemical programs in many cooling water and process water systems.

Impressed-current protection is a similar corrosion control technique that reverses the corrosion cell's normal current flow by impressing a stronger current of opposite polarity. Direct current is applied to an anode-inert material (platinum, graphite) or expendable material (aluminum, cast iron) reversing the galvanic flow and converting the steel from a corroding anode to a protective cathode. This method is very effective in protecting essential equipment such as elevated water-storage tanks, steel tanks, or softeners.

For buried CHW distribution systems, additional detail on cathodic protection is provided in Chapter 4.

**Table 8.3** Typical Corrosion Inhibitors

| **Corrosion Inhibitor Type** | | |
|---|---|---|
| Anodic | Mainly Cathodic | General |
| Molybdate | Bicarbonate | Soluble oils |
| Nitrite | Polyphosphate | Other organics, such as azole or carboxylate |
| Orthophosphate | Phosphonate | |
| Silicate | Zinc | |
| Polysilicate | | |

## White Rust on Galvanized Steel Cooling Towers

White rust is a zinc corrosion product that forms on galvanized surfaces. It appears as a white, waxy, or fluffy deposit composed of loosely adhering zinc carbonate. The loose crystal structure allows continued access of the corrosive water to exposed zinc. Unusually rapid corrosion of galvanized steel, as evidenced by white rust, can affect galvanized steel cooling towers under certain conditions. Before chromates in cooling tower water were banned, the common treatment system consisted of chromates for corrosion control and sulfuric acid for scale control. This control method generally has been replaced by alkaline treatment involving scale inhibitors at a higher pH. Alkaline water chemistry is naturally less corrosive to steel and copper, but creates an environment where white rust on galvanized steel can occur. Also, some scale prevention programs soften the water to reduce hardness, rather than use acid to reduce alkalinity. The resulting soft water is corrosive to galvanized steel.

*Prevention.* White rust can be prevented by promoting the formation of a nonporous surface layer of basic zinc carbonate. This barrier layer is formed during a process called passivation and normally protects the galvanized steel for many years. Passivation is best accomplished by controlling pH during the initial operation of the cooling tower. Control of the cooling water pH in the range of seven to eight for 45 to 60 days usually allows passivation of galvanized surface to occur. In addition to pH control, operation with moderate hardness levels of 100 to 300 ppm as $CaCO_3$ and alkalinity levels of 100 to 300 ppm as $CaCO_3$ will promote passivation. Where pH control is not possible, certain phosphate-based inhibitors may help protect galvanized steel. A water treatment specialist should be consulted for specific formulations.

## SCALE CONTROL

Scale is a dense coating of predominantly inorganic material formed from the precipitation of water-soluble constituents. Some common scales are:

- Calcium carbonate
- Calcium phosphate
- Magnesium salts
- Silica

The following principal factors determine whether or not a water is scale forming:

- Temperature
- Alkalinity or acidity (pH)
- Amount of scale-forming material present
- Influence of other dissolved materials, which may or may not be scale-forming

As any of these factors change, scaling tendencies also change. Most salts become more soluble as temperature increases. However, some salts such as calcium carbonate become less soluble as temperature increases, and therefore they often cause deposits at higher temperatures.

A change in pH or alkalinity can greatly affect scale formation. For example, as pH or alkalinity increases, calcium carbonate, the most common scale constituent in cooling systems, decreases in solubility and deposits on surfaces. Some materials, such as silica ($SiO_2$), are less soluble at lower alkalinities. When the amount of scale-forming material dissolved in water exceeds its saturation point, scale may result. In addition, other dissolved solids may influence scale-forming tendencies. In general, a higher level of scale-forming dissolved solids results in a greater chance for scale formation. Indices such as the Langelier Saturation Index (Langelier 1936) and the Ryznar Stability Index (Ryznar 1944) can be useful tools to predict the calcium carbonate scaling tendency of water. These indices are calculated using the pH, alkalinity, calcium hardness, temperature, and total dissolved solids of the water, and indicate whether the water will favor precipitating or dissolving of calcium carbonate.

Methods used to control scale formation include:

- Limit the concentration of scale-forming minerals by controlling cycles of concentration or by removing the minerals before they enter the system (see the section on external treatments later in this chapter). Cycles of concentration are the ratio of makeup rate to the sum of blowdown and drift rates. The cycles of concentration can be monitored by calculating the ratio of chloride ion, which is highly soluble, in the system water to that in the makeup water.
- Make mechanical changes in the system to reduce the chances for scale formation. Increased water flow and heat exchangers with larger surface areas are examples.
- Feed acid to keep the common scale-forming minerals (e.g., calcium carbonate) dissolved.
- Treat with chemicals designed to prevent scale. Chemical scale inhibitors work by the following mechanisms:
    1. Threshold inhibition chemicals prevent scale formation by keeping the scale-forming minerals in solution and not allowing a deposit to form. Threshold inhibitors include organic phosphates, polyphosphates, and polymeric compounds.
    2. Scale conditioners modify the crystal structure of scale, creating a bulky, transportable sludge instead of a hard deposit. Scale conditioners include lignins, tannins, and polymeric compounds.

## Nonchemical Methods

Equipment based on magnetic, electromagnetic, or electrostatic technology has been used for scale control in boiler water, cooling water, and other process applications.

Magnetic systems are designed to cause scale-forming minerals to precipitate in a low-temperature area away from heat exchanger surfaces, thus producing nonadherent particles (e.g., aragonite form of calcium carbonate versus the hard, adherent calcite form). The precipitated particles can then be removed by blowdown, mechanical means, or physical flushing.

The objective of electrostatics is to prevent scale-forming reactions by imposing a surface charge on dissolved ions that causes them to repel.

Results of side-by-side comparative tests with conventional water treatment have been mixed. A Federal Technology Alert report regarding these technologies stresses that success of the application depends largely on the experience of the installer (DOE 1998). The report includes a discussion of the potential benefits achieved and the necessary precautions to consider when applying these systems. ASHRAE Research Project RP-1155 studied physical water treatment (PWT) (Cho 2002). For this study, a PWT device was defined as a nonchemical method of water treatment for scale prevention or mitigation. Bulk precipitation was proposed as the mechanism of scale prevention. Three different devices described as permanent magnets, a solenoid coil device, and a high-voltage electrode were tested under laboratory conditions. Fouling-resistance data obtained in a heat transfer test section supported the benefit of all three devices when configured in optimum conditions.

## External Treatments

Minerals may also be removed by various external pretreatment methods such as reverse osmosis and ion exchange. Zeolite softening, demineralization, and dealkalization are examples of ion exchange processes.

# BIOLOGICAL GROWTH CONTROL

Biological growth (algae, bacteria, and fungi) can interfere with a cooling operation due to fouling or corrosion and may present a health hazard if present in aerosols produced by the equipment. Thermal energy storage systems that are open may also introduce additional opportunity for biological growth and contamination into a DCS. Heating equipment operates above normal biological limits and therefore has fewer microbial problems. When considering biological growth in a cooling system, it is important to distinguish between free-living planktonic organisms and sessile (attached) organisms. Sessile organisms cause the majority of the problems, though they may have entered and multiplied as planktonic organisms.

Biological fouling can be caused by a wide variety of organisms that produce biofilm and slime masses. Slimes can be formed by bacteria, algae, yeasts, or molds and frequently consist of a mixture of these organisms combined with organic and inorganic debris. Organisms such as barnacles and mussels may cause fouling when river, estuarine, or sea water is used. Biological fouling can significantly reduce the efficiency of cooling by reducing heat transfer, increasing back pressure on recirculation pumps, disrupting flow patterns over cooling media, plugging heat exchangers, and blocking distribution systems. In extreme cases, the additional mass of slime has caused the cooling media to collapse.

Microorganisms can dramatically enhance, accelerate, or in some cases, initiate localized corrosion (pitting). Microorganisms can influence localized corrosion directly by their metabolism or indirectly by the deposits they form. Indirect influence may not be mediated by simply killing the microorganisms; deposit removal is usually necessary, while direct influence can be substantially mediated by inhibiting microorganism metabolism.

Algae use energy from the sun to convert bicarbonate or carbon dioxide into biomass. Masses of algae can block piping, distribution holes, and nozzles. A distribution deck cover, which drastically reduces the sunlight reaching the algae, is one of the most cost-effective control devices for a cooling tower. Biocides are also used to assist in the control of algae.

Algae can also provide nutrients for other microorganisms in the cooling system, increasing the biomass in the water. Bacteria can grow in systems even when nutrient levels are relatively low. Yeasts and fungi are much slower growing than bacteria and find it difficult to compete in bulk waters for the available food. Fungi do thrive in partially wetted and high humidity areas such as the cooling media. Wood-destroying species of fungi can be a major concern for wooden cooling towers as the fungi consume the cellulose and/or lignin in the wood, reducing its structural integrity.

Most waters contain organisms capable of producing biological slime, but optimal conditions for growth are poorly understood. Equipment near nutrient sources or equipment that has process leaks acting as a food source is particularly susceptible to slime formation. Even a thin layer of biofilm significantly reduces heat transfer rates in heat exchangers.

## Control Measures

Eliminating sunlight from wetted surfaces such as distribution troughs, cooling media, and sumps significantly reduces algae growth. Eliminating dead-legs and low-flow areas in the piping and the cooling loop reduces biological growth in those areas. Careful selection of materials of construction can remove nutrient sources and environmental niches for growth. Maintaining a high-quality makeup-water supply with low bacteria counts also helps minimize biological growth. Equipment should also be designed with adequate access for inspection, sampling, and manual cleaning.

Sometimes the effective control of slime and algae requires a combination of mechanical and chemical treatments. For example, when a system already contains a considerable accumulation of slime, a preliminary mechanical cleaning makes the subsequent application of a biocidal chemical more effective in killing the growth and more effective in preventing further growth. A buildup of scale deposits, corrosion product, and sediment in a cooling system also reduces the effectiveness of chemical biocides. Routine manual cleaning of cooling towers, including the use of high-level chlorination and a biodispersant (surfactant), helps control *Legionella* bacteria as well as other microorganisms.

*Microbiocides.* Chemical biocides used to control biological growth in cooling systems fall into two broad categories: oxidizing and nonoxidizing biocides.

*Oxidizing biocides.* Oxidizing biocides (chlorine, chlorine-yielding compounds, bromine, bromochlorodimethylhydantoin (BCDMH, or BCD), ozone, iodine, and chlorine dioxide) are among the most effective microbiocidal chemicals. However, they are not always appropriate for control in cooling systems with a high organic loading. In wooden cooling towers, excessive concentrations of oxidizing biocides can cause delignification and overdosing of oxidizing biocides may cause corrosion of metallic components. In systems large enough to justify the cost of equipment to control feeding of oxidizing biocides accurately, the application may be safe and economical. The most effective use of oxidizing biocides is to maintain a constant low-level residual in the system. However, if halogen-based oxidizing biocides are fed intermittently (slug dosed), a pH near seven is advantageous because at this neutral pH, halogens are present as the hypohalous acid (HOR, where R represents the halogen) form over the hypohalous ion ($OR^-$) form. The effectiveness of this shock feeding is enhanced due to the faster killing action of hypohalous acid over that of hypohalous ion. The residual biocide concentration should be tested, using a field test kit, on a routine basis. Most halogenation programs can benefit from the use of dispersants or surfactants (chlorine helpers) to break up microbiological masses.

Chlorine has been the oxidizing biocide of choice for many years, either as chlorine gas or in the liquid form as sodium hypochlorite. Other forms of chlorine, such as powders or pellets, are also available. The use of chlorine gas is declining due to the health and safety concerns involved in handling this material, and in part due to environmental pressures concerning the formation of chloramines and trihalomethane.

Bromine is produced either by the reaction of sodium hypochlorite with sodium bromide on site, or by release from pellets. Bromine has certain advantages over chlorine: it is less volatile, and bromamines break down more rapidly than chloramines in the environment. Also, when slug feeding biocide in high pH systems, hypobromous acid may have an advantage because its dissociation constant is lower than that of chlorine. This effect is less important when biocides are fed continuously.

Ozone has several advantages compared to chlorine: it does not produce chloramines or trihalomethane, it breaks down to nontoxic compounds rapidly in the environment, it controls biofilm better, and it requires significantly less chemical handling. The use of ozone-generating equipment in an enclosed space, however, requires care be taken to protect operators from the toxic gas. Also, research by ASHRAE has shown that ozone is only marginally effective as a scale and corrosion inhibitor (Gan et al. 1996; Nasrazadani and Chao 1996).

Water conditions should be reviewed to determine the need for scale and corrosion inhibitors and then, as with all oxidizing biocides, inhibitor chemicals should be carefully selected to ensure compatibility. To maximize the biocidal performance of the ozone, the injection equipment should be designed to provide adequate contact of the ozone with the circulating water. In larger systems, care should be taken to ensure that the ozone is not depleted before the water has circulated through the entire system.

Iodine is provided in pelletized form, often from a rechargeable cartridge. Iodine is a relatively expensive chemical for use on cooling towers and is probably only suitable for use on smaller systems.

*Nonoxidizing Biocides.* When selecting a nonoxidizing microbiocide, the pH of the circulating water and the chemical compatibility with the corrosion and/or scale inhibitor product must be considered. The following list, while not exhaustive, identifies some of these products:

- Quaternary ammonium compounds
- Methylene bis(thiocyanate) (MBT)
- Isothiazolones
- Thiadiazine thione
- Dithiocarbamates
- Decyl thioethanamine (DTEA)
- Glutaraldehyde
- Dodecylguanidine
- Benzotriazole
- Tetrakis(hydroxymethyl)phosphonium sulfate (THPS)
- Dibromo-nitrilopropionamide (DBNPA)
- Bromo-nitropropane-diol
- Bromo-nitrostyrene (BNS)
- Proprietary blends

The manner in which nonoxidizing biocides are fed is important. Sometimes the continuous feeding of low dosages is neither effective nor economical. Slug feeding large concentrations to achieve a toxic level of the chemical in the water for a sufficient time to kill the organisms present can show better results. Water blowdown rate and biocide hydrolysis (chemical degradation) rate affect the required dosage. The hydrolysis rate of the biocide is affected by the type of biocide, along with the temperature and pH of the system water. Dosage rates are proportional to system volume; dosage concentrations should be sufficient to ensure that the contact time of the biocide is long enough to obtain a high kill rate of microorganisms before the minimum inhibitory concentration of the biocide is reached. The period between nonoxidizing biocide additions should be based on the system half-life, with sequential additions timed to prevent regrowth of bacteria in the water.

*Handling Microbiocides.* All microbiocides must be handled with care to ensure personal safety. In the US, cooling water microbiocides are approved and regulated through the EPA (Environmental Protection Agency), and by law must be handled in accordance with labeled instructions. Maintenance staff handling the biocides should read the material safety data sheets and be provided with all the appropriate safety equipment to handle the substance. Automatic feed systems that minimize and eliminate the handling of biocides should be used by maintenance personnel.

*Other Biocides.* Ultraviolet irradiation deactivates the microorganisms as the water passes through a quartz tube. The intensity of the light and thorough contact with the water are critical in obtaining a satisfactory kill of microorganisms. Suspended solids in the water or deposits on the quartz tube significantly reduce the effectiveness of this treatment method. Therefore, a filter is often installed upstream of the lamp to minimize these problems. Because the ultraviolet light leaves no residual material in the water, sessile organisms and organisms that do not pass the light source are not affected by the ultraviolet treatment. Ultraviolet irradiation may be effective on humidifiers and air washers where the application of biocidal chemicals is unacceptable and where 100% of the recirculating water passes the lamp. Ultraviolet irradiation is less effective where all the microorganisms cannot be exposed to the treatment, such as in cooling towers. Ultraviolet lamps require replacement after approximately every 8000 h of operation.

*Metallic ions*, namely copper and silver, effectively control microbial populations under very specific circumstances. Either singularly or in combination, copper and silver ions are released into the water via electrochemical means to generate 1 to 2 ppm of copper and/or 0.5 to 1.0 ppm of silver. The ions assist in the control of bacterial populations in the presence of a free chlorine residual of at least 0.2 ppm. Copper in particular effectively controls algae.

Liu et al. (1994) reported control of *Legionella* pneumophila bacteria in a hospital hot-water supply using copper-silver ionization. In this case, *Legionella* colonization decreased significantly when copper and silver concentrations exceeded 0.4 and 0.04 ppm, respectively. Also, residual disinfection prevented *Legionella* colonization for two months after the copper-silver unit was inactivated.

Significant limitations exist in the use of copper and silver ion for cooling systems. Many states are restricting the discharge of these ions to surface waters, and if the pH of the system water rises above 7.8, the efficacy of the treatment is significantly reduced. Systems that have steel or aluminum heat exchangers should not be treated by this method, as the potential for the deposition of the copper ion and subsequent galvanic corrosion is significant.

### Legionnaires’ Disease

Like other living things, *Legionella pneumophila*, the bacterium that causes Legionnaires' disease (legionellosis), requires moisture for survival. *Legionella* bacteria are widely distributed in natural water systems and are present in many drinking water supplies. The *Legionella* bacteria is a very small bar shaped cell measuring 0.000079 in. by 0.000012 in. (2 by 0.3 micrometers). Potable hot-water systems between 80°F and 120°F, cooling towers, certain types of humidifiers, evaporative condensers, whirlpools and spas, and the various components of air conditioners are considered to be amplifiers. These bacteria are killed in a matter of minutes when exposed to temperatures above 140°F.

Due to its small cell size, legionellosis can be acquired by inhalation of *Legionella* organisms in aerosols. Aerosols can be produced by cooling towers, evaporative condensers, decorative fountains, showers, and misters. It has been reported that the aerosol from cooling towers can be transmitted over a distance of up to 2 mi (3.2 km). If air inlet ducts of nearby air conditioners draw the aerosol from contaminated cooling towers into the building, the air distribution system itself can transmit the disease. When an outbreak of Legionnaires’ disease occurs, cooling towers are often the suspected source. However, other water systems may produce an aerosol and should not be neglected. Amplification of *Legionella* within protozoans has been demonstrated, and *Legionella* bacteria are thought to be protected from biocides while growing intracellularly. Amplification of *Legionella* bacteria in biofilm and slime masses has been shown by a number of researchers. Microbial control programs should consider the effectiveness of the products against slimes as part of the *Legionella* control program.

*Prevention and Control.* The *Legionella* count required to cause illness has not been firmly established because many factors are involved, including the virulence and number of *Legionella* in the air, the rate at which the aerosol dries, the wind direction, and the susceptibility to the disease of the person breathing the air. The organism is often found in sites not associated with an outbreak of the disease. It has been shown that it is feasible to operate cooling systems with *Legionella* bacteria below the limit of detection and that the only method to prove that a system is operating at these levels is to specifically test for *Legionella* bacteria, rather than to infer from total bacteria count measurements.

Periodic monitoring of circulating water for total bacteria count and *Legionella* count can be accomplished using culture methods. Monitoring system cleanliness and using a microbial control agent that has proven efficacy, or is generally regarded as effective in controlling *Legionella* populations, are also important. Other measures to decrease risk include optimizing cooling tower design to minimize drift, eliminating dead legs or low flow areas, selecting materials that do not promote the growth of *Legionella*, and locating the tower so that drift is not injected into the air handlers. ANSI/ASHRAE Standard 188-2018 (2018) was developed to establish minimum risk management requirements for building water systems including condenser water and is written in code-enforceable language. This standard requires that a water management program be established to monitor and control water systems to reduce the risk of legionellosis due to building water system operation.

## SUSPENDED SOLIDS AND DEPOSITION CONTROL

In water-cooled DCSs, the cooling tower acts as a great air filter. Any airborne debris that is drawn into the tower will make its way to the basin or sump and accumulate. Since most particles in condenser water from cooling towers are smaller than 0.0004 in. (10 micrometers) and they are organic, they are great food sources for microorganisms and bacteria and could also lead to scaling of heat transfer surfaces. Hence condenser-water filtration is extremely important and effective filtration down to 0.00001 in. (0.25 micrometers) will assist in filtering out *Legionella* cells. There are several types of filtration equipment that are available for use and they are described below.

### Mechanical Filtration

Straineres, filters, and separators may be used to reduce suspended solids to an acceptable low level. Generally, if the screen is 200 mesh, equivalent to about 0.003 in. (8 micrometers), it is called a strainer; if it is finer than 200 mesh, it is called a filter.

*Strainers*. A strainer is a closed vessel with a cleanable screen designed to remove and retain foreign particles down to 0.001 in. (25 micrometers) diameter from various flowing fluids. Strainers extract material that is not wanted in the fluid, and allow saving the extracted product if it is valuable. Strainers are available as single-basket or duplex units, manual, or automatic cleaning units, and may be made of cast iron, bronze, stainless steel, copper-nickel alloys, or plastic. Magnetic inserts are available where microscopic iron or steel particles are present in the fluid.

*Cartridge Filters.* These are typically used as final filters to remove nearly all suspended particles from about 0.004 in. (100 micrometers) down to 0.00004 in. (1.0 micrometers) or less. Cartridge filters are typically disposable (i.e., once plugged, they must be replaced). The frequency of replacement, and thus the economic feasibility of their use, depends on the concentration of suspended solids in the fluid, the size of the smallest particles to be removed, and the removal efficiency of the cartridge filter selected.

In general, cartridge filters are favored in systems where contamination levels are less than 0.01% by mass (<100 ppm), and are available in many different materials of construction and configurations. Filter media materials include yarns, felts, papers, nonwoven materials, resin-bonded fabric, woven wire cloths, sintered metal, and ceramic structures. The standard configuration is a cylinder with an overall length of approximately l0 in. (250 mm), an outside diameter of approximately 2.5 to 2.75 in. (65 to 70 mm), and an inside diameter of about 1 to 1.5 in. (25 to 40 mm), where the filtered fluid collects in the perforated internal core. Overall lengths from 4 to 40 in. (100 to 1000 mm) are readily available.

Cartridges made of yarns, resin-bonded, or melt-blown fibers normally have a structure that increases in density towards the center. These depth-type filters capture particles throughout the total media thickness. Thin media, such as pleated paper (membrane types), have a narrow pore-size distribution design to capture particles at or near the surface of the filter. Surface-type filters can normally handle higher flow rates and provide higher removal efficiency than equivalent depth filters. Cartridge filters are rated according to manufacturers' guidelines. Surface-type filters have an absolute rating, while depth-type filters have a nominal rating that reflects their general classification function. Higher efficiency melt-blown depth filters are available with absolute ratings as needed.

*Sand Filters.* A downflow filter is used to remove suspended solids from a water stream. The degree of suspended-solids removal depends on the combinations and grades of the medium being used in the vessel. During the filtration mode, water enters the top of the filter vessel. After passing through a flow impingement plate, it enters the quiescent (calm) freeboard area above the medium.

In multimedia down flow vessels, various grain sizes and types of media are used to filter the water. This design increases the suspended-solids holding capacity of the system, which in turn increases the backwashing interval. Multimedia vessels might also be used for low suspended-solids applications, where chemical additives are required. In the multimedia vessel, the fluid enters the top layer of anthracite media, which has an effective size of 0.04 in. (1.0 mm) This relatively coarse layer removes the larger suspended particles, a substantial portion of the smaller particles, and small quantities of free oil. Flow continues down through the next layer of fine garnet material, which has an effective size of 0.012 in. (0.30 mm). A more finely divided range of suspended solids is removed in this polishing layer. The fluid continues into the final layer, a coarse garnet material that has an effective size of 0.08 in. (2.0 mm). Contained in this layer is the header/lateral assembly that collects the filtered water.

When the vessel has retained enough suspended solids to develop a substantial pressure drop, the unit must be backwashed either manually or automatically by reversing the direction of flow. This operation removes the accumulated solids out through the top of the vessel.

There are also hybrid filters available that combine the action of centrifugal separation and sand filtration in the same vessel that offers superior filtration down to 0.45 mm, but they come at a cost premium.

Sand filters may also be used on a side-stream basis with the injection of a coagulating chemical upstream of the filter that creates larger particles and aids in them being filtered out by the sand filter.

*Centrifugal-Gravity Separators.* In this type of separator, liquids/solids enter the unit tangentially, which sets up a circular flow. Liquids/solids are drawn through tangential slots and accelerated into the separation chamber. Centrifugal action tosses the particles heavier than the liquid to the perimeter of the separation chamber. Solids gently drop along the perimeter and into the separator's quiescent collection chamber. Solids-free liquid is drawn into the separator's vortex (low-pressure area) and up through the separator's outlet. Solids are either purged periodically or continuously bled from the separator by either a manual or automatic valve system. Centrifugal separators are typically less able to produce particulate-free fluid than filters and strainers. In addition, their effectiveness is compromised by large variations in water flow as separation effectiveness is driven by velocity.

*Bag-Type Filters*. These filters are composed of a bag of mesh or felt (through which the filtered media must pass) supported by a removable perforated metal basket, placed in a closed housing with an inlet and outlet. The housing is a welded, tubular pressure vessel with a hinged cover on top for access to the bag and basket. Housings are made of carbon steel or stainless steel. The inlet can be in the cover, in the side (above the bag), or in the bottom (and internally piped to the bag). The side inlet is the simplest type. In any case, the liquid enters the top of the bag. The outlet is located at the bottom of the side (below the bag). Pipe connections can be threaded or flanged. Single-basket housings can handle up to 220 gpm (15 L/s), multi-baskets up to 3500 gpm (220 L/s).

The support basket is usually of 304 stainless steel perforated with 1/8 in. (3 mm) holes. (Heavy wire mesh baskets also exist.) The baskets can be lined with fine wire mesh and used by themselves as strainers without adding a filter bag. Some manufacturers offer a second, inner basket (and bag) that fits inside the primary basket. This provides for two-stage filtering: first a coarse filtering stage, then a finer one. The benefits are longer service time and possible elimination of a second housing to accomplish the same function.

The filter bags are made of many materials (cotton, nylon, polypropylene, and polyester) with a range of ratings from 0.00004 to 0.033 in. (1.0 to 838 m). Most common are felted materials because of their depth-filtering quality, which provides high dirt-loading capability, and their fine pores. Mesh bags are generally coarser, but are reusable, and therefore less costly. The bags have a metal ring sewn into their opening; this holds the bag open and seats it on top of the basket rim.

In operation, the liquid enters the bag from above, flows out through the basket, and exits the housing cleaned of particulate down to the desired size. The contaminant is trapped inside the bag, making it easy to remove without spilling any downstream.

*Special Methods*. Localized areas frequently can be protected by special methods. Thus, pump-packing glands or mechanical shaft seals can be protected by fresh water makeup or by circulating water from the pump casing through a cyclone separator or filter, then into the lubricating chamber.

In smaller equipment, a good dirt-control measure is to install backflush connections and shutoff valves on all condensers and heat exchangers so that accumulated settled dirt can be removed by back-flushing with makeup water or detergent solutions. These connections can also be used for acid cleaning to remove calcium carbonate scale.

In specifying filtration systems, third-party testing by a qualified university or private test agency should be requested. The test report documentation should include a description of methods, piping diagrams, performance data, and certification.

In summary, Table 8.4 provides a summary of particle size ranges for various filter technologies.

**Table 8.4** Summary Table of Filter Technology Particle Size Range

| Filter Technology (listed in order of effectiveness) | Typical Range of Smallest Particle Filtration Level, μ |
|---|---|
| Cartridge filter | 0.01 to +100 |
| Sand Filter (including hybrid design) | 0.25 to 40 |
| Bag Filter | 1.0 to +100 |
| Centrifugal Separators | 5 to 75 |
| Disc Filter | 15 to 25 |
| Automatic Self-Cleaning Strainer | 15 to 50 |

**Example 8.1: Sidestream Filter**

This example is provided to illustrate the process of filter selection and the filter attributes and costs are for the purposes of illustration and not intended to be taken as accurate either in an absolute or relative sense. It is desired to filter an existing cooling tower adequate to mitigate the growth of *Legionella* in the basin and sump via a side-stream filter arrangement. The existing cooling tower had four cells with a flow rate in each cell of 6000 gpm (380 L/s) for a total tower flow rate of 24,000 gpm (1520 L/s). Multiple vendors with different filtration technologies were contacted using an RFQ process to determine the material cost of their filter. It was left up to the filter vendors to determine the adequate flow rate to be most effective at removing the particulate size for their equipment. A life-cycle cost analysis was used to determine the most economical selection over a ten-year life; therefore, the quantity and cost of back-flushing, media life and replacement costs, booster pump energy, etc., was included in the analysis using appropriate utility costs. As shown in the following table, the hybrid sand filter had the lowest life cycle cost and filtered the smallest particle.

| Filter Technology (Listed in Order of Effectiveness) | Vendor-Stated Smallest Particle Filtration Level | Sidestream Flow, mpg (L/s) | Backwash Flow, gal/day (L/day) | Filter First Cost | Net Present Value[1] |
|---|---|---|---|---|---|
| Sand Filter-Hybrid Sand | 0.25 | 1120 (70) | 8960 (33,900) | $205,000 | $65,000 |
| Sand Filter | 3 | 1200 (75) | 3820 (14,500) | $245,000 | $130,000 |
| Centrifugal Separators | 40 | 2500 (160) | 200 (750) | $92,000 | $200,000 |
| Disc Filter | 15 | 2400 (150) | 1920 (7270) | $190,000 | $350,000 |

*1. Economic assumptions for this example were a 10 year period at discount rate of 10% and escalation of utilities by 2.5% and salaries by 3% assuming $67/hr*

## SELECTION OF WATER TREATMENT

As discussed in the previous sections, many methods are available to prevent or correct corrosion, scaling, and biofouling. The selection of the proper water treatment method, and the chemicals and equipment necessary to apply that method, depends on many factors. The chemical characteristics of the water, which change with the operation of the equipment, are important. Other factors contributing to the selection of proper water treatment are:

- Economics
- Chemistry control mechanisms
- Dynamics of the operating system
- Design of major components (e.g., the cooling tower or boiler)

- Number of operators available
- Training and qualifications of personnel
- Preventive maintenance program

A general water treatment guidance for the types of systems found in DCSs follows.

## Once-Through Systems (Seawater or Surface Water Cooling)

Economics is an overriding concern in treating water for once through systems (in which a very large volume of water passes through the system only once). Protection can be obtained with relatively little treatment per unit mass of water because the water does not change significantly in composition while passing through equipment. However, the quantity of water to be treated is usually so large that any treatment other than simple filtration or the addition of a few parts per million of a polyphosphate, silicate, or other inexpensive chemicals may not be practical or affordable. Intermittent treatment with polyelectrolytes can help maintain clean conditions when the cooling water is sediment-laden. In such systems, it is generally less expensive to invest more in corrosion-resistant construction materials than to attempt to treat the water. Disposal of the water in a once-through system may also be complicated by the addition of treatment chemicals.

## Open Recirculating Systems (Cooling Towers)

In an open recirculating system with chemical treatment, more chemical must be present because the water composition changes significantly by evaporation. Corrosive and scaling constituents are concentrated. However, treatment chemicals also concentrate by evaporation; therefore, after the initial dosage, only moderate dosages maintain the higher level of treatment needed. The selection of a water treatment program for an open recirculating system depends on the following major factors:

- Economics
- Water quality
- Performance criteria (e.g., corrosion rate, bacteria count, etc.)
- System metallurgy
- Available staffing
- Automation capabilities
- Environmental requirements
- Water treatment supplier (some technologies are superior to others in terms of economics, ease of use, safety, and impact on the environment)

An open recirculating system is typically treated with a scale inhibitor, corrosion inhibitor, oxidizing biocide, nonoxidizing biocide, and possibly a dispersant. The exact treatment program depends on the previously mentioned conditions.

A water treatment control scheme for a cooling tower might include:

- Chemistry and cycles of concentration control using a conductivity controller
- Alkalinity control using automatic injection of sulfuric acid based on pH
- Scale control using contacting water meters, proportional feed, or traced control technology
- Oxidizing biocide control using an ORP (oxidation-reduction potential) controller
- Nonoxidizing biocide control using timers and pump systems

For cooling tower condenser-water systems that will be shut down and then restarted, ASHRAE (2015) provides recommended procedures.

## Closed Recirculating Systems (Distribution System)

In a closed recirculating system, water composition remains fairly constant with very little loss of either water or treatment chemical. Closed systems are often defined as those requiring less than 5% makeup water per year. The need for water treatment in such systems is often ignored based on the rationalization that the total amount of scale from the water initially filling the system would be insufficient to interfere significantly with heat transfer, and that corrosion would not be serious. However, leakage losses are common, especially in some types of CHW distribution systems (as discussed in Chapter 4), and corrosion products can accumulate sufficiently to foul heat transfer surfaces or deteriorate the interior of the piping system. Therefore, all systems should be adequately treated to control corrosion. Systems with high makeup rates should be treated to control scale as well.

The selection of a treatment program for closed systems should consider the following factors:

- Economics
- System metallurgy
- Operating conditions
- Makeup rate
- System size

Possible treatment technologies include:

- Buffered nitrite
- Molybdate
- Silicates
- Polyphosphates
- Oxygen scavengers
- Organic blends

Before new systems are treated, they must be cleaned and flushed. Grease, oil, construction dust, dirt, and mill scale are always present in varying degrees and must be removed from the metallic surfaces to ensure adequate heat transfer and to reduce the opportunity for localized corrosion.

Detergent cleaners with organic dispersants are available for proper cleaning and preparation of new closed systems.

## European Practice in Closed Distribution Systems

For low-temperature hot-water district heating, Europeans have established practices for water treatment in the distribution systems. These distribution systems are of high integrity and have very low rates of leakage/makeup. The European practice, in particular that of the Nordic countries, relies less on corrosion inhibitors than North American practices. Makeup water is filtered, demineralized or softened, and deaerated. Sodium hydroxide is then added to raise the pH to 9.5 to 10. Subsequently, the corrosion rate and concentrations are monitored. The systems are closely monitored, normally by the operating staff, rather than by the supplier of the corrosion inhibitor chemicals as is the normal practice in North America. Bellamy and Brandon (1996) compare the North American approach, which relies more heavily on corrosion inhibitors, to the European approach in a case study that concludes that while both provide adequate corrosion protection, the European approach will likely be less costly.

## Water Treatment in Steam Systems

For district cooling plants that use steam-driven absorption chillers, steam-turbine-driven chillers, or other combinations of technologies that involve steam, the reader is referred to *ASHRAE Handbook—HVAC Applications* for water treatment guidance for steam systems (ASHRAE 2015).

# MAINTENANCE PROGRAMS FOR DISTRICT COOLING SYSTEMS

There are several avenues DCS owners and operators can take in establishing, administering, and executing maintenance programs for their plant and system. Each have their respective merits for particular situations, and in some instances, several complementary approaches may be combined. Before considering approaches to maintenance, however, an often underappreciated prerequisite is overall management of the assets themselves. Simply put, DCS staff should have a comprehensive inventory of the equipment under their control, which can help ensure that all system elements are maintained. Minor malfunctions or failures of seemingly less consequential components can potentially cascade into much larger and more destructive process failures. This effect compounds itself as volume and complexity of components increases. Application of a robust CMMS system and, if warranted, an asset identification and tracking system using equipment barcoding can be an important aid to DCS upkeep, regardless of the overall maintenance approach chosen.

There is no one-size-fits-all approach to maintenance, and they will vary widely from one DCS to another. Resources—financial, technical, and personnel among others—will probably have the greatest influence upon the mode pursued. In many operators' situations, it will perhaps not be so much a choice from a menu of options as a direction taken out of necessity given a set of circumstances. Still, there are many factors to consider when deploying a maintenance program. Aside from the type of equipment used (e.g., compression versus absorption chillers, insulated piping versus uninsulated, etc.) other factors such as the operating environment and the age of the plant will have major impacts on the required maintenance. Certain climatic or other conditions may stress certain system components more than is customary or anticipated. Equipment may be operated for extended periods at peak or even overload conditions, or conversely, at unusually low load conditions. These effects as well as other operational contingencies or upsets may contribute to significant duty by machinery that is outside the recommended/specified operating envelope for the equipment. The presence of these effects and their frequency/prevalence must be considered and evaluated when implementing the program. For major pieces of equipment such as chillers, pumps, and towers, primary principles will be available from the equipment manufacturer, but that guidance must of course be leavened by knowledge of site-specific effects such as those mentioned. Moving down the scale in terms of asset value and prominence in the overall system, upkeep of these components also must be addressed. This would include balance-of-plant equipment, including items such as anything from heat exchangers, electrical gear and panels, refrigerant handling and recovery systems, valves and fittings, vessels, and tanks to the entire range of building support systems that could include control air systems, flood control, HVAC, emergency/standby power, and other systems. OEM manuals and documentation are still generally a primary resource in this realm, but sometimes other means must supplement them, including establishment of in-house guidance. These components are typically less capital intensive (on an individual basis, not necessarily in the aggregate) but must also be adequately maintained and are often quite numerous. It should be reiterated that ensuring the care of these assets is where the value of a CMMS and asset inventory/management system will truly present itself. Beyond this, and for other field-erected portions of the central plant, including the distribution system and the consumer interconnect, it is usually necessary to develop a maintenance program based on the methods of construction, the components used, and the age of the system. Certain elementary functions and activities are so integral to the operation of a chilled-water plant that an argument can be made that they are themselves operational in nature, more so than maintenance duties. Even a DCS that subscribes (or defaults) to a predominantly reactive program should see to it that in the course of everyday operation, certain fundamental duties are undertaken. Examples (as applicable) of these simple, but important tasks would include water quality testing and adjustment, cleaning/replacement of air, water, fuel and lubricating oil filters, exercising/lubricating/packing maintenance of valves. These and other basic duties contribute in an important way to the maintenance effort of a plant, and if neglected, can create equipment problems that will snowball into much larger and more problematic maintenance headaches. The benefits of incorporating ongoing light maintenance activities into the daily rounds and monitoring activities of plant operators should be self-evident.

There are three basic maintenance strategies: run-to-failure, preventative maintenance, and condition-based maintenance. *ASHRAE Handbook—HVAC Applications* defines these strategies as follows (ASHRAE 2015):

- Run-to-failure is a strategy applied when the cost of maintenance or repair may exceed the cost of replacement or losses in the event of failure. Only minimum maintenance such as cleaning or filter change is performed. The equipment may or may not be monitored for proper operation, depending on the consequences of failure. For example, a window air conditioner may be run although it is vibrating and making noise, then replaced rather than repaired.
- Preventive maintenance classifies resources allotted to ensure proper operation of a system or equipment under the maintenance program. Durability, reliability, efficiency, and safety are the principal objectives.
- Condition-based maintenance uses manual and automated inspection and monitoring to establish the current condition of equipment. It also uses condition and performance indices to optimize repair intervals.

It should be noted that with preventative maintenance programs, frequencies may be prescribed on a calendar time period (i.e., monthly) or based according to a run-hour approach, particularly in situations where some units see proportionally heavier or lighter duty relative to one another, or if the plant's operating mode involves cycling of equipment more so than steady-state operation.

Circumstances on the ground will help drive, if not dictate, the selection of a maintenance philosophy. At the extreme, if a facility's management (hypothetically) chose to have no coordinated program for regular maintenance, this in reality is itself a selection of a variant of a run-to-failure program. Maintenance in this case will therefore be very close to purely reactive/breakdown in nature.

Run-to-failure is not generally pursued, at least as a stated objective for a DCS. However, in reality it is sometimes encountered and implemented to some extent, again due to circumstances. While it has few defenders as an overall strategy, it is not entirely without merit in certain situations. In a plant in which adequate redundancy exists and a chiller or cooling tower, for example, is reaching the end of its useful life (or has long exceeded it) and is slated for wholesale replacement within a certain time-frame may be a candidate, as periodic maintenance in the machine's terminal stage may not be economic. Another example might simply involve resource constraints. DCS assets could be assessed for relative risk and the impact of individual failure on the system. Higher-risk items could be triaged for maintenance attention at recommended intervals, while those deemed less critical could be to some extent disregarded until failed (i.e., the window A/C example above). Another example of where run-to-failure might be a reasonable option is where access to the equipment in questions is very difficult and costs associated with just gaining access overshadow the component costs and any operational costs associated with its replacement. Thus, though it rarely has value beyond short-term expediency, run-to-fail is not without utility in select situations.

Preventive maintenance is somewhat interchangeable with the term-time-based maintenance. Generally speaking, it is the most often pursued strategy, whether effectively executed or not, by most enterprises, including DCSs. It is the most tried and true as an approach. Often an approach in manufacturer-recommended maintenance guidance, time-based maintenance will often be pursued from the outset, often because of warranty considerations, which will require strict adherence to an OEM schedule based on run-hours or a similar metric. That alone is a valid rationale for its adoption. Managers may still follow the interval long after the expiration of the warranty period. This is not necessarily to be discouraged, but in some instances, it can become evident that the maintenance work is unnecessary or too frequent, (or its converse, that the interval between upkeep is too long, increasing the risk of premature failure). Therefore, criticisms of a straight preventive, time-based approach include the assertion that time-in-service and failure probability are not always linear relationships. Potentially, adherence to a fixed schedule without reference to observed or monitored underlying equipment status has the potential shortcoming of being either wasteful of resources or perhaps not optimally suited to avoiding premature failures either.

Condition-based strategies are intended to try to align maintenance with the need for a given activity, based on some observed or monitored status or condition. In that sense it is designed to overcome some of the perceived drawbacks associated with a maintenance schedule strictly tied to a calendar. Condition-based maintenance, even only applied to select equipment on an opportunistic basis and within limitations, can be a boon to DCS facilities. For this approach to be deployed effectively, the means to determine the condition of the equipment must be in place. Depending on the particular asset in question, determining this can range from simple observation, to monitoring of operating parameters, to the use of sophisticated instrumentation and testing to compile data. Evaluation of this information in turn allows the facility to diagnose the need for maintenance, and deploy resources according to that need, rather than on a rigid schedule. Several advantages are afforded by this approach. Reliability would conceivably be improved as maintenance is now focused somewhat on anticipation of actual need. Resources that would have been directed to scheduled maintenance that has been made unnecessary, or reasonably postponed based on this methodology, can be reduced in number and/or redeployed to other productive activities. A condition-based approach is itself not free of drawbacks. First-cost to implement such a system can be higher, because some means of compiling condition data has to be put in place. Further it also goes without saying that the integrity of the various means of monitoring is of utmost importance. The increasing availability of sophisticated instrumentation and computational technology is helping considerably in this regard however. All told, where feasible, it is intuitive that relating a maintenance strategy to address verified actual condition of equipment is desirable.

Most maintenance programs will combine several of these approaches. Predictive maintenance, which is a type of condition-based maintenance, attempts to further refine the maintenance program by making projections that are normally statistically based. Normally, the statistically based projections are supplemented with nondestructive testing such as infrared imaging, temperature measurement, and vibration measurement and analysis.

Reliability-centered maintenance (RCM) is a sophisticated approach to maintenance that has further developed many of these themes. Adherents to RCM recognize that failures do not necessarily relate to asset age and have extensively studied the nature of equipment failure in depth. RCM also takes into account that both the probabilities and consequences of failure differ for different assets and that resources and maintenance processes should take this into account.

Additional detail on maintenance management is contained in *ASHRAE Handbook—Applications* (2015).

In DCSs where potentially hazardous conditions could be the result of maintenance strategy, risk assessment should also be a component of the maintenance program. Risk-based inspection procedures have long been used in the power, petroleum, and petrochemical industries. Recently the process has been codified by the American Society of Mechanical Engineers (ASME) in a standard (ASME 2007). An overview of this new ASME standard is presented by Sharp et al. (2009). Additionally, a framework for determining "Fitness-For-Service" has been codified by API/ASME (2016), which is particularly useful when evaluating pressure vessels.

The ability to perform maintenance will be significantly impacted by the design of each aspect of the DC system. Favorable maintainability is defined as the ability to maintain the system easily, safely, and in a cost-effective manner. Maintainability should be a design objective and a maintainability review should be part of the design process.

## Chilled-Water Distribution System Maintenance

The chilled-water distribution system is often the most expensive portion of a DCS and thus substantial effort should be made to insure its service life and operational integrity. And while the chilled-water distribution system has few if any moving parts, processes detrimental to its survival and efficiency are often ever present, e.g., corrosion and moisture migration. In addition, while redundancy is often provided for equipment in the central plant, often a chilled-water distribution system will have many sections of piping that represent single points of failure. These vulnerabilities can cause service interruptions that impact a large portion of the customers, or even all of them in the worst-case scenario. Loops in the distribution system can be used to provide redundancy, but only for areas of very high load density or those with critical loads will loops be economically viable.

As discussed in Chapter 4, most chilled-water distribution systems will be directly buried as opposed to being aboveground or in tunnels. The inability to easily inspect and maintain the vast majority of the system once it has been constructed is a major drawback to a buried distribution system. In addition, the nearly universal presence of water in soil presents several challenges for a buried chilled-water distribution system:

- Maintaining a dry environment for the thermal insulation where used
- Providing corrosion protection for all metallic portions of the buried system
- Providing dry environments for appurtenances such as valves, drains, vents, etc.

In areas where feedwater for the DCS is a scarce resource, and thus a significant expense, it becomes even more imperative that the distribution system be of high integrity and leak free. There may also be environmental regulatory requirements that would prohibit significant losses of treated DCS water into the surrounding soil.

The buried chilled-water system can be broken down into two basic components, the buried piping and the manholes. For the buried piping continuous monitoring of its condition is only possible where a leak detection system has been provided. In such cases, monitoring of the leak detection system and rapid response to alarms, or outages, in the leak detection system itself, are essential if the full benefit of the leak detection system is to be achieved. In the absence of a leak detection system, leaks within the distribution system are normally only detected by monitoring makeup water rate, or in the worst case the appearance of a sink hole or some other obvious manifestation of a significant leak. For significant leaks in buried chilled water distribution systems methods such as acoustics that have been used to locate leaks in potable water or pressurized sewers systems may be successful in locating leaks; for example, see Lahlau (undated). Infrared (IR) thermography has been successfully used to locate major leaks in heat distribution systems, and in special circumstances even locate areas where leaks are very small or thermal insulation has been otherwise degraded. For buried chilled-water systems the circumstances where IR thermography will be able to detect the thermal signature of a leaking or damaged pipe will be much more limited than for heat distribution systems owing to the much lower temperature difference between the carrier medium (hot water, steam, chilled water) and the surrounding soil temperatures. The most advantageous circumstances will be high ground temperatures, large pipes, and shallow burial. For cases where a thermal signature is detectable at the ground surface, empirical methods for quantification of IR thermography (Zinko et al. 1996) have been developed but they would not be directly applicable to chilled water distribution systems. However, the analytical method of Phetteplace (1998) should be applicable.

Some buried chilled-water distribution systems have directly buried valves that are basically inaccessible except for the provision for operation that normally will be accessed at the surface via a valve box or access tube. For such valves, routine maintenance should include locating and accessing them as well as exercising via operation, bearing in mind of course the implications of opening/closing valves either on connected customers or the distribution network hydraulics (i.e., water hammer). The frequency of this activity will normally be dictated by the burial conditions; for example valves buried below roads and streets in areas where deicing chemicals are used will likely require much higher frequency of maintenance.

Manholes are normally only found on larger chilled-water distribution systems where they will normally contain valves, leak detection system infrastructure, heat exchangers, metering, etc. The environment within most manholes is often adverse for mechanical equipment, especially those items susceptible to corrosion. Manholes, even when they are well drained (either naturally or by pumps), will still be very high humidity environments. The components of a chilled-water distribution system will often be at temperatures below the dew point for the manhole interior and thus condensation on valves, exposed piping, anchors, etc., will normally be ever-present. Routine maintenance for manholes includes making sure all insulation is intact, sump pumps are operational where equipped, and floors/sumps/drain grates are free of any soil, dirt, or debris. The walls of the manhole as well as the penetrations for the chilled water piping and/or utilities (electrical, communication, forced drain piping, etc.) should be inspected for ground water leakage and corrective actions taken. Stains on walls/floors or other equipment will often provide clues of leakage even when leaks are not present at the time of inspection; look for high water marks as evidence of prior flooding. Inspections should also be made for any leaks on the carrier piping. The structure of the manhole should be inspected by looking for cracks in concrete or rust of steel reinforcement, etc. The frequency of manhole inspections will normally be dictated by the burial conditions, for example manholes buried below the water table or in roads/streets of areas where deicing chemicals are used will likely require much higher frequency of maintenance. Chapter 4 contains many recommendations for manhole design to provide the best possible environment and should be consulted for possible retrofit options where manholes prove to be continuing maintenance problems.

## REFERENCES


ASHRAE. 2007. *ASHRAE handbook—HVAC applications.* Atlanta: ASHRAE.

ASHRAE. 2013a. ANSI/ASHRAE Standard 34, Designation and safety classification of refrigerants. Atlanta: ASHRAE.

ASHRAE. 2013b. ANSI/ASHRAE Standard 15, *Safety standard for refrigeration systems and designation and classification of refrigerants*. Atlanta: ASHRAE.

ASHRAE. 2015. *ASHRAE handbook—HVAC applications.* Atlanta: ASHRAE.

ASHRAE. 2016. *ASHRAE handbook—HVAC systems and equipment*. Atlanta: ASHRAE.

ASHRAE. 2018a. ANSI/ASHRAE Standard 188-2018, *Legionellosis: Risk management for building water systems*. Atlanta: ASHRAE.

ASME. 2007. ASME Standard PCC-3-2007, *Inspection planning using risk-based methods*. New York: ASME.

ASME. 2016. API 579-1/ASME FFS-1, *Fitness-For-Service*. New York: American Society of Mechanical Engineers.

Bellamy, J., and R. Brandon. 1996. A review of European and North American water treatment practices. 1996:N8. Netherlands Agency for Energy and Environment (NOVEM), operating agent for International Energy Agency (IEA).

Cho, Y.I. 2002. *Efficiency of physical water treatments in controlling calcium scale accumulation in recirculating open cooling water system*. ASHRAE Research Project RP-1155 final report. ASHRAE: Atlanta.

DOE. 1998. *Non-chemical technologies for scale and hardness control*. Federal Technology Alert DOE/EE-0162. Washington, DC: Department of Energy.

Gan, F., D.-T. Chin, and A. Meitz. 1996. Laboratory evaluation of ozone as a corrosion inhibitor for carbon steel, copper, and galvanized steel in cooling water. *ASHRAE Transactions* 102(1):395–409.

IDEA. 2008. *District cooling best practices guide*. Westborough, MA: International District Energy Association.

Lahlau, Z. Undated. Leak detection and water loss control. Available at www.nesc.wvu.edu/ndwc/pdf/OT/TB/TB_LeakDetection.pdf. Morgantown, WV: National Drinking Water Clearinghouse at West Virginia University.

Langelier, W.F. 1936. The analytical control of anticorrosion water treatment. *Journal of the American Water Works Association* 28:1500.

Liu, Z., J.E. Stout, L. Tedesco, M. Boldin, C. Hwang, W.F. Diven, and V.L. Yu. 1994. Controlled evaluation of copper-silver ionization in eradicating Legionella pneumophila from a hospital water distribution system. *The Journal of Infectious Diseases* 169:919–22.

Mungani, D., and J. Visser. 2013. Maintenance approaches for different production methods. *South African Journal of Industrial Engineering* 24(3).

Nasrazadani, S., and T.J. Chao. 1996. Laboratory evaluations of ozone as a scale inhibitor for use in open recirculating cooling systems. *ASHRAE Transactions* 102(2):65–72.

Phetteplace, G. 1998. *Heat loss determination for district heating systems using surface temperature measurements*. Report Number ET-ES 98-13. Department of Energy Engineering, Energy Systems Section, Technical University of Denmark.

Ryznar, J.W. 1944. A new index for determining amount of calcium carbonate scale formed by a water. *Journal of the American Water Works Association* 36:472.

Sharp, D., D. Peters, D. Mauney, and M. Tanner. 2009. Probability and consequence. *Mechanical Engineering*: March.

Sperko, W. 2009. Dissimilar metals in heating and AC piping systems. *ASHRAE Journal* 51(4):28–32.

Sullivan, G., R. Pugh, A. Melendez, and W. Hunt. 2010. Operations & maintenance best practices—A guide to achieving operational efficiency, Release 3.0. Washington, DC: Department of Energy's Federal Energy Management Program (FEMP).

Toth, E., and T. Merrill. 2009. Maintaining a "safety culture": Distribution system a focus at NRG thermal. *District Energy*: First Quarter.

WBDG (Whole Building Design Guide). 2012. Secure/safe. www.wbdg.org/design/secure_safe.php.

Zinko, H., Bjärklev, J., Bjurström, H., Borgström, M., Bøhm, B., Koskelainen, L., and Phetteplace, G. 1996. *Quantitative heat loss determination by means of infrared thermography—The TX model*. Annex 4 project report. Sittard, Netherlands: International Energy Agency, District Heating and Cooling. Available from the IEA DHC operating agent: Netherlands Agency for Energy and Environment (NOVEM).

## BIBLIOGRAPHY

ASHRAE. 2018. *ASHRAE handbook—Refrigeration.* Atlanta: ASHRAE.

Lecamwasam, L., J. Wilson, and G. Chokolich (GHD). 2012. Guide to best practice & maintenance & operation of HVAC systems for energy efficiency. Commonwealth of Australia: National Strategy on Energy Efficiency.

Levine, R. 2001. Why absorption chillers fail. *Plant Engineering Magazine*: December.

Majid, M., M. Nasir, and J. Waluyo. 2012. Operation and performance of a thermal energy storage system: A case study of campus cooling using co-generation plant. *Energy Procedia* 14:1280–58.

Meador, R. 1995. Maintaining the solution to operations and maintenance efficiency improvement. *Proceedings of the World Energy Engineering Congress*, Atlanta, Georgia.

Moubray, J. 1997. *Reliability-centered maintenance*, 2d ed. Oxford: Butterworth-Heinemen.

# Appendix A

# Heat Transfer at the Ground's Surface and Subsurface Temperatures

The surface type (e.g., asphalt, concrete, grass) can have a large impact on the heat balance at the ground's surface and the resulting soil temperatures below. The type of surface impacts the heat transfer from radiation, convection, and precipitation. The impacts are well known, and McCabe et al. (1995) observed significant temperature variations because of the type of surfaces and predicted significant impacts for DCSs. While there has not been any detailed study beyond the work of McCabe et al. (1995) on the impacts of surface type on soil temperatures surrounding DCSs specifically, there has been significant study of the impacts of surface type on soil freezing and thawing. For this application, a method of adjusting the air temperature to find an effective surface temperature has been developed. This method is referred to as the *n*-factor method, with *n*-factors having been determined empirically by a number of investigators. Because the impacts of solar radiation in particular are so important, *n*-factors have been developed for the summer (thawing) and winter (freezing) seasons and these factors vary appreciably with surface and climate types. For more discussion of *n*-factors, the reader is referred to Lunardini (1978) and Lunardini (1981), where the theory is explained and values are tabulated. Freitag and McFadden (1997) also contains tabulated values of *n*-factors.

The reader is cautioned when using *n*-factors, because they are not only specific to the surface, they are also site-specific, and thus one should only extrapolate with caution and understanding. With that understanding as a first approximation, lacking other data, the *n*-factor method can be used to estimate soil temperatures beneath various surfaces; an example of the impact will be provided in the example soil temperature calculation.

To illustrate the use of the *n*-factor concept, Figure A.1 has been prepared in a manner similar to Figure 4.9. The same climate and soil has been assumed for Figure A.1 as for the calculation of Figure 4.9. A surface of concrete pavement has been assumed and the *n*-factors have been estimated, based on the data provided by Lunardini (1981), as 0.66 during the freezing season and 1.7 during the thawing season. The calculation is somewhat complex to detail here, but to summarize, it proceeds by first calculating surface temperatures using the air temperatures calculated by the Equation 4.9 at zero burial depth and the assumed *n*-factors. Subsequently, a sinusoidal curve is fitted to these surface temperatures using the method found in Phetteplace et al. (2013). The constants from that sinusoidal curve fit are then used in Equation 4.9 as before, noting that no adjustment would be made to depths for the convective coefficient at the surface as this impact has been included in the *n*-factor. The use of *n*-factors for this purpose is a significant extrap-

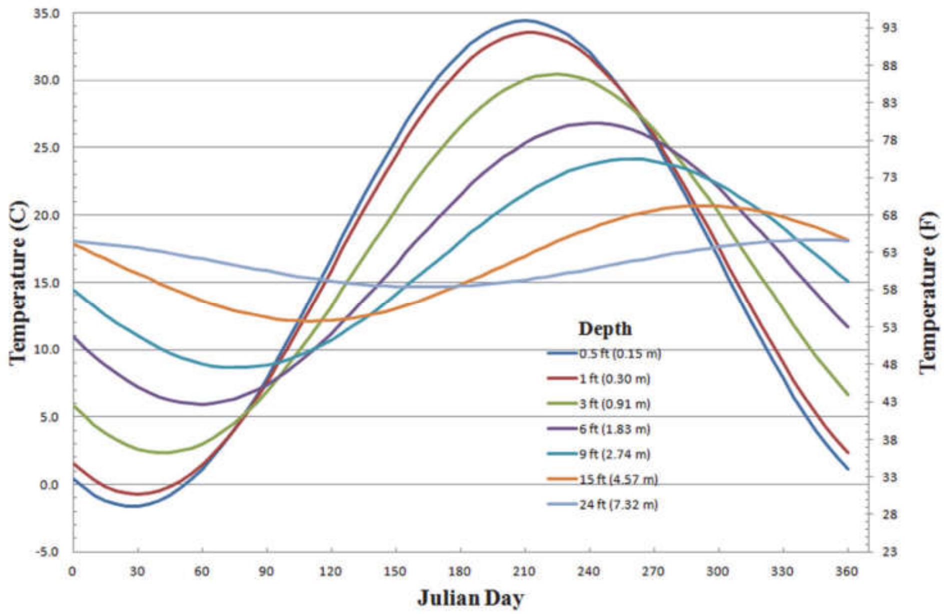



**Figure A.1** Soil temperatures calculated with Equation 4.9 for a coastal Massachusetts climate and the use of *n*-factors to adjust for a concrete pavement surface.

olation of the technique as discussed above, and thus these results should be taken as very approximate. That being said, by comparing Figures 4.3 and A.1, one can see that at 3 ft (0.91 m) of depth, the highest temperature reached in the summer under the concrete pavement is predicted to be about 23°F (13°C) greater than our calculation, which ignored any surface-type impacts and assumed the air temperature and surface temperature were equal.

It is interesting to compare the results of this approximation method with the measurements of McCabe et al. (1995), who found peak summer temperatures of 82°F (28°C) under pavement approximately at a 3 ft (0.91 m) depth. Using the method described above with climatic constants from https://tc0602.ashraetcs.org/Climatic_constants_using_ASHRAE_CD_Ver_6.0.pdf for the Ithaca, New York area, where the measurements of McCabe et al. (1995) were made, the peak ground temperature under a concrete pavement at a depth of 3 ft (0.91 m) is predicted to be 85°F (29°C). This is considered a reasonable agreement given the approximate nature of the method outlined here, as well as the difficulty in making measurements of soil temperatures. Clearly, as noted by McCabe et al. (1995), consideration should be given to surface impacts on subsurface soil temperatures when making calculations to determine appropriate insulation thickness. In addition, other impacts such as those on the materials within CHW distribution systems should be considered.

Accurate undisturbed soil temperatures are a significant concern in DCS design as the temperatures of the carrier fluid are quite close to undisturbed soil temperatures. Thus, errors in undisturbed soil temperature estimation may result in significant errors in estimated heat gain for a DCS. Consider, for example, the peak heat gains from a 40°F (4.4°C) CHW supply pipe buried at 3 ft (0.91 m) in the coastal Massachusetts climate used in Example 4.2. Peak temperature at that depth is estimated at 64°F (18°C) when the surface heat transfer impacts are excluded and 87°F (31°C) when the estimated impacts of a concrete surface pavement are included. The heat gains for the 87°F (31°C) undisturbed soil temperature would be 1.96 times greater [(87–40)/(64–40)] than those for the

64°F (18°C) soil temperature. If, for example, the decision not to insulate the buried piping system (discussed in Chapter 4) had been made with the assumption of the lower ground temperature, it is likely that the result would not be valid for the higher ground temperature. Also, design heat gains for the system in the summer months would be significantly higher than expected and delivered water temperature to the end consumer would be higher than expected.

## REFERENCES

Freitag, D.R., and T. McFadden. 1997. *Introduction to cold regions engineering*. New York: American Society of Civil Engineers Press.

Lunardini, V.J. 1978. Theory of *n*-factors and correlation of data. *Proceedings of the Third International Conference on Permafrost*, pp. 40–46. Edmonton, Alberta.

Lunardini, V.J. 1981. Heat transfer in cold climates. New York: Van Nostrand Reinhold.

McCabe R.E., J.J. Bender, and K.R. Potter. 1995. Subsurface ground temperature—Implications for a district cooling system. *ASHRAE Journal* 37(12):40–45.

Phetteplace, G., Mildenstein, J. Overgaard, K. Rafferty, D.W. Wade, I. Oliker, P.M. Overbye, V. Meyer, and S. Tredinnick. 2013. *District heating guide*. Atlanta: ASHRAE.

# Appendix B

# Case Studies

## CASE STUDY: BUSINESS BAY EXECUTIVE TOWERS

### System Overview

System location: Dubai, United Arab Emirates, Business Bay, across parallel road of Sheikh Zayed Rd, between Intersection 1 and 2
Year of first operation: November 2009
Number of central plants: One plant
Total chiller capacity: 35,200 tons (124,000 kW)
Pumping arrangement: Primary-Secondary
Distribution network length: 4.5 km (2.8 mi)
Maximum distribution pipe size: 1200 mm (48 in.)
Number of customers: 122
Number of buildings connected: 22
Total area connected: 21,313,120 ft$^2$ (1,980,053 m$^2$)
District heating supplied by same plant/provider: No

### System Performance Metrics

Maximum peak load supplied to date: 9200 tons/h
Annual cooling supplied (ton-h): 17,134,022 ton-h (60,260,355 kWh)/8 months
CHWS temperature: 4.5°C (40°F)
Design CHW $\Delta T$: 9°C (16°F)
Average CHW $\Delta T$ achieved: 5°C (9°F)
CHW $\Delta T$ range: 4.8°C–6.2°C (8.6°F–11.2°F)
Average overall plant performance: 1.05–1.08 kW/ton (0.299–0.307 kW/kW)
Distribution system makeup water rate (% of circulation): 0.04 gal/ton-h (0.043 L/kWh)

### Chiller Details

Number of chillers and capacity: 16 chillers at 2200 tons (7700 kW) each
Chiller type: Centrifugal compressor, water cooled
Chiller prime mover: Electric
VSD on chillers: Yes
Chiller arrangement: Series counterflow
Refrigerant: R-134a
Chiller heat exchanger construction: Shell-and-tube total flood type (Steel to Copper)

## Pumping

Number of primary CHW pumps: 9
Pump type: Double suction, horizontal mounted
Rated power: 300 hp (224 kW)
Drive type: CS
Number of secondary CHW pumps: 7
Pump type: Double suction, horizontal mounted
Rated power: 700 hp (522 kW)
Drive type: VS
Is tertiary CHW pumping used: No
Maximum design system circulating head: 5 bar (500 kPa)
Number of condenser-water pumps: 9
Condenser-water pump type: Double suction, horizontal mounted
Condenser-water pumps rated power: 500 hp (373 kW)
Condenser-water pump drive type: CS

## Water Treatment

CHW treatment methods: Nitrite-based treatment for corrosion inhibition with Isothiazolone based biocide for microbiological treatment.
Treatment performed in-house or contracted: In-house
Type of CHW filters/strainers: Bag filters/side-stream strainer
Condenser-water source: Currently using municipal water supply, plan to convert to treated sewage effluent in near future.
Amount of condenser-water storage on site (if any): 3000 $m^3$ (106,000 $ft^3$)
Type of condenser-water filters/strainers: Centrifugal filtration/side-stream strainer
Treatment of condenser water: Organic phosphate-based treatment for scale/corrosion inhibition. Quaternary ammonium salts as dispersants and activated chlorine/bromine as biocides.

## Cooling Towers

Location of cooling towers: Roof
Number of towers: 9 cells
Tower capacity and rating conditions: 4200 tons/12,000 cfm (14,800 kW/5700 L/s])
Tower Type: Induced draft
Tower construction material: High density polyethylene
Tower fill material: Rigid PVC
VS fans used: Yes

## Distribution System

Maximum pipe size: 48 in. (1200 mm)
Minimum pipe size: 20 in. (500 mm)
Carrier pipe material: Carbon steel
Piping location(s) (i.e., direct burial, tunnels, etc.): Direct buried
Method of pipe joining: Butt welding
Type and amount of insulation: Polyurethane. 4 in. (100 mm) thick
Jacket material: HDPE
Method of making field closures of jacket at joints: Manual
Leak detection system installed: No
Cathodic protection used: No
Are manholes used on buried portions: Yes
Manhole construction material: Concrete

## Consumer Interconnect

Number of directly connected buildings: 22
Numbers and types of heat exchangers used: 53/Plate heat exchanger
Design approach temperature for heat exchangers: 16°F (9°C)
Ownership of interconnection: EMPOWER, the district cooling system
Type of metering: Bulk metering/Submetering
Remote monitoring of consumer station and metering: Yes
Remote control of consumer station: Yes
$\Delta T$ or demand penalties in consumer rate structure: Yes

## Special Features

Waste heat recovery: No
Cogeneration: No
Advanced fluids: No
Any other unusual aspects of system: No

## Contact for More Information

Moustapha Assayed
Senior Engineering Manager
EMPOWER Energy Solutions
Emirates Central Cooling Systems Corporation
PO Box 8081
Dubai, United Arab Emirates
Telephone: +9-714-375-5340
GSM: +97150-4691306
Fax: +9-714-375-5500
Email: Moustapha.assayed@empower.ae
Website: www.empower.ae

## CASE STUDY: TEXAS MEDICAL CENTER

### System Overview

System location: Houston, Texas
Year of first operation: 1969
Number of central plants: 2
Total chiller capacity: 120,000 tons
Pumping arrangement: Primary-Secondary
Distribution network length: 35 mi (56.3 km)
Maximum distribution pipe size: 60 in. (1500 mm)
Number of customers: 18
Number of buildings connected: 43
Total area connected: 18.0 million $ft^2$ (1.67 million $m^2$)
District heating supplied by same plant/provider: Yes

### System Performance Metrics

Maximum peak load supplied to date: 65,000 tons (229,000 kW)
Annual cooling supplied: 287,298,000 ton-h ($10^9$ kWh)
CHWS temperature: 40°F (4.4°C)
Design CHW $\Delta T$: 14°F (7.8°C)
Average CHW $\Delta T$ achieved: 14°F (7.8°C)
CHW $\Delta T$ range: 8°F–18°F (4.4°C–10°C)
Average overall plant performance: 0.89 kW/ton (0.25 kW/kW)
Distribution system makeup water rate (% of circulation): <1%

### Chiller Details

Number of chillers and capacity: 22 (1350–7500 tons [4750–26,400 kW]) plus TES
Chiller prime mover: Electric, 10,000 tons (35,200 kW) of steam turbine driven
VSD on chillers: 4–8000 ton chillers
Chiller arrangement: Parallel
Refrigerant: R-22, R-134A, R-12
Chiller heat exchanger construction: Tube-and-shell

### Pumping

Number of primary CHW pumps: 17
Rated power: 350–1250 hp (260–930 kW)
Drive type: Electric and steam
Maximum design system circulating head: 150 psig (1030 kPa)
Number of condenser-water pumps: 19
Condenser-water pumps rated power: 100–900 hp (75–670 kW)
Condenser-water pump drive type: Electric

### Water Treatment

Treatment performed in-house or contracted: Contracted with some self performance
Condenser-water source: Municipal/Well

### Cooling Towers

Location of cooling towers: On site
Number of towers: 10
Tower capacity and rating conditions: 154,000 tons (540,000 kW)
Tower Type: Cross-flow
Tower construction material: Wooden/Concrete/Fiberglass

Tower fill material: PVC/Ceramic/Wood
VS fans used: 6 towers are VSD, online June 2011

## Thermal Storage

Type: CHW
Capacity: 8.75 million gal (33 million L)
Vessel construction material: Steel tank, 100 ft (30 m) diameter, 150 ft (45 m) tall

## Distribution System

Maximum pipe size: 60 in. (1500 mm)
Minimum pipe size: 4 in. (100 mm)
Carrier pipe material: Welded steel, schedule 40
Piping location(s) (i.e., direct burial, tunnels, etc.): Direct buried/tunnel
Method of pipe joining: Welded
Type and amount of insulation: CHW (coal-tar epoxy), Steam (Gilsulate)
Cathodic protection used: Yes
Are manholes used on buried portions: Yes
Manhole construction material: Steel
Manhole drainage method: Pumped

## Consumer Interconnect

Number of directly connected buildings: 43
Remote monitoring of consumer station and metering: Yes
Remote control of consumer station: No
$\Delta T$ or demand penalties in consumer rate structure: Yes

## Special Features

Waster heat recovery: Yes
Cogeneration: Yes
Advanced fluids: No
Any other unusual aspects of system: No

## Contact for More Information

Steve Swinson
President and CEO
Thermal Energy Corporation
1615 Braeswood Blvd.
Houston, TX 77030
Phone: 713-791-6765
Email: sswinson@teco.tmc.edu

**Figure B.1** Aerial view of Texas Medical Center Plant.

# CASE STUDY: DISTRICT COOLING ST. PAUL

## System Overview

System location: St. Paul, Minnesota
Year of first operation: 1993
Number of central plants: Two plants plus satellite chillers
Total chiller capacity: 24,637 tons (86,650 kW)
Pumping arrangement: Primary-Secondary
Distribution network length: 37,800 ft (11,500 m) (dual-pipe system)
Maximum distribution pipe size: 30 in. (750 mm)
Number of buildings connected: 100
Total area connected: 19.2 million $ft^2$ (1.78 million $m^2$)
District heating supplied by same plant/provider: Yes, since 1983

## System Performance Metrics

Maximum peak load supplied to date: Approximately 24,000 tons (84,400 kW)
Annual cooling supplied: 33.6 million ton-h (118 million kWh)
CHWS temperature: 42°F (5.6°C)
Design CHW $\Delta T$: 16°F (9°C)
Average CHW $\Delta T$ achieved: 14°F (7.8°C)
Average overall plant performance: Approximately 0.9 kW/ton (0.26 kW/kW)

## Electric Details

Primary voltage and frequency:13.8 kV
Secondary voltage and frequency:Various

## Chiller Details

Number of chillers and capacity: 15 electric drive centrifugal (23,637 tons [83,131 kW]); 2 steam absorption (1000 tons [3500 kW])
Chiller type: See above
Chiller prime mover: Electric/Low-pressure steam
VSD on chillers: No
Chiller arrangement: Parallel
Refrigerant: Various (R-22, R-134, R-123)

## Water Treatment

CHW treatment methods: Molybdenum-based
Treatment performed in-house or contracted: Contracted
Type of CHW filters/strainers: High-efficiency media filters
Condenser-water source: City water
Amount of condenser-water storage on site if any: None
Type of condenser-water filters/strainers: Sand filter
Treatment of condenser-water: Typical open-loop chemistry

## Cooling Towers

Location of cooling towers: Roof
Number of towers: 10
Tower Type: Various

## Thermal Storage

Type: CHW
Capacity: 2 tanks with total capacity of 6.7 million gal/66,000 ton-h (25,400 $m^3$)
Vessel construction material: Steel

## Distribution System

Maximum pipe size: 30 in. (750 mm)
Minimum pipe size: 3 in. (75 mm)
Carrier pipe material: Steel with protective coating wrap (majority), HDPE (2710 ft [826 m] dual-pipe system)
Piping location(s), i.e., direct burial, tunnels, etc.: Direct burial (majority), tunnels (380 ft [115 m] dual-pipe system)
Method of pipe joining: Welded (steel), fused (HDPE)
Type and amount of insulation: None, if depth is greater than 2 ft (600 mm); flat insulation board over shallow pipes if depth is less than 2 ft (600 mm)
Jacket material: NA
Method of making field closures of jacket at joints: NA
Leak detection system installed: No, track makeup water
Cathodic protection used: Yes, passive anodes
Are manholes used on buried portions: No
Manhole construction material: N/A
Manhole drainage method: N/A

## Consumer Interconnect

Number of directly connected building: Approximately 50
Numbers and types of heat exchangers used: Various
Design approach temperature for heat exchangers:
Ownership of interconnection: Typically owned by customer/building
Type of metering: Magnetic flowmeter with BTU computer
Remote monitoring of consumer station and metering: Yes
Remote control of consumer station: No
$\Delta T$ or demand penalties in consumer rate structure: Yes, $\Delta T$

## Special Features

Cogeneration: Yes, with biomass

## Environmental and Economic Benefits

Renewable (Biomass) fuel proving up to 70% of the fuel input for the combined district heating and electric generation. $CO_2$ emissions reduced by up to 280,000 tons/year (985,000 kW/year).

## Published Articles on the System or Websites with Details

http://www.districtenergy.com/pdf/AnnualReport2008.pdf

Schuerger, M. 1994. A decade of service leads to new business opportunities. *District Energy* 80(1):5–7.

Sherwood, T. 1998. St. Paul Then and now: fifteen years of progress. *District Energy* 83(4):22–28.

## Contact for More Information

www.districtenergy.com/

## CASE STUDY: ABDALI AREA, AMMAN, JORDAN

### System Overview

System location: Abdali Area in Amman, Jordan
Year of first operation: 2009 (first phase)
Number of central plants: 1
Total chiller capacity: 30,500 TR (chillers + TES Tank). Chillers alone: 10,000 TR
Pumping arrangement: Primary and secondary pumping groups
Distribution network length: 6.2 miles (10 km)
Maximum distribution pipe size: 48 in. (1200 mm) diameter
Number of customers: —
Number of buildings connected: 59 ETSs
Total square footage connected: 19 million $ft^2$ (1.8 million $m^2$)
District heating supplied by same plant/provider: Yes

### System Performance Metrics

Maximum peak load supplied to date: 16,000 TR
Annual cooling supplied (ton-h): 44,851,200
Chilled water supply temperature: 40°F (4.4°C)
Design chilled water $\Delta T$: 16°F (8.9°C)
Average chilled water $\Delta T$ achieved: 14°F (7.8°C)
Chilled water $\Delta T$ range:—
Average overall plant performance (kW/ton): 0.62 kW/TR
Distribution system makeup water rate (% of circulation): No makeup water system

### Chiller Details

Number of chillers and capacity: 4 chillers. Each chiller capacity is 2500 TR.
Chiller type: Air cooled
Chiller prime mover: Electrical medium voltage motors (3300 V/50Hz)
Variable-speed drive on chillers: No
Chiller arrangement: Series
Refrigerant: Ammonia (R717)
Chiller heat exchanger construction: Stainless steel plate heat exchangers evaporators

### Pumping

Number of primary chilled water pumps: 4

Pump type: Centrifugal type (end suction horizontal split case)

Rated horsepower: 160 kW per pump

Drive type: Direct online

Number of secondary chilled water pumps: 5

Pump type: Centrifugal type

Rated horsepower: 560 kW

Drive type: Variable-speed drive

Is tertiary chilled water pumping used: No

Maximum design system circulating head: 246 ft (75 m)

Number of condenser water pumps: No condenser water pumps

Condenser water pump type: Not applicable

Condenser water pumps rated horsepower: Not applicable

Condenser water pump drive type: Not applicable

## Water Treatment

No cooling tower, so no condenser water treatment
Chilled water treatment methods: Molybdenum based chemical treatment
Treatment performed in-house or contracted: Contracted
Type of chilled water filters/strainers: Screen filters
Condenser water source: There is no condenser water
Amount of condenser water storage on site if any: Not applicable
Type of condenser water filters/strainers: Not applicable
Treatment of condenser water: Not applicable

## Air-Cooled Condensers

Location of air cooled condensers: Over the cooling plant
Number of air cooled condenser circuits: There are 2 circuits, one per two chiller
Tower capacity and rating conditions:
Air cooled condenser construction material: Stainless steel.
Variable-speed fans used: —

## Thermal Storage

Type: Underground, naturally stratified water with H-Type FRP diffusers
Capacity: 180,000 TR·h
Vessel construction material: Reinforced concrete

## Distribution System

Maximum pipe size: 1200 mm diameter
Minimum pipe size: 100 mm diameter
Carrier pipe material: Carbon Steel
Piping location(s), (i.e., direct burial, tunnels, etc): Direct burial
Method of pipe joining: Welding
Type and amount of insulation: Preinsulated pipe—Polyurethane (PUR) foam of a thickness range from 2 to 4 in. (50 to 100 mm) depending on the pipe size
Jacket material: High-density polyethylene (HDPE)
Method of making field closures of jacket at joints: Shrinkable joint
Leak detection system installed: No
Cathodic protection used: No
Are manholes used on buried portions: Valve chambers
Manhole construction material: Reinforced concrete
Manhole drainage method: Pit with submersible pumps

## Consumer interconnect

Number of directly connected building: 59
Numbers and types of heat exchangers used: 425
Design approach temperature for heat exchangers: 1.8°F (1°C)
Ownership of interconnection: Plant owner
Type of metering: Magnetic flowmeters and temperature transmitters connected to an energy meter. Individual PLC + SCADA for ETS control and data record.
Remote monitoring of consumer station and metering: Yes
Remote control of consumer station: Yes
$\Delta T$ or demand penalties in consumer rate structure: No

## Special Features

Waste heat recovery: Heating system is designed with heat pumps so the heat from the chillers condensers is rejected to atmosphere/wasted but it is used as useful heating energy.

Cogeneration: No

Advanced fluids: R717 refrigerant

Any other unusual aspects of system: 2500 TR air-cooled chillers.

## Environmental and Economic Benefits

The Abdali district cooling plant is quite unique because the yearly efficiency is quite similar to a typical DC plant with cooling towers but with zero water consumption. This is the reason why the plant is very beneficial especially in areas like the Middle East where water saving is as important or even more important as energy saving.

According to the reports (see list of published articles and websites at the end of this case study) made by several reputable consultants and organizations (like the National Energy Research Center in Jordan), the Abdali district heating and cooling plant significantly cuts the energy bill by 40% when compared to standalone air-cooled chillers and decreases water consumption by 95%, compared to other plants and designs of the same nature. It was demonstrated that the central plant contributes to the reduction of $CO_2$ emissions by 13,700 tons (12.5 million kg) annually, compared to a standalone air-cooled chiller.

The water savings are especially important for a country with water scarcity. Almost 40 million gallons (500,000 $m^3$) of water are saved thanks to the selected configuration with high efficiency and high capacity air-cooled chillers.

The use of an environmentally friendly refrigerant like R717 (natural refrigerant) is also important to preserve the environment. The global warming potential of R717 is zero.

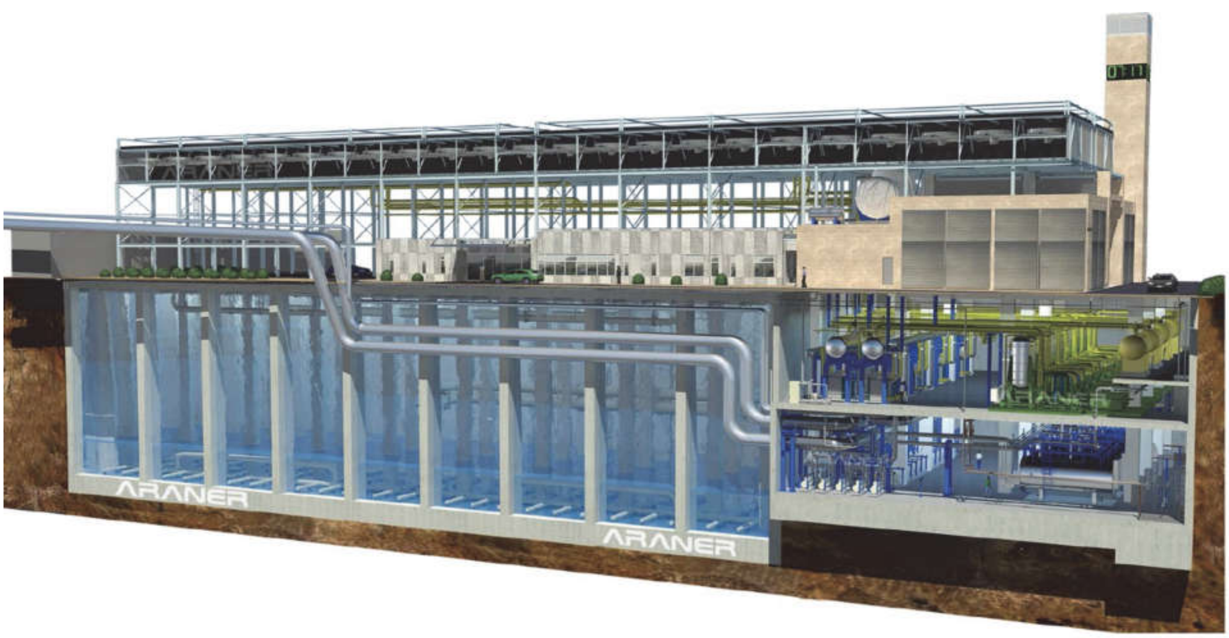



**Figure B.2** Conceptual view of the Abdali Area Central Plant.

## Published Articles on the System or Websites with Details

Abdali Development. 2012. Abdali PSC Overview. http://www.abdali.jo/index.php?r=site/page&id=14.

ARANER. 2016. Abdali (Jordan): A District Cooling Reference in the Middle East. http://www.araner.com/blog/abdali-jordan-district-cooling-solution/.

ARANER. Undated. District cooling: Efficiency improvement in district cooling system using direct condensation. Presented at the International District Energy Association Conference. https://www.districtenergy.org/HigherLogic/System/DownloadDocumentFile.ashx?DocumentFileKey=faebaffc-7f07-a207-3a77-98f655a136b3&forceDialog=0.

Climate Control. 2017. District Cooling in the air in Jordan. http://climatecontrolme.com/2017/04/district-cooling-in-the-air-in-jordan/.

Danish Board of District Heating. 2014. EBRD provides $30m loan to promote energy efficiency in Jordan. https://dbdh.dk/ebrd-provides-30m-loan-to-promote-energy-efficiency-in-jordan/.

European Bank for Reconstruction and Development. 2014. Abdali District Heating and Cooling. http://www.ebrd.com/work-with-us/projects/psd/abdali-district-heating-and-cooling.html.

EU-GCC Clean Energy Technology Network. 2018. District cooling: Heat rejection. http://www.eugcc-cleanergy.net/sites/default/files/events/20180410_dubai/clean_cooling_workshop_dubai_09.04.2018_session_2b_alejandro_subiza_araner.pdf.

## Contact for More Information

Guillermo Martinez
Commercial Manager
ARANER
www.araner.com
email: g.martinez@araner.com

# Appendix C

# Terminology for District Cooling

Some of the following content is excerpted from *ASHRAE Terminology of Heating, Ventilating, Air Conditioning, and Refrigeration* (ASHRAE 1991); however, terminology specific to district cooling systems has been added.

## A

**Absorbate:** that substance absorbed by an absorbent.

**Absorbent:** material which, due to an affinity, extracts one or more substances from a liquid or gaseous medium with which it is in contact and which changes physically or chemically, or both, during the process. Calcium chloride is an example of a solid absorbent, while solutions of lithium chloride, lithium bromide, and the ethylene glycols are examples of liquid absorbents.

**Absorber:** device containing fluid, or other material, for absorbing refrigerant vapor or other vapors.

**Absorption:** process whereby a porous material extracts one or more substances from an atmosphere, a mixture of gases, or a mixture of liquids.

**Adjustable-frequency drive (AFD):** electronic device that varies its output frequency to vary the rotating speed of a motor, given a fixed input frequency. Used with fans or pumps to vary the flow in the system as a function of a maintained pressure.

**Adsorbent:** material that has the ability to cause molecules of gases, liquids, or solids to adhere to its surfaces without changing the adsorbent physically or chemically. Certain commercially available solid materials, such as silica gel, activated carbon, and activated alumina, have this property.

**Adsorption:**

1. process in which fluid molecules are concentrated on a surface by chemical or physical forces, or both.
2. surface adherence of a material in extracting one or more substances present in an atmosphere or mixture of gases and liquids, unaccompanied by physical or chemical change.

**Air eliminator (air vent):** in a steam or water distribution system, a device which closes if either steam or water is present in the vent body, and opens when air or noncondensables reach it.

**Authority:** (of a controller such as a control valve)**:** ratio of effect on a manipulated variable of one input signal as compared to that of another.

## B

**Blowdown:**

1. discharge of water from a steam boiler or open recirculating system that contains high total dissolved solids. The addition of makeup water will reduce the concentration of dissolved solids to minimize their precipitation.
2. in pressure-relief devices, the difference between actuation pressure of a pressure-relief valve, and reseating pressure, expressed as a percentage of set pressure, or in pressure units.

**Branch:** in piping, or conduit; another section of the same size or smaller, at an angle with the main.

## C

**Carrier pipe:** pipe that carries the heating or cooling medium (steam, hot water, chilled water). Has more stringent requirements than a service pipe in a domestic water system. For example, compliance with the pressure vessel code is required.

**Carrier pipe insulation:** insulation that surrounds the carrier pipe. Usually mineral fiber, calcium silicate, foam glass, polyurethane, or polyisocyanuate foam. May be more than one layer and more than one type of material.

**Casing:** a thin-wall pipe that encapsulates the carrier pipe and carrier pipe insulation to prevent the insulation from getting wet. Usually steel, high density polyethylene or fiberglass-reinforced plastic (FRP).

**Cell** (in a cooling tower)**:** smallest tower subdivision which can function as an independent heat exchange unit. It is bounded by exterior walls or partitions. Each cell may have one or more fans or stacks and one or more distribution systems.

**Cellular glass insulation:** An insulation material manufactured of glass and carbon to form a structure with millions of hermetically sealed cells.

**Chiller:**

1. refrigerating machine used to transfer heat between fluids.
2. complete, indirect refrigerating system of compressor, condenser, and evaporator with all operating and safety controls.

   ***absorption chiller:*** refrigerating machine using heat energy and absorption input to generate chilled water.
   ***mechanical chiller:*** refrigerating machine using mechanical energy input to generate chilled water.

**Chiller barrel:** shell-and-tube evaporator used to cool water or a secondary coolant. Note: Term applies only to water-chilling packages, not to heat exchangers.

**Cogeneration:** sequential production of either electrical or mechanical power and useful thermal energy (heating or cooling) from a single energy form. See also electric power generation (cogeneration).

**Combined heat and power (CHP) system:** system combining power production with the use of a lower quality heat by-product of power generation for district heating.

**Community energy system:** centralized facility for generation and distribution of the heating and cooling needs of a community, rather than individual heat or cold generators (i.e., furnace or air conditioner) at each residential, commercial, or institutional site.

**Compressor surge:** condition achieved in a centrifugal compressor when the momentum of the refrigerant gas through the compressor is insufficient to overcome the thermal lift requirement. Direction of flow temporarily reverses through the compressor until the lift requirement decreases. The condition repeats until the operating condition is corrected. Accelerated wear and damage can eventually result.

**Consumer interconnection:** See **consumer interface**.

**Consumer interface:** the interface between the consumer of district heating or cooling and the district heating and cooling utility, normally within the building. The consumer interface will normally include controls and may also include metering and heat exchanger(s) where required by the installation specifics.

**Cycle of concentration:**

1. in boilers, the ratio of chlorides in the boiler water to the chlorides in the feedwater.
2. in cooling tower operation, the ratio of chlorides in the recirculating cooling tower water to the chlorides in the makeup water.

## D

**DCP building services:** plant MEP (nonprocess) services such as HVAC, lighting, plumbing, etc.

**DCP process:** plant equipment that is responsible for maintaining the CHW generation such as chillers, CHW pumps, cooling towers, etc.

**Design professional:** individual responsible for the design and preparation of architectural or engineering contract documents. Also see **Engineer-of-Record**.

**Desorption:** liberation of a gas held in a substance by sorption.

**District cooling:** concept of providing and distributing, from a central plant, cooling to a surrounding area (district) of tenants or clients (residences, commercial businesses, or institutional sites). Compare **district heating.**

> ***district cooling system cooling density*:** measure of cooling demand per unit area. Customary units are kW/hectare, or tons/acre.

## E

**Energy transfer station:** See **consumer interface**.

**Engineer of record (EOR):** the technical person who is legally responsible for the design of the project. A person registered by a state or government to be qualified to make design and construction decisions for the specific type of project being designed and constructed. A person with experience with several projects of the same type. Also see **Professional Engineer** and **Registered Engineer**.

**Entry pit:** a structure located immediately outside of a building or inside of a building that is being serviced by an underground heating or cooling distribution system, usually partially below grade. It has many of the same features of a valve vault, including adequate room for maintenance on appurtenances, the ability to keep groundwater out, a positive drainage system, electrical power for electrical sump pumps and lights, effective ventilation to control temperature and humidity, and safety features for maintenance workers.

**Expansion bend:** bend, usually a loop, put into a pipe run to relieve stresses induced by expansion and contraction from temperature changes.

**Expansion joint:** device in a structure, a pipe run, etc., that can by linear compensation accept variation of length from expansion or contraction due to temperature changes.

## F

**FRP:** fiberglass-reinforced plastic, a material used primarily in the construction of pipes and tanks.

**Field joint:** location in a piping system where the carrier piping segments are joined together. In a preinsulated piping system the field joint includes joining both the carrier piping as well as insulating the joint and providing a leaktight jacket across the joint.

## G

**GRP:** see **fiberglass-reinforced plastic (FRP)**.

## H

**HDPE:** high density polyethylene, a common material used in piping systems. Also see **PEX**.

**HHOT:** hand-held operator terminal. This is a compact, portable device, which can be connected directly to a communications port, or via wireless technology, on a controller to interrogate, program, and control system parameters.

**Hydraulic decoupler:** a cross-connection between supply and return piping, e.g. at the chilled-water plant used to decouple the flow through the chillers from that in the distribution system.

## I

**Ice builder:** refrigerated coils immersed in a tank of water used for forming ice, and to provide ice water. Compare thermal storage device.

**Ice harvester:** machine that manufactures ice on a cooling surface, then delivers it to storage.

**Ice slurry (liquid ice):** suspension of ice crystals in a secondary coolant.

**Ice storage system:** thermal storage system, used for chilling processes or for comfort cooling that uses the phase change of ice to water. Ice is formed during periods of low refrigerating demand for delivery of cooling during periods of high refrigerating demand.

## J

**Jacket:**

1. sealed space around a piece of equipment or a storage unit through which a thermal medium can be circulated.
2. integral covering, sometimes fabric reinforced, which is applied over insulation, or the core, shield, or armor of a cable to provide mechanical or environmental protection.
3. a thin-wall pipe or watertight plastic wrap on the outside of the insulation that is exposed to soil or the weather. Metal jackets are typically aluminum or stainless steel. Plastic jackets are typically polyvinylchloride, high density polyethylene and fiberglass-reinforced plastic.

## L

**Leak plate:** a flat, circular ring on the outside of the conduit that is bonded to the conduit and to the valve vault wall to prevent groundwater from entering the valve vault at the conduit wall penetration.

## M

**Mechanical joint:** general form for gastight joints obtained by joining metal parts through a positive holding mechanical construction (such as flanged joint, screwed joint, flared joint). In utility, piping may consist of joints sealed by O-rings or lip type seals, as distinguished from a fused (e.g., welded) or cemented joint.

## P

**Peer-to-peer:** communication structure over which the network connected devices can be configured to communicate directly without needing any arbiter in between (masters and slaves are not declared), and can gain access to the media without any communication restrictions.

**PEX:** high density polyethylene (HDPE) that has been cross-linked; a process that provides higher strength at elevated temperatures. Also see **HDPE**.

**Pressure dependent (PD):** varying flow rate through a flow control device in response to changes in pressure.

**Pressure independent (PI):** uniform flow rate through a flow control device unaffected by changes in system pressure.

**Product integrated control (PIC):** direct digital control (DDC) panel, factory mounted and connected, able to monitor, control, and diagnose the significant functions of the equipment of which it is a part.

**Professional engineer (licensed engineer):** designation reserved, usually by law, for a person professionally qualified and duly licensed to perform engineering services such as civil, electrical, mechanical, sanitary, and structural. See also **Engineer-of-Record** and **Registered Engineer**.

## R

**Registered engineer:** appropriately qualified and licensed professional engineer. See also **Professional Engineer** and **Engineer-of-Record (EOR)**.

## S

**Sorbate:** substance absorbed by or adsorbed on a sorbent.

**Sorbent:** material which extracts one or more substances present in an atmosphere or mixture of gases or liquids with which it is in contact, due to an affinity for such substances.

**Sorption:** general term covering both absorption and adsorption.

**Storage cycle (thermal storage):** complete charge and discharge of a thermal storage device.

**Stratification:** division into a series of layers, as with thermal gradients across a stream.

## T

**TCP/IP:** Transmission Control Protocol, a protocol suite developed by the US Department of Defense to permit different types of computers to communicate and exchange information with one another.

**Thermal insulation:** material or assembly of materials used to provide resistance to heat flow.

> ***blanket thermal insulation*:** relatively flat and flexible insulation in coherent form, furnished in units of substantial area.
>
> ***block thermal insulation*:** rigid insulation preformed into rectangular units.
>
> ***board (slab) thermal insulation*:** semirigid insulation preformed into rectangular units having a degree of suppleness particularly related to their geometrical dimensions.
>
> ***cellular elastomeric (cellular rubber) thermal insulation*:** insulation composed principally of natural or synthetic elastomers, or both, processed to form a flexible, semirigid or rigid foam, having a predominately closed-cell structure.
>
> ***cellular polystyrene thermal insulation board*:** insulation composed of cellular polystyrene in the form of boards, produced by heat and pressure from expansion of foamable polystyrene beads within a mold (bead board), or by in-situ foaming of molten polystyrene in an extrusion mode (extruded board).

***cellular polyurethane thermal insulation*****:** insulation composed principally of the catalyzed reaction product of polyisicyanate and polyhydroxy compounds, processed usually with fluorocarbon gas to form a rigid foam having a predominately closed-cell structure.

***fill thermal insulation (loose-fill)*****:** insulation in granular, nodular, fibrous, powdery, or similar form designed for installation by pouring, blowing, or hand placement. Examples are mineral or glass fiber, cellulosic fiber, diatomaceous silica, perlite, silica aerogel, and vermiculite.

***foamed-in-place thermal insulation (foam-in-situ insulation)*****:** insulation formed by introducing into prepared cavities a chemical component and a foaming agent that react to fill the space with a foamed plastic.

***mineral fiber thermal insulation*****:** insulation composed principally of fibers manufactured from rock, slag, or glass, with or without binders.

***perlite thermal insulation*****:** insulation composed of natural perlite ore, a glassy volcanic rock expanded by heat to form a cellular structure.

***reflective thermal insulation*****:** insulation which reduces radiant heat transfer across spaces by use of one or more surfaces of high reflectance and low emittance, for example, aluminum foil.

**Thermal storage:**

1. temporary storage of high or low-temperature energy for later use.
2. accumulation of energy in a body or system in the form of sensible heat (temperature rise) or latent heat (change of phase).
3. technology or systems of accumulating cooling or heating capacity for subsequent use. See also thermal storage system.

***cool storage*****:** technology or systems used to store cooling capacity.

***ice-on-coil thermal storage*****:** container (tank) in which ice is formed on tubes or on pipes.

***heat storage*****:** technology or systems used to store heating capacity.

***latent storage*****:** use of a phase change of a medium for storing heating or cooling capacity.

***naturally stratified storage*****:** thermal storage in which temperature stratification is achieved and maintained by density differences alone, and not by mechanical separators.

***sensible storage*****:** use of a change in temperature of a medium for storing heating or cooling capacity.

***stratified storage*****:** thermal storage vessel in which a thermocline exists.

**Thermal storage system:** system wherein the load demand is met by stored thermal energy.

***chiller-aided storage*****:** thermal storage system that has a chiller to supplement cooling capacity of the storage. Also called live load chilling.

***compressor-aided storage*****:** operation of the compressor of an ice storage system during the discharging period.

***demand-limited storage*****:** thermal storage system controlled to limit the electric power demand.

***direct ice contact (external melt)*****:** ice storage system using a method of heat exchange in which ice is formed by direct refrigeration and melted by immersion in circulating water or secondary coolant. Also called static direct contact storage.

***full storage*****:** thermal storage system having capacity to meet all on-peak cooling or heating requirements by being charged off-peak, and without energy added on-peak.

***indirect ice contact (internal melt)*****:** ice storage system using a method of heat exchange in which ice in containers is formed and melted by a circulating secondary coolant enclosed in a pipe or tube.

**Partial storage system:**

1. system wherein the load demand is met by a combination of stored thermal energy and an energy conversion device.

2. system that has to be operated during on-peak as well as in off-peak periods. See demand-limited storage; compressor-aided storage; load-leveling storage.

**Thermocline:** layer of fluid in which the temperature and density gradient is greater than, and which separates, the cooler fluid below it and the warmer fluid above it.

**Cooling tower ton:** total heat rejection capacity of a cooling tower serving an electric centrifugal chiller; traditionally, 15,000 Btu/h (4.396 kW). Note: This value is based on 25% compressor heat added to a ton of refrigeration.

**Ton-day of refrigeration:** heat removed by a ton of refrigeration operating for a day, 288,000 Btu (approximately 84.3 kW). It is a quantity approximately equal to the latent heat of fusion or melting of 1 ton (2000 lb [907.2 kg]) of ice, from and at 32°F (0°C).

**Ton-hour:** quantity of thermal energy in tons (12,000 Btu [3.517 kW]) absorbed or rejected in one hour.

**Treated sewage effluent:** the end product of the sewage treatment process which may be used for applications such as condenser water in regions where fresh water is limited.

## V

**Valve vault:** A valve vault is distinguished from a manhole because of extra features that help improve the life expectancy of a heating or cooling distribution system. A valve vault has adequate room for maintenance on appurtenances, provisions to attempt to preclude the entrance of ground water out, a positive drainage system, electrical power for electrical sump pumps and lights, effective ventilation to control temperature and humidity, and safety features for maintenance workers.

## Comprehensive, Current Design Guidance for District Cooling Systems

District Cooling Guide, Second Edition, is a complete revision of the first edition, providing updated design guidance for all major aspects of district cooling systems, including central chiller plants, chilled-water distribution systems, and consumer interconnection.

Each chapter of this guide has been updated to reflect the global growth of district cooling systems. The core chapters on central plants, chilled-water distribution systems, and consumer interconnection have been expanded to include new examples, an efficiency analysis between chiller arrangements, geotechnical considerations, and design measures for avoiding low-ΔT syndrome. The chapter on thermal energy storage provides more information on latent heat storage and a comparison of energy storage technologies. Guidance on operations and maintenance has been rewritten to include several new topics, including metrics. Appendices provide updated case studies, including a case study of an air-cooled district cooling system in a high-ambient-temperature region.

For more in-depth analysis, District Cooling Guide, Second Edition, contains a wealth of references to information sources and publications where additional details may be found.

Drawing on the expertise of a diverse international team, this guide is a useful resource for both the inexperienced designer as well as those immersed in the industry, such as consulting engineers, utility engineers, central plant design engineers, and chilled-water system designers.

ISBN 978-1-947192-15-7 (paperback)
ISBN 978-1-947192-16-4 (PDF)

9 781947 192157

Product code: 90565 4/19

ASHRAE
1791 Tullie Circle
Atlanta, GA 30329-2305
404-636-8400 (worldwide)
www.ashrae.org

ASHRAE®

# OWNER'S GUIDE for BUILDINGS SERVED by DISTRICT COOLING

## Comprehensive Reference for Building Owners

District Cooling System Attributes • Design Necessities • Coping with Deficiencies

# Owner's Guide for Buildings Served by District Cooling

Funding for this publication was generously provided by
**Empower Energy Solutions, Dubai, United Arab Emirates.**

## PREPARED BY

**Steve Tredinnick, PE, CEM**
Burns & McDonnell Eng. Co., Inc.
Downers Grove, IL

**Gary Phetteplace, PhD, PE**
Principal Investigator
GWA Research LLC
Lyme, NH

**Brian P. Kirk**
Consultant
Tuckahoe, NY

## REVIEWERS

**Tariq Alyasi**
Empower Energy Solutions
Dubai, United Arab Emirates

**Samer Khoudeir**
Empower Energy Solutions
Dubai, United Arab Emirates

**Alaa A. Olama**
Independent Consultant
Cairo, Egypt

**Hassan Younes**
Griffin Project Development Consultants
Dubai, United Arab Emirates

**Updates and errata for this publication will be posted on the ASHRAE website at www.ashrae.org/publicationupdates.**

# Owner's Guide for Buildings Served by District Cooling

ASHRAE

Atlanta

ISBN 978-1-947192-26-3 (paperback)
ISBN 978-1-947192-27-0 (PDF)


1791 Tullie Circle, NE
Atlanta, GA 30329
www.ashrae.org


Cover design based on the cover of *District Cooling Guide*, First Edition, by Laura Haass



---

Library of Congress Cataloging-in-Publication Data

Names: ASHRAE (Firm), author.
Title: Owner's guide for buildings served by district cooling.
Description: Atlanta, GA : ASHRAE, [2019] | Includes bibliographical references.
Identifiers: LCCN 2019018637| ISBN 9781947192263 (pbk.) | ISBN 9781947192270 (pdf)
Subjects: LCSH: Air conditioning from central stations--Handbooks, manuals, etc.
Classification: LCC TH7687.75 .D47 2019 Suppl. | DDC 697.9/3--dc23 LC record available at https://lccn.loc.gov/2019018637

---

**ASHRAE STAFF**

**SPECIAL PUBLICATIONS**
**Cindy Sheffield Michaels**, Editor
**James Madison Walker**, Managing Editor of Standards
**Lauren Ramsdell**, Associate Editor
**Mary Bolton**, Editorial Assistant
**Michshell Phillips**, Editorial Coordinator

**PUBLISHING SERVICES**
**David Soltis**, Group Manager of Electronic Products and Publishing Services
**Jayne Jackson**, Publication Traffic Administrator

**DIRECTOR OF PUBLICATIONS AND EDUCATION**
**Mark Owen**

# Contents

## *Chapter 3 · Existing Buildings: When Design Deficiencies or Other Constraints Prevent Achieving Acceptable ΔT*



## *Appendix A · Plant Efficiency Impacts from Low ΔT at Customers*



## *Appendix B · Case Study in Mitigation of Low ΔT*

# Preface

The purpose of this guide is to help building owners understand what district cooling is and why they should consider it when deciding how to meet the cooling needs of their building(s). The guide also explains what is different about buildings that are served by district cooling when compared to buildings that meet their cooling need with on-site cooling equipment. High-level guidance on designing buildings for service with district cooling that the building owner should make his designer aware of is provided, and references that the designer may refer to for more detailed information are also provided. Dealing with the issues of retrofitting buildings previously not served by district cooling to district cooling service is also covered.

# Acknowledgments

District cooling (DC) continues to see increasing interest worldwide. Over five years ago, ASHRAE, via their research program, and with supplemental funding from Empower Energy Solutions, undertook the development of a *District Cooling Guide*. Since that first edition of the *District Cooling Guide* was published, significant developments have occurred, and ASHRAE, again with sponsorship from Empower Energy Solutions, has created an updated second edition of the *District Cooling Guide*. While the *District Cooling Guide*, Second Edition provides comprehensive guidance to the designers and operators of DC systems, a guide tailored to owners of buildings served by DC or owners evaluating the option of using DC was needed. The *Owner's Guide for Buildings Served by District Cooling* is intended to fill that need for building owners and also provide guidance to the designers they employ who may not be familiar with building design for DC service.

As the principal investigator for the project to develop this guide, I would like to first thank ASHRAE for allowing my team the opportunity to undertake this effort and Empower Energy Solutions (Ahmad M. Bin Shafar, CEO) for sponsoring the effort. Special thanks also go to W. Stephen Comstock of ASHRAE for getting all the parties together and setting the project in motion, and to all of the ASHRAE staff involved in the editing and publication process. Our team of four well-qualified reviewers kept us on track and improved the publication in many ways with their input, including reviewer Hassan Younes of Griffin Project Development Consultants, who also contributed a case study. I would also like to express my gratitude to my team members who added content and expertise in their respective areas of specialty.

And finally, thanks to my loving wife Karen Phetteplace, who not only put up with my long and unscripted hours in my office, but also proofread the entire document.

Gary Phetteplace
May 2019

# 1
# To the Building Owner

## INTRODUCTION

District cooling (DC) is the practice of meeting the cooling needs of buildings (i.e., air-conditioning) from a central plant located off the building site. Normally the needed cooling effect is delivered in the form of chilled water that is then used in the building air-conditioning system, in the same way that chilled water created on site by a chiller plant would be. In the process of cooling the building, the chilled water is warmed and then returned to the off-site central plant to be chilled and delivered again. The delivery of the building cooling by this method becomes a utility service not unlike potable water supply and sewage collection. There are many advantages to the building owner by meeting his or her building's cooling needs with DC. Most benefits are quantitative, i.e., a value can be placed upon them, but some qualitative benefits, for which it can be difficult to provide a cost benefit, can be priceless to building owners. From a cost perspective, the quantitative benefits can be broken down into the familiar categories of capital costs and operating costs.

## CAPITAL COST ADVANTAGES

The most significant capital cost saving when DC is chosen is the elimination of all equipment in the building or otherwise on the site that would generate the chilled water. This includes not only the chillers themselves but also the cooling towers, pumps, and water-treatment systems. The space that they would normally occupy can be used for other uses or repurposed in the case of buildings with existing chillers.

In-building chiller plants require significant electrical power, and to provide it, the servicing utility will likely impose additional charges on the building owner. In addition, there will be electrical wiring and switchgear, required for the chillers, pumps, and cooling tower fans, whose cost can be eliminated when DC is chosen to serve the building.

For buildings that have critical cooling needs, such as data centers or medical facilities, redundancy of chilled-water generating equipment will be required, further increasing the expense and footprint of the equipment in the building or on site. When DC is chosen, all of the required redundancy is provided by the district cooling service provider, who is able to do this at much lower cost than a building owner, as their central plants will have multiple chillers, cooling towers, pumps etc., to meet the varying demand of the district cooling system's aggregated customer base, thus yielding very high levels of reliability in the supplying of chilled water. District cooling systems (DCSs) provide unparalleled reliability of supply, owing to equipment redundancy and the high level of operational supervision and maintenance that is prevalent in a DCS. On average, the International District Energy Association (IDEA) (2008) reports that DCSs have reliability/availability exceeding 99.94%.

During the life cycle of a building it will normally be necessary to replace equipment that has been provided to generate chilled water on site. The major items will be the chillers, cooling towers, and pumps. The life of this equipment can be less certain, especially when maintenance practices are not optimal, and significant costs may occur that cannot be scheduled with any degree of certainty. With DC, future costs can be contractually set, taking most of the uncertainty out of the significant building operating expense represented by meeting cooling needs.

## OPERATING COST ADVANTAGES

The largest operating cost advantage of subscribing to a DC service will normally be the elimination of the electricity costs for operating the in-building chillers, pumps, and cooling tower fans. These costs are in effect shifted to the DCS operator. The difference for the DCS operator, however, pertain to economies of scale, increased equipment efficiency for larger equipment, and being able to match load closely while operating equipment near its point of highest efficiency.

Second only to energy cost savings will normally be the savings from not having to provide qualified building operation and maintenance staff to cover the building's chilled-water plant equipment. This includes not having to deal with additional employee costs such as workplace safety and training, which for an in-building chiller plant may be entirely foreign to the building owner's primary employee competencies and requirements. It will still be necessary for the building owner to maintain their side of the HVAC system on an ongoing basis. This will include diligent inspection, maintenance and housekeeping, primarily of air-side heat transfer surfaces (coil fins, reheat coils if present, etc.), and upkeep of HVAC filters.

There will also be a number of other lesser but still significant cost savings from subscribing to a DC service. These include elimination of cooling tower upkeep costs, including both the water commodity required for continuing makeup due to evaporation, as well as the chemicals necessary to treat this water. Also avoided is the requirement to keep spare parts on hand, and as well as the need to insure the plant and equipment.

As with capital costs, subscribing to a DC service eliminates many of the uncertainties that a building owner is faced with when cooling requirements are produced on-site. Both capital costs and operating costs are replaced with an established contract cost. The contract is typically made with an experienced provider of cooling services who has production redundancy and a base of assembled customers. Relative to the typical building owner, an established district cooling provider is likely to have the required resiliency and flexibility to continue reliable service in the face of a number of upset scenarios and contingencies.

## LIFE-CYCLE COST EXAMPLE

A detailed cost comparison between on-site chilled-water generation and subscription to a DC chilled-water service will have all of the previously mentioned factors, as well as others, and it must be done on a life-cycle cost basis. A detailed example has been included in the companion document to this guide, *District Cooling Guide*, Second Edition (Phetteplace et al. 2019), as Example 2.1 and will not be repeated here. It is instructive, however, to examine the relative costs to the building owner for on-site generation of chilled water from that example shown in Figure 1.1.

While the individual costs, and thus their relative magnitude, will vary from one case to another, these relative costs are felt to be representative of a typical case, more details are provided in Phetteplace et al. (2019). For that particular example, the results found that subscribing to the district cooling service was lower both in initial capital outlay and life cycle costs for the building owner. The complexity and accompanying uncertainty of considering all the cost components shown by Figure 1.1 is balanced against the simplicity and predictability of the costs associated with subscribing to a district cooling service that will normally just have three major components: an energy charge, a capacity charge, and perhaps a fixed service charge. There may be other charges associated with a low return water temperature (known as low $\Delta T$), but these are entirely avoidable with proper design of the building as discussed in Chapter 2.

## INTANGIBLE BENEFITS OF DISTRICT COOLING

The use of DC as opposed to on-site generation of chilled water carries advantages to which a definitive cost is difficult or impossible to assign. These costs, or conversely benefits, may accrue to the building owner and often society at large. Many of these benefits are discussed below.

### Qualitative Benefits

Often, a cost cannot be assigned to a DC benefit. One of the best examples of a qualitative benefit of connecting to a district cooling plant (DCP) is that the building owner may now make use of valuable space within the building that was typically reserved for

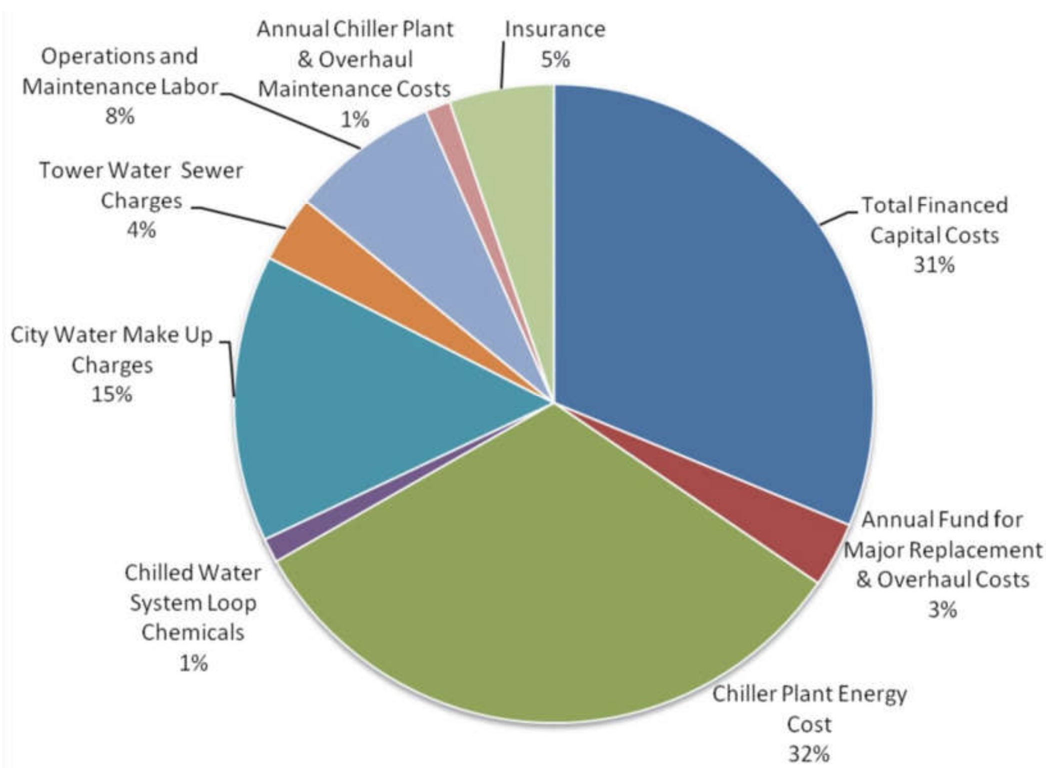



**Figure 1.1** Relative magnitude of the costs for on-site generation of chilled water. (Phetteplace et al. 2019, Example 2.1)

cooling equipment. Several decades ago, the normal design for building towers in the Middle East was to have a parking structure with an air-cooled chiller plant on its roof behind the tower. With the popularity of DC, the rooftop of the parking ramps and even the towers can now be landscaped with gardens, swimming pools, and restaurants taking advantage of this highly valuable space. This space can now serve a higher use and make money for the building owner as a premium space. Similarly, in the US, high-rise buildings typically had the chiller plants located in the penthouse on the highest floor close to the roof-mounted cooling towers. Once again, when connecting to DC, the uppermost floor of the tower was most desirable and could charge the highest rent or sales fees. Recouping these spaces adds revenue to the owner's investment.

Other qualitative benefits include not having to deal with the maintenance headaches of equipment that is run hard during a Middle Eastern summer. Typically, a critical piece of gear will breakdown when you need it the most and when it is working its hardest. Having on-demand DC service alleviates this. There is a certain "sleep better at night factor" with DC, knowing that it is always on, operated by experienced staff, and typically maintained to a higher standard than other cooling plants.

## Refrigerants

Central to the production of cooling will be some form of refrigeration process to extract heat from the working fluid (chilled water), which is distributed to provide cooling to the end user. In most cases this will be the predominant process in use today; mechanical compression refrigeration. From an engineering standpoint, this is not a terribly complicated process and is among the most fundamental processes a student of thermodynamics will learn. Most building owners will not need to understand the details of this process, but there are many references available for those who are interested—for example, Tomczyk et al. (2017). What is important for the building owner to understand is that this process is fundamentally the same for a refrigerator, window air-conditioner, or a large water chiller. Each of these pieces of equipment requires a working fluid called the refrigerant. Early in the history of refrigeration these working fluids were naturally occurring compounds such as ammonia, and in certain specialized applications, these remain in use today. To better serve a wider range of applications efficiently and safely, many chemical compounds were developed to serve as refrigerants.

Many of these compounds were used for years as refrigerants, and some had other purposes such as blowing agents and cleaners. Eventually it was learned that some of these compounds had deleterious effects on the environment, including greenhouse gas characteristics that have been correlated to environmental issues such as depletion of the ozone layer and global warming. This led to efforts to find alternatives, and while many of those that were relatively easily identified and put into use, they still have impacts on the environment, albeit to a lesser extent. As concern has continued to grow about climate change, momentum has built for even greater greenhouse gas (GHG) mitigation efforts. This has resulted in a web of regulatory requirements/agreements, many of which have been developed by global bodies, e.g., The Montreal Protocol. Few will have not been touched in some way in their day-to-day lives by these regulations/policies already, yet it is important to understand that ever more stringent measures will be imposed going forward to address the climate change issue.

Given that many refrigerants still in use today are identified as contributing to GHG accumulation in the atmosphere, this results in uncertainty from the perspective of the building owner. There are potential business risks associated with the permissibility, and even availability, of these substances for use, as these refrigerants are in many cases being phased out in the near term and going forward. The subscriber to a DC service is not entirely isolated from these impacts. However, as a dedicated specialist, it is the responsibility of the DC service provider to continually monitor and remain cognizant of the ever-changing regulatory situation, and to plan, invest, and respond as necessary. This is an enormous advantage to the building owners who use DC, freeing them to focus on their core business rather than the complications of evolving global refrigerant policies and challenges. The DC provider is much better equipped to handle these challenges as refrigerant management is central to their business. Furthermore, the DC provider's core competence in this arena imparts advantages of scale and expertise not generally available to the typical building owner. For more information on the current state of refrigerants and their global regulation, see https://wedocs.unep.org/bitstream/handle/20.500.11822/26589/HFC_Phasedown_EN.pdf?sequence=1&isAllowed=y.

## Environmental

Apart from the climate change issue, there are other environmental advantages to using DC. Assembled buildings with diverse loads mean that generating chilled water in a central plant is normally more efficient than using in-building equipment. Operating equipment at optimized efficiency means that environmental impacts can be reduced. Greater efficiencies are also achieved due to the larger, more efficient equipment and the ability to stage that equipment to closely match the load, yet remain within the equipment's range of highest efficiency.

DCSs may also take advantage of technologies that are often out of reach to on-site installations. For example, thermal energy storage (TES), which is often included in DCSs but not economically viable on-site, allows reduced electric demand charges associated with the increased refrigeration capacity required on-peak times by shifting cooling production to off-peak times. This is especially important to cope with the extra capacity needed, for example, at incidences of high pollen count, poor outdoor air quality, and recurrent seasonal sandstorms associated with urban communities in the Middle East. The marked increase in respiratory difficulties associated with these conditions for the elderly, newly born babies in hospitals, and those suffering with acute reactions to increased pollution levels and at ozone alerts time, can then be alleviated without recourse to high electric demand charges. TES also allows DCSs to take advantage of free or water-side economizer cooling, a process by which a plant can configure its cooling tower to produce cold water using cooler ambient temperature conditions during the winter and shoulder-season periods. Free cooling can greatly reduce a plant's electric demand assuming the ambient conditions are compatible. For electric-driven DCPs, higher efficiency becomes the central environmental benefit since in-building plants are normally electric driven as well.

DC also often affords the opportunity for other environmental benefits. One example is the ability to use treated sewage effluent as cooling tower makeup water—an option that would not be feasible for in-building systems. In certain settings, the ability to use seawater or fresh surface water for heat rejection directly can be deployed—again, an option likely not feasible on a single building scale. Furthermore, in some instances, deep sea water or deep fresh surface water may be cold enough to provide cooling directly without the use of refrigeration equipment, greatly reducing electric demand and consumption. For example, deep sea cooling is possible with water sourced at depths of 1000 to 2000 ft (300 to 600 m) in northern and southern latitudes, and 2000 to 3300 ft (600 to 1000 m) in subtropical latitudes, as at these depths temperatures of 44°F (4°C) are normally available. These conditions exist in coastal areas of the Middle East on the Arabian/Indian sea, the Red Sea, and the Mediterranean, but not the Gulf, where depths are only 300 to 400 ft (90 to 120 m).

Environmental benefits will normally also be realized due to the increased efficiency that comes with vigilant maintenance and operational optimization realized when the cooling source is operated as a business itself, rather than an often neglected and minor ancillary activity or cost center serving an entirely unrelated core business.

Due to its many environmental advantages, the United Nations Environment Programme (UNEP 2015) has endorsed district energy (district heating and district cooling) for its potential role in achieving energy efficiency and renewable energy use in cities. In some cases, DC may even be a primary means to achieve Green Building status or other environmental incentives not otherwise feasible. These opportunities especially present themselves when a DC utility employs environmentally desirable practices such as combined heat and power or renewable energy in generating cooling, or as previously mentioned, using treated sewage effluent in cooling towers.

## Alternate Energy Sources

Large central plants, among them some that also produce district heating, can in select situations, generate cooling from a heat source by a process known as absorption refrigeration. Waste heat from industrial processes or electric power generation along with geothermal energy are potential energy sources for heat-driven refrigeration. These heat sources are much easier to use on the scale of a DCS as opposed to smaller, in-building equipment. Furthermore, large-scale mechanical compression refrigeration may be powered not only by electricity as discussed above, but also may be directly driven by engines or turbines. When fuels are burned to generate cooling via absorption or gas/steam turbine and/or engine-driven chillers, emissions from central plants are more capable of mitigation than those from individual plants, and in the aggregate generate less pollutants due to higher quality of equipment, higher seasonal efficiencies, improved load factor/demand diversity, and higher level of maintenance. A central plant that burns high-sulfur coal can economically remove noxious sulfur emissions, where individual combustors could not. Similarly, the thermal energy from municipal wastes can be used by a DCS as an energy source in an environmentally sound matter, an option not likely to be available on a building-scale system.

## Planning

DC provides far more flexibility when compared to the potentially sunk cost associated with an in-house chilling plant. While it is necessary to accurately determine and contractually specify the load initially when DC is adopted, at a later date it is possible to either increase or decrease that load at far less of a cost and efficiency penalty than would occur with an in-building cooling plant. While it is always desirable to have future changes in load known, it is not always possible to do so. Because it aggregates customers with varying load profiles, DC, is more capable of adjusting and adapting to mismatches between projected and actual loads. Furthermore, on the commercial side, there are contractual means by which both the building owner and the DC provider can manage their respective risks should the load change.

Notwithstanding the planning flexibility noted above, the need to accurately determine the contacted load requires special emphasis. In the design of buildings that are not served by a DCS large safety factors, or gross oversizing for other reasons, will generate increases in first costs. For buildings not served by a DCS, oversizing often increases operating costs as well; however, those increases will normally only be marginal (but not always). Neither of these impacts is desirable for the building owner, of course, but they are nevertheless common place and often go unnoticed because there is no metering of the actual loads experienced once the building is operation and occupied. Buildings served by DC providers are metered so actual loads are known; DC service providers and their consultants report finding that *AC loads are often overstated by four to five times the actual maximum load* the buildings experience. In design of a building that will be served by a DCS, gross overstatement of the loads will have two significant consequences. First, the contractual capacity with the DC service provider will be higher than necessary resulting in higher charges as explained in the section below on DC tariff structures. The second impact will be the potential that the building could operate under a condition called low $\Delta T$ that may result in additional charges to the building owner, again this is discussed in the section below on DC tariff structures. Both of these costs, as well as the additional capital cost of the in-building equipment on the building owner's side, can be easily avoided up front with a proper "right-sized" design, additional details for the designer are contained in Chapter 2 and Phetteplace et al. (2019). For existing buildings, Chapter 3 contains information that will help reduce additional charges that may stem from low $\Delta T$.

### Noise, Vibration, and Aesthetics

A building that is connected to DC can be isolated from all the noise and vibration that normally comes from an in-building cooling plant due to the refrigeration equipment, cooling towers, and or air-cooled condensers. There also will be no unsightly cooling towers or air-cooled equipment, even on the roofs.

### Waste Streams and Cooling Tower Drift

In-building cooling plants that are water cooled will normally use cooling towers. Cooling towers produce a waste stream of water that is known as blowdown that is high in dissolved solids and possibly suspended solids and biological agents as well. This water must be disposed of, normally to sewer systems and its discharge is normally regulated. Chemicals will have been added to the cooling tower water to help control its chemistry and biological activity (i.e. the potential for Legionella for example) and thus its disposal is often controlled. Proper water treatment is of critical importance, and failure in this regard can have dire consequences. Improperly treated cooling tower water under certain conditions can result in a dangerous legionella outbreak. Apart from that, poor water treatment can also prematurely age and cause significant damage to the most costly and important elements of cooling equipment. These are a few of the pitfalls associated with major maintenance responsibilities that come with an in-building cooling plant. District cooling can alleviate these entirely for the building owner.

### High Reliability

DCSs provide unparalleled reliability of supply owing to equipment redundancy and the high level of operational supervision and maintenance that is prevalent in district cooling systems. As stated earlier, on average IDEA (2008) reports that district cooling systems have reliability exceeding 99.94%.

## DISTRICT COOLING TARIFF STRUCTURES

The DC provider will normally have a tariff structure largely analogous to an electric or gas utility with these major components:

- *Consumption charge*. This charge will represent the actual cooling used by the building over the billing period and will thus vary with the change in seasons, occupancy, building envelope improvements, etc. The consumption charge will normally be measured in kWhs, Btus, or ton-hrs.[1]
- *Demand charge*. This charge represents the commitment of chiller plant and distribution system capacity that the district cooling provider has made to the building

1. A ton of refrigeration (TR) is a customary unit of refrigeration capacity dating from when blocks of ice were used for refrigeration. It represents the amount of refrigeration capacity required to make a ton of ice from unfrozen water (at 32°F [0°C]) in 24 hours; it is equivalent to 12,000 Btu/h. The unit ton-hrs is a unit of refrigeration energy supplied/used.

owner. The greater the capacity committed, the greater the charge will be. The building owner is cautioned that it is cost effective to have his/her engineer accurately determine loads and not overspecify the needed capacity as is often done with in-building cooling equipment (see Chapter 2). The demand charge will normally be measured in kW, Btu/h, or tons of refrigeration (TR).

- *Connection charge*. A connection charge will be one-time charges made by the DC provider to help offset the costs of serving the building. This is normally an initial cost, but could be amortized over some period. It may be associated also with investment in a meter or other equipment the provider installs on the customer site.
- *Penalty charge*. A penalty charge may be assessed by a DC provider if the chilled water is not used efficiently by the building and is returned from the building at a low temperature. This condition is referred to as low delta T or low $\Delta T$, and is discussed in the section below. What the building owner must know is that this condition is entirely avoidable for a building properly designed for DC service, as explained for the building designer in Chapter 2. The return temperature penalty would normally be assessed on the average of the difference between the contracted return temperature and the measured return temperature. Normally the average would be weighted for the measured flow rate.
- *Compliance bonus*. To further emphasize the importance of avoiding low $\Delta T$, some DC providers may use tariff structures to incentivize obtaining design $\Delta T$ with a credit or rebate of some percentage of each bill. The rebate percentage may even be further increased for exceeding design $\Delta T$. A small range of low $\Delta T$s slightly below the design $\Delta T$, but before penalties are imposed, may be used with such incentive provisions. Thus, creating a progressive scale that also provides a small range of leniency.

These are the typical components of most DC providers' tariff structures. There may also be other terms agreed to contractually between the building owner and district cooling provider such as provisions for anticipated load growth or reduction. Other unique provisions that might be provided for in a contract between the DC provider and the building owner might include:

- Assistance from the DC provider to the building owner in the form of design review or even design services for initial design or post-construction problem resolution.
- Provisions within the contract for conditions and terms under which contracted capacity may be adjusted either up or down.
- Contract provisions that will allow the building owner to make energy conservation efforts without penalties from the DC provider that would absorb all the saving, perhaps means by which the savings and/or incentives may be shared.

There may also be surcharges/rebates that are based on current source energy and/or water prices, for example. Interruptible rates that would allow the DC provider to curtail or limit the supply of chilled water might be an alternative where the building use would allow for it. Further, if a district cooling provider becomes capacity constrained for some reason, or just wants to reduce demand during peak periods, they might offer the building owner an incentive to reduce his/her load by building energy conservation retrofits, for example.

Moreover, when a contract is received by the building owner from a DCP, it is recommended to conduct a best-value analysis in lieu of only looking at costs looking at all the benefits whether quantifiable or not. Best-value analyses account for both technical and financial advantages and disadvantages of DC compared to on-site building cooling systems and certain nonquantifiable parameters may have increased value or worth more than others.

## LOW CHILLED-WATER RETURN TEMPERATURE, OR LOW DELTA *T*

As noted above, if the chilled water is not used efficiently by the building and is returned from the building at a temperature that is lower than optimal, a condition referred to as low delta $T$ (low $\Delta T$) is said to exist. In the district cooling industry, it has even been elevated to a status of a "syndrome". Furthermore, this condition is entirely preventable with proper design of the building and continued operation and maintenance in keeping with design intent. A simple example best illustrates why it is so important to a DCS operator to have the connected buildings achieve an optimized delta $T$ ($\Delta T$). An average office building will have an approximate load density of 280 ft$^2$/ton (7.4 m$^2$/kW). (See Table 2.1 in Phetteplace et al. [2019] for this and other typical load density values). A DCS with properly designed buildings will have a $\Delta T$ of 16°F (8.9°C) per Phetteplace et al. (2019). Many buildings operate at $\Delta T$s of much less than that, many not even half that temperature difference, but for the purposes of this example we will assume a $\Delta T$ of one half of the desired value, or 8°F (4.4°C). Figure 1.1 shows how much water the DC provider would have to deliver to the building of approximately 112,000 ft$^2$ (10, 400 m$^2$) for each day at the design loading both with a normal $\Delta T$ of 16°F (8.9°C) and a low $\Delta T$ of 8°F (4.4°C). The amount of chilled water needed is doubled. This is no surprise as it's inversely proportional to $\Delta T$, but what is eye opening is the amount of water—1,726,281 gallons for the case of the building with low $\Delta T$. This is 863,140 gallons per day more than the building that is properly designed would need.

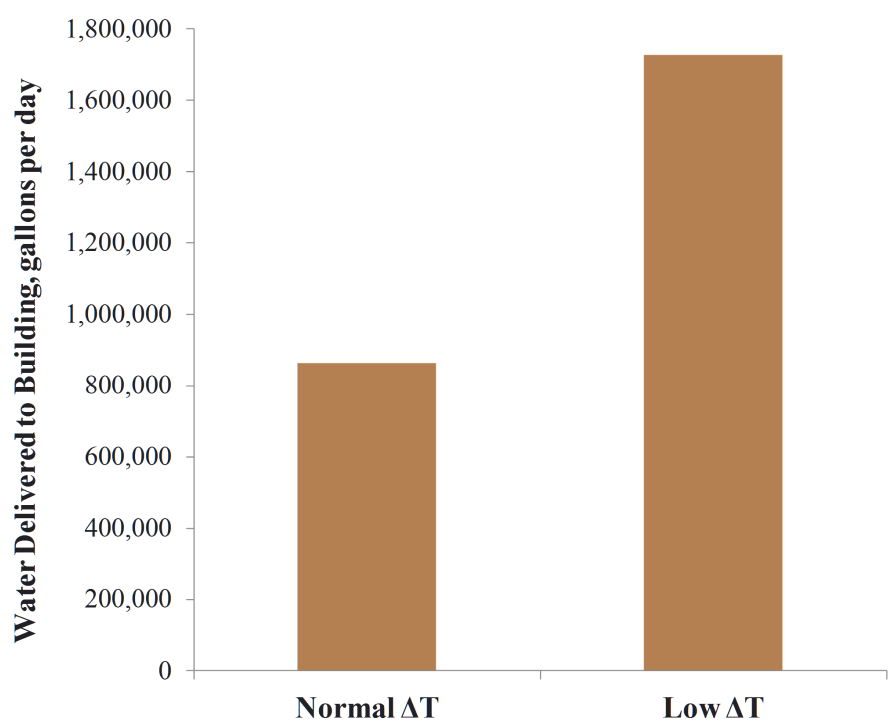



**Figure 1.2** Impact of low $\Delta T$ on amount of chilled water delivered to customer.

Of course, this extra water is not lost; it is returned to the plant for recirculation, but the pumping energy associated with that water is lost. Worse yet, the pumping energy turns up as heat added to the chilled water that must be removed by the chiller plant and makes the overall cost of supplying chilled water to the building owner go up. Low $\Delta T$ can be likened to a grossly leaking water faucet in a building. The water is delivered to the building and then is discharged to the sewer unused, consuming energy and resources just in the delivery but also diluting the sewage, which just makes it more expensive to process. Similarly, the same is true for the chilled water returned from a building with low-$\Delta T$ syndrome; to chill the low temperature return water back to supply chilled water temperature is less efficient because its temperature is low and the chillers use more energy when operating at lower temperatures. To these effects, others must be added. In summary, the impacts include:

- extra pumping energy,
- removing the extra pumping energy, i.e., heat, from the chilled water,
- chiller efficiency degradation at lower average operating temperature,
- increased energy consumption by the condenser water pump and the cooling tower fans,
- increased heat gain to the chilled-water distribution system due to the lower chilled-water return temperature, and
- increased usage of makeup water due to more towers being energized.

For our case of low $\Delta T$ described above, Appendix A contains an analysis of the consequences of all these on the net energy which must be consumed at the chiller plant to supply the same amount of cooling to a consumer operating at low $\Delta T$, the net impact is a 143% increase. Figure 1.3 provides the results of the analysis. Obviously, this result is specific to the

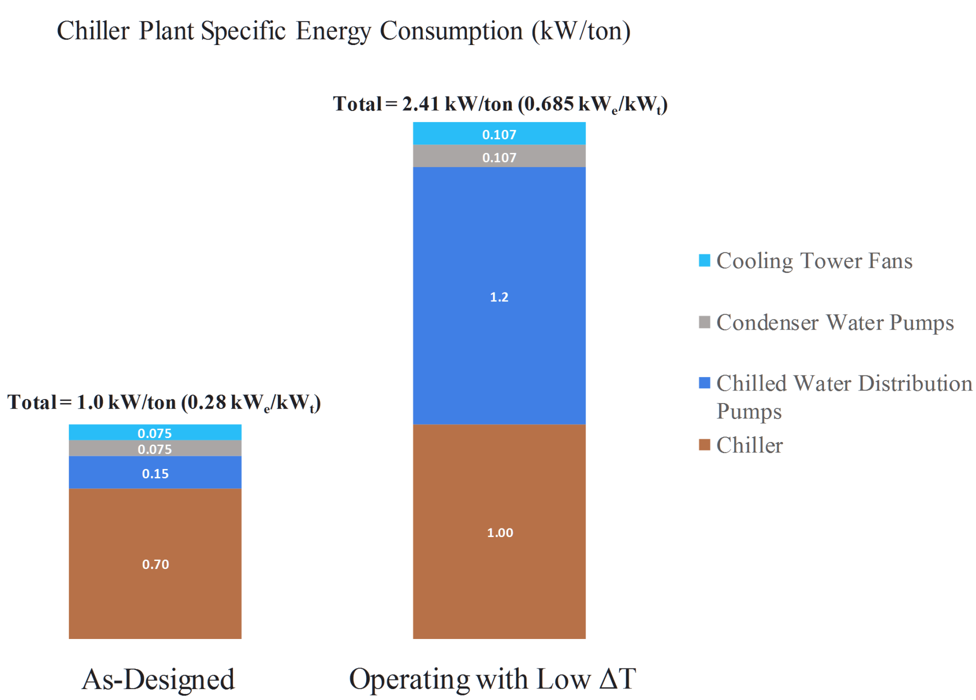



**Figure 1.3** Impact of 8°F versus 16°F (4.4°C versus 8.9°C) $\Delta T$ on chiller plant energy consumption. (Note: impacts of distribution system heat gains are not included in this figure; see Appendix A for those impacts.)

assumptions of this example case, all of which have been provided in Appendix A. The assumptions we have made are, we feel, representative for the conditions noted. Many of these values can vary widely across DCSs; however, the methodology that has been laid out could easily be used for a DCS operating under much different conditions.

## SELECTING THE DESIGN FIRM FOR A BUILDING TO BE SERVED BY DISTRICT COOLING

Proper design of the in-building HVAC equipment will determine the building operating $\Delta T$; if low $\Delta T$ and the disastrous effects discussed earlier occur, the responsibility will likely fall on the building owner and his/her designer. Thus, a note of caution is required here with regard to retaining an engineering firm to design, supervise construction, and commission the in-building equipment for a building served by a DCS. While the engineering for DCS service is not more difficult than many other similar design tasks, there are unique aspects that only an experienced designer of buildings served by a DCS will know. Use of this guide as well as the companion document (Phetteplace et al. 2019) will help acquaint a designer new to buildings served by DCSs; however, there is no substitute for experience. The presumption should not be made that a firm with broad experience will necessarily have experience in design for DCS service; in fact, the assumption should always be that the firm has no specific experience unless shown otherwise. A detailed discussion of how to select an engineering firm for a combined heat and power systems can be found in Meckler and Hyman (2010), which can serve as a guide for selecting a design firm for a building served by a DCS.

## REFERENCES

IDEA. 2008. *District cooling best practices guide*. Westborough, MA: International District Energy Association.

Meckler, M., and L. Hyman. 2010. *Sustainable on-site CHP systems, design, construction, and operations*. New York: McGraw Hill.

Phetteplace, G., S. Abdullah, J. Andrepont, D. Bahnfleth, A. Ghani, B. Kirk, V. Meyer, and S. Tredinnick. 2019. *District cooling guide*, 2d ed. Atlanta: ASHRAE.

Tomczyk, J., E. Silberstein, W. Whitman, and W. Johnson. 2017. *Refrigeration and air conditioning technology,* 8th ed. Boston: Cengage Learning.

UNEP. 2015. *District energy in cities: Unlocking the potential for energy efficiency and renewable energy*. Paris: United Nations Environmental Programme.

# 2
# To the Building Designer

The chilled-water energy produced at the central plant is transported via the distribution network and is finally transferred to the connected buildings or to consumers. The district cooling (DC) interconnection to the building has been called many different terms, such as the energy transfer station (ETS), energy substation, the end user interface, or a customer/consumer interface. However, the purpose is the same—to transfer energy and custody of the chilled water from the provider to the customer. The consumer interconnection to the system is a critical aspect of DC that has impacts not only at the consumer building but also far beyond.

The design of the HVAC system for a “DC Ready” or DC-connected building is not entirely unlike the design of a building having its own cooling plant. However, there are a few critical aspects of the design to which special attention must be paid in order to satisfy the building occupants’ comfort conditions in a cost-effective way. As noted in the previous chapter, the paramount concern is to avoid the situation where the chilled water is not used efficiently by the building and is returned from the building at a low temperature. This condition is referred to as low delta-T ($\Delta T$) syndrome and will result in inefficiencies at the DC central plant, in the distribution of chilled water, and likely additional costs for the building owner. (See Chapter 1 for more discussion on this topic.) Fortunately, low-$\Delta T$ syndrome is preventable with proper design of the building HVAC systems and continued operation following the design intent along with proper maintenance practices.

For buildings serviced by DC, the need to accurately determine the contacted load requires special emphasis. In the design of buildings that are not served by a DCS, large safety factors, or gross oversizing for other reasons, will generate increases in first costs. For buildings not served by a DCS, oversizing often increases operating costs as well; however, those increases will normally only be marginal (but not always). Neither of these impacts is desirable of course, but they are nevertheless commonplace and often go unnoticed because there is no metering of the actual loads experienced once the building is operational and occupied. Buildings served by DC service providers are metered so actual loads are known, and DC providers and their consultants report finding that overstatement of AC loads can be four to five times the actual maximum load the building experiences. In the design of a building that will be served by a DCS, gross overstatement of the loads will have two significant consequences. First, the contractual capacity with the DC service provider will be higher than necessary resulting in higher charges as explained in the "District Cooling Tariff Structures" section in Chapter 1. The second impact will be the potential that the building could operate under a condition called low $\Delta T$ that may result in additional charges to the building owner; again, this is discussed in the section on DC tariff structures in Chapter 1. Both of these costs, as well as the additional capital cost of the in-building equipment on the building owner's side, can be easily avoided up front with a "right-sized" design. Design loads should be properly established using methods such as those provided in ASHRAE (2017b) or ASHRAE (2014). For existing buildings, Chapter 3 contains information that will help reduce additional charges that may stem from low $\Delta T$.

The vast majority of this chapter has been taken from the comprehensive guide to the design, operations, and maintenance of DC systems, *District Cooling Guide*, Second Edition (Phetteplace et al. 2019). However, significant new information is provided in this owner's guide as it pertains to who typically provides what equipment in the building interconnection, as well as the typical division between the parties of other responsibilities. In addition, the added information also lays out the details of what those responsibilities are. If the designer of the building systems will in fact be responsible for other aspects of a DCS design, they are encouraged to consult Phetteplace et al. (2019) as well as the other comprehensive documents on the topic, notably IDEA (2008) and the references listed at the end of this chapter.

## DEFINITION OF RESPONSIBILITIES AND BUILDING REQUIREMENTS

While not a complex design issue, there are several major components of the building interconnection that are furnished by parties other than the entity who eventually installs and maintains the equipment. The following table of responsibilities (Table 2.1) has been developed based on experience and standard practice. Each DC provider typically has their own ETS connection manual or owner's requirements for district-energy-ready buildings that should be followed. Actual responsibilities may differ based on specific DC provider contract requirements. Primary side is defined as the district energy provider side and secondary side is the building or customer side of the connection. While the building owner or customer provides the majority of the equipment on the secondary side, the DC provider typically desires to review and comment on the selections to confirm their guidelines are followed.

The main topics listed above are covered in more detail in the following subsections to provide a better understanding as to what is required and expected for the connection to a DCS.

### Energy Transfer Station Room/Machinery Space

The ETS room is intended to house all the equipment necessary to provide a complete and efficient interconnection. ETS rooms should be adequate in size for the proper servicing of all equipment. They should be located so as to provide access for replacement of all mechanical equipment, should be large enough to provide adequate space for spare parts storage, and should be located on the lower levels of the customer's building.

ETS rooms shall be accessible by a standard stair or elevator. Ship's ladders and steep stairs are NOT acceptable as maintenance carts and tool boxes are routinely transported through the space. Single doors shall be a minimum of 36 in. (0.9 m) wide. Adjoining pieces of equipment shall be separated by a minimum of 36 in. (0.9 m). The room shall be locked and only accessible to authorized DC provider personnel.

The design should provide space within all mechanical or ETS rooms to store two changes of air filters, lubricants, etc. Adequate clearances to be provided to customer equipment as well as minimum 80 in. (2 m) clearance under piping and equipment. Ser-

**Table 2.1** Building Interconnection Responsibility Matrix

| | Description | Furnished Party | Installation Party | Operated and Maintained by |
|---|---|---|---|---|
| ETS Room | Lighting and convenience receptacles | Customer | Customer | Customer |
| | Power to control panel | Customer | Customer | Customer |
| | Ventilation and air-conditioning of ETS Room | Customer | Customer | Customer |
| | Fire detection and protection | Customer | Customer | Customer |
| | Plumbing—floor drains, potable water supply | Customer | Customer | Customer |
| | Secure and adequate space for ETS equipment | Customer | Customer | Customer |
| | Adequate clearances for maintenance and equipment egress | Customer | Customer | Customer |
| ETS Equipment | Heat exchangers (quantity and size) | DC provider | Customer | DC provider |
| | Building chilled-water pumps | Customer | Customer | Customer |
| | Expansion tank and air separator | Customer | Customer | Customer |
| | Makeup water | Customer | Customer | Customer |
| | Chemical treatment | Customer | Customer | Customer |
| | Primary side isolation valves | DC provider | DC provider | DC provider |
| | Secondary side isolation valves | Customer | Customer | Customer |
| | Primary side piping, valves, fittings, vents, drains, and insulation | DC provider | DC provider | DC provider |
| | Primary and secondary side pressure relief valves | DC provider | DC provider | DC provider |
| | Secondary side piping and fittings | Customer | Customer | Customer |
| | Temperature and pressure gauges | Customer | Customer | Customer |
| | Secondary side strainers | Customer | Customer | Customer |
| | Secondary side circulation pumps, valves, fittings, vents, drains, and insulation | Customer | Customer | Customer |
| Controls and Instrumentation | Primary side temperature transmitters | DC provider | DC provider | DC provider |
| | Flowmeter | DC provider | DC provider | DC provider |
| | Primary side pressure transmitters | DC provider | DC provider | DC provider |
| | Delta T Control Valve | DC provider | DC provider | DC provider |
| | ETS Controls (PLC, UPS, Network Communications, etc.) | DC provider | DC provider | DC provider |
| | Secondary Side Sub-Metering | Customer | Customer | Customer |
| | Secondary side Controls | Customer | Customer | Customer |
| Activities | Hydrostatic Testing, cleaning and flushing of Primary and Secondary Side | Customer | Customer | Customer |
| | Review of Customer ETS Design | DC provider | Customer | N/A |
| | Maximize the chilled water return temperature | Customer | Customer | Customer |
| | Commissioning of ETS Equipment | DC provider | DC provider | DC provider |

vice access should comply with local codes and manufacturer's recommendations and should be reasonably planned for human access.

ETS rooms located above occupied floor levels are recommended to be curbed, waterproof, sealed floors, and have all floor penetrations sleeved to 2 in. (50 mm) above the floor to prevent liquid spills and leaks from traveling out of the space. ETS rooms should be well lit, maintaining a minimum of 25 foot-candles. Lighting shall be switched at each exit. Consider adding some lighting on the emergency generator power source, and provide low voltage convenience outlets in mechanical rooms to provide for ready servicing of equipment.

Air-conditioning or ventilation should be provided to comply with energy codes. An adequate number of floor drains should be provided. Locate drains to avoid running of condensate drains and other similar equipment across floors. Provide trap primers as required per code. All floor-mounted equipment should be mounted on concrete housekeeping pads. If the floor is not slab on grade, provide appropriate vibration isolation bases and inertia bases and flexible pipe connections and spring hangers for piping.

Typically, the number of heat exchangers is dictated by the size of the load and building type/function. Refer to the "Heat Exchanger" subsection later in this chapter. Furthermore, if the customer plans on submetering any part of the secondary system, the DC provider should review the metering and controls specification and follow the DC provider's published guidelines.

## Water Treatment, Pipe Testing, and Pipe Cleaning

The customer is typically responsible for the initial fill and future continual makeup of water for their system, including water needed for testing, cleaning, and flushing. The piping system should be hydraulic pressure tested at a pressure of minimum 1.5 times of the total working pressure and such pressure shall be maintained for a period of not less than 24 h without loss of pressure to ensure that the pipes are free from leaks. The results of all pressure tests should be recorded in test documents that are provided by the contractor/owners site representative and forwarded to the DC provider.

The customer should use the services of a water treatment subcontractor to provide the necessary chemicals, materials, and supervision for a complete cleaning and flushing of all piping from the point of delivery to the point of connection to the customer's chilled-water header. After satisfactory water quality analysis results have been obtained (according to DC provider's water treatment contractor), system start-up and commissioning may occur. A certification from the water treatment contractor will verify that the water quality is acceptable. Then, and only then, is it permitted to circulate water through the DC connection and ETS.

The customer should perform a complete and thorough flushing of the chilled-water pipework using cleaning and passivating chemicals and potable water as recommended by the water treatment subcontractor. Before connecting to DC water, temporary bypasses should be installed across the connection for the purpose of allowing the flushing water to recirculate in the customer's chilled-water pipework to reduce the water demand and wastewater discharge. The customer-side cleaning method, including flushing and chemical treatment details, shall be submitted by the approved consumer to the DC provider for comment and approval prior to commencement of each activity.

Clean equipment should be kept isolated from the portion of the system that remains to be cleaned and flushed. This is true for both direct and indirect connections to ensure that the district side is not contaminated by the customer's water or the heat exchanger is not plugged the first day of operation with customer-side debris.

The customer should also submit flushing, cleaning, and water treatment reports to the DC provider for comment and approval before the supply of DC water to ensure that the quality of chilled water used in the consumer's air-conditioning installation will not adversely affect the DC provider's equipment and the operation of the DCS.

Typical water quality parameters that are acceptable to both sides of the building interconnection are listed in Table 2.2, but they should be confirmed by the customer and the DC provider's chemical treatment specialist.

**Table 2.2** Chilled-Water Quality Chemical Parameters

| Parameter | Unit | Value |
|---|---|---|
| pH | — | 8.5 to 10.5 |
| Total Dissolved Solids | ppm | < 10 |
| Conductivity | ppm | < 3000 |
| Total Iron as Fe | ppm | < 1.0 |
| Nitrite as $NaNO_2$ | ppm | > 800 |
| Chlorides as $Cl^-$ | ppm | < 150 |
| Boron | ppm | 100 to 200 |
| Total Alkalinity as $CaCO_3$ | ppm | < 250 |
| Total Calcium Hardness as $CaCO_3$ | ppm | < 100 |
| Ammonia | ppm | < 20 |
| Soluble Copper | ppm | < 0.20 |
| Chemical Oxygen Demand | ppm | < 100 |
| Microbiological Limits: | | |
| • Total Aerobic Plate Count: | organisms/ml | < 1000 |
| • Total Anaerobic Plate Count | organisms/ml | < 100 |
| • Nitrate Reducers | organisms/ml | < 100 |
| • Sulfate Reducers | organisms/ml | < 0 |
| • Iron Bacteria | organisms/ml | <0 |

### Commissioning

Prior to the commissioning of the ETS, the building owner should pressure test, flush, and, when necessary, clean the building's internal existing chilled-water system as described above. During the commissioning process, the building operator will be responsible for the building's internal chilled-water system. The strainer on the customer's side will also be cleaned.

## TEMPERATURE DIFFERENTIAL CONTROL

The success of a DCS is often measured in terms of the temperature differential achieved between supply and return chilled-water temperatures at the central plant. Generally, maintaining a high temperature differential ($\Delta T$) between supply and return lines is the more cost-effective method because it allows smaller pipes to be used in the distribution system and will reduce the pumping energy consumed. Furthermore, as the example in Appendix A illustrates, it can increase the central plant efficiency significantly.

Largely, the consumer interconnection and the in-building equipment at the consumer's location determine the $\Delta T$ between supply and return CHW at that connection. Low $\Delta T$ is a chronic problem in DCSs and has earned the moniker "low $\Delta T$ syndrome." Low $\Delta T$ may be the Achilles' heel of a DCS that is otherwise well designed and operated. For this reason, the consumer's building HVAC equipment and the interconnection with the DCSs should be afforded much more attention in design than might initially be apparent. Successful operation of a DCS will normally require that the customer's $\Delta T$ be monitored, controlled, and optimized in most circumstances.

It's worth noting that low $\Delta T$ is also a problem in chilled-water systems for buildings that are not served by DCSs, however, the consequences are normally not nearly as great. This is due to the fact that these systems do not possess the extensive piping network and the attendant high pumping penalties of a DCS. Looking at the relative costs of the many elements of increased operating cost from low $\Delta T$ as shown in Figure 1.3, it's clear just how significant this single penalty is, and it must also be remembered that a significant portion of the other elevated costs comes about due to having to remove the excess heat input to the system by the additional pumping work; see Appendix A for the detailed example.

When DCSs are connected to existing buildings, the HVAC equipment within those buildings will often place severe limitations on the ability of the DCSs to achieve adequate $\Delta T$ at that connection while maintaining acceptable comfort within the building. Retrofit and potential optimization of building HVAC equipment to achieve acceptable $\Delta T$ may be expensive but should be fully considered in the context of the impacts on system $\Delta T$. The savings possible from increased $\Delta T$ must be weighed against any higher building conversion costs that may result. For DCSs operating as commercial ventures, where consumers are not able to achieve adequate $\Delta T$, tariff structures should be considered that adequately compensate the DC utility for the impacts imposed due to a given customer's inability to return water at the proper conditions.

When new buildings that will be connected to a DCS are constructed, it is easier to design the in-building HVAC equipment to achieve acceptable $\Delta T$s by following the guidance below. While the solution to achieving high $\Delta T$ is similar to an existing building connection, the additional costs for doing so in a new building will not be as onerous; however, it may be difficult to convince the building owner and/or designer to make the necessary modifications to the conventional design used with in-building CHW generation. *The DC utility/proponent must be prepared to make additional effort to convince others of the necessity of designing the building for DC service—that is the motivation for the development of this guide.*

To optimize the $\Delta T$ and meet the customer's varying chilled-water demand, the flow from the central plant should vary accordingly. Varying the flow also saves pump energy. Chilled-water flow on the customer's side must also be varied and will be discussed in the "Pumps and Pump Controls" section. Terminal units in the building connected to the chilled-water loop (i.e., air-handling units, fan-coils, etc.) may require modifications (e.g., changing three-way valves to two-way, increased coil size, etc.) to operate with variable water flow to ensure a maximum return-water temperature.

For cooling coils, six-row, 12 to 14 fins-per-in. (5 to 6 fins-per-cm) coils are the minimum size coil applied to central station air-handling units to provide adequate performance. With this type of coil, the $\Delta T$ should range from 12°F to 16°F (6.7°C to 8.9°C) or higher at full load. Coil performance at reduced loads should be considered as well where the fluid velocity in the coil tubes should remain high to stay in the turbulent flow range based on expected load turndown. To maintain a reasonable temperature differential at design conditions, fan coil units should be sized for an entering-water temperature several degrees above the main chilled-water plant supply temperature. Often, the most cost-effective retrofit is to replace the chilled-water control valve with a pressure independent control valve (PICV). PICVs are essentially two valves in one, combining a pressure-reducing device with a flow throttling valve. PICVs prohibit overflowing the coil based on higher differential pressures occurring in the system based on proximity of the coil to the distribution pumps and load conditions.

A summary of additional suggestions that will aid in achieving high $\Delta T$ is provided below. Some of these suggestions have been previously discussed and others will be discussed later in this chapter:

- Use variable district side flow.
- Use variable customer side flow.
- Provide cooling coils with a minimum of 6 rows and 8 to 10 fins-per-in. (3 to 4 fins-per-cm).

- Size fan coils to be 2°F to 3°F (1.1°C to 1.7°C) above the main chilled-water supply temperature.
- Eliminate three-way valves from building terminal equipment to maximum extent possible and use high quality two-way PICVs at air-handling units and fan coils.
- Use direct connections at the buildings where possible.
- Size heat exchangers for lowest economical approach temperatures (2°F [1.1°C]), where a customer requires low supply temperature.
- Where a chilled-water demand control valve is used, the valve should be a high quality (industrial) valve capable of control and positive shut-off under the highest expected pressures. The demand control valve should only be used only with either an indirect connection or customer pumps with a decoupler on a direct connection.
- On an ongoing basis, diligent inspection, maintenance, housekeeping (primarily of air-side heat transfer surfaces [coil fins; reheat coils (if present)]), and upkeep of HVAC filters will aid in maintaining proper return temperatures.

## CONNECTION TYPES

Chilled water may be used directly by the building HVAC system or process loads or indirectly via a heat exchanger that transfers energy from one media to another. When chilled water is used directly, it may be reduced in pressure commensurate with the buildings' systems. The design engineer must perform an analysis to determine which connection type is best. Table 2.3 outlines the relative merits of direct versus indirect

**Table 2.3** Relative Merits of Direct and Indirect Consumer Interconnection

| Issue | Direct Connection | Indirect Connection |
|---|---|---|
| Water quality | DCS water is exposed to a building system that may have lower levels of treatment and filtering. Components within existing building systems may have may have scale and corrosion. | Water quality of the DCS is isolated from building system and can be controlled. |
| Water consumption | Leakage and consumption of DCS water within the building may be difficult to control and correct. | Water leakage is within the control of the district cooling utility. |
| Contractual | Demarcation of consumer's building system may not be clear. | Clear delineation between the consumer and district cooling utility equipment. |
| Cost | Generally lower in overall cost due to the absence of a heat exchanger and possible deletion of building pumps and controls. | Higher cost due to a heat exchanger and additional controls. |
| Reliability | Failures within the building may cause problems or potentially even outages for the district cooling system. | The DCS is largely isolated of any problems in the building beyond the interconnection. |
| Pressure isolation | Building systems may need to be protected from higher pressure in a DCS or for tall buildings, a DCS may be subjected to higher pressures by the building system. | The heat exchanger provides isolation from building system pressure from the DCS pressure and each may operate at their preferred pressures without influence from the other. |
| $\Delta T$ | Potential for greater $\Delta T$ due to the absence of a heat exchanger. | Approach temperature in heat exchanger is a detriment to $\Delta T$. |
| In-building space requirements | Low space requirements. | Additional space required for a heat exchanger and controls. |

connections; additional details are contained in the proceeding subsections that are devoted to each type of connection, and an overview of various types of connections and their control is provided by Rishel (2007).

For commercially operated systems, a contract boundary or point of delivery divides responsibilities between the energy provider and the customer. This point can be at a piece of equipment, as in a heat exchanger with an indirect connection, or flanges as in a direct connection. A chemical treatment analysis must be performed (regardless of the type of connection) to determine the compatibility of each side of the system (district and consumer) prior to energizing. Cathodic isolating flanges should also be provided; normally these flanges would be located at the point where the CHW distribution system first enters the building. This is especially important where the CHW distribution systems has a cathodic protection system. For more information on corrosion and cathodic protection see Tredinnick (2008) and Sperko (2009).

## DIRECT CONNECTION

Because a direct connection offers no barrier between the DC provider's water and the building's own system (e.g., air-handling unit cooling coils, fan coils, process loads), the water circulated at the district plant has the same quality as the customer's water. Direct connections, therefore, are at a greater risk of incurring damage or contamination based on the poor water quality of either party. Typically, DCSs have established service agreements with water treatment vendors and/or trained in-house staff and monitor water quality continuously. This may not be the case with all consumer buildings in a DCS. A direct connection is often more economical than an indirect connection because it is not burdened by the installation of heat exchangers, additional circulation pumps, or water treatment systems; therefore, investment costs are reduced and return temperatures identical to design values are possible, i.e., there is no loss of $\Delta T$ in a heat exchanger. In general, a direct connection should be considered when a number of the following parameters are true: the building owner is the DCS owner or they are related entities; first-cost control is important; connected buildings are generally low rise in nature; building systems are new or in good condition; in-building space for the interconnection is limited; and/or the building owner, if different than the district cooling utility, respects the need for high $\Delta T$ and will maintain the building systems accordingly and retrofit the building equipment where necessary to achieve adequate $\Delta T$.

An example of a direct connection is shown in Figure 2.1. This traditional connection includes secondary or tertiary pumping within the building with a variable-speed drive controlled by a differential pressure sensor at a location that is representative of the most hydraulically remote point; hence if the differential pressure sensor is satisfied, then all coils are satisfied. In this example, the consumer's return temperature is being controlled. With this type of control, it may also be necessary to provide an override control on the consumer supply temperature in order to maintain adequate humidity control in some circumstances. A thermostatic control valve is used to control each individual in-building terminal unit (cooling coil control valve, fan coil control valve, etc.).

It is possible to have an interconnection without secondary or tertiary pumping or an ETS return temperature control valve, as shown in Figure 2.2. The ETS return temperature control valve is not required if the return temperature from the building meets the system design parameters. It is a more simple and holistic approach to ETS connections, and relies on the individual thermostatic control valves at each individual in-building terminal unit for ultimate control. This type of connection is the Holy Grail of ETS designs; if it works correctly, it is the most efficient and cost-effective design. However, it is only successful if the building engineer of record (EOR) has done a superb job of ensuring

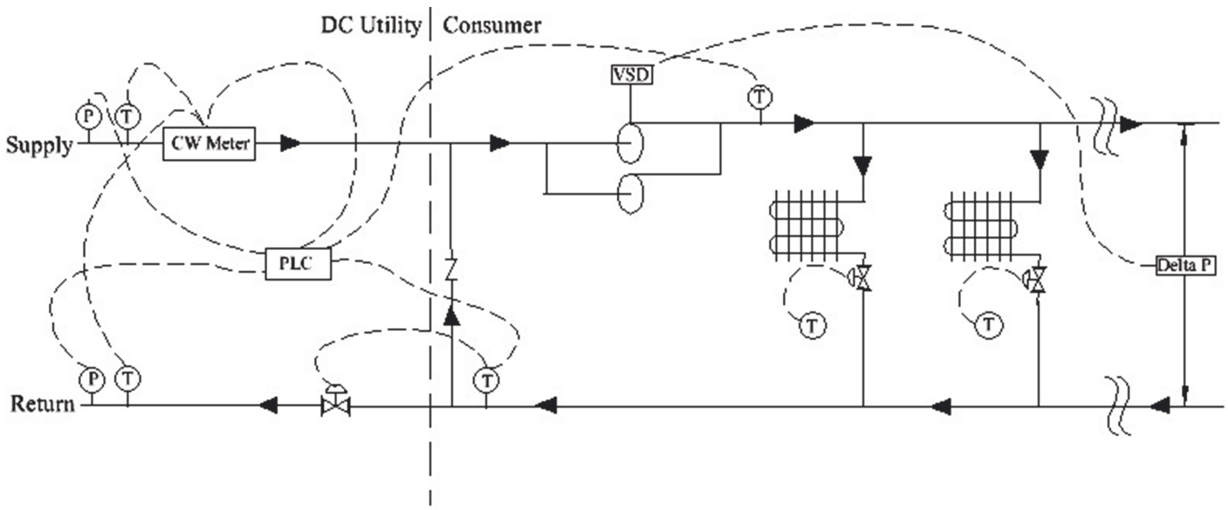



**Figure 2.1** Direct consumer interconnection with in-building pumping.

proper coil and control valve selections at the terminal units. Hence, a connection of this type leaves the DC utility entirely at the mercy of the consumer's equipment with respect to $\Delta T$ control. If the in-building heat exchangers are undersized or not of the proper configuration (i.e., not enough rows of coils), adequate $\Delta T$ may not be achieved. If the cooling coil control valve is not properly sized or is a three-way type, excess flow will be present, which will also cause degradation of chilled-water $\Delta T$. This can occur even in an otherwise well designed and balanced system when loads less than the design load are encountered, which will be discussed in Chapter 3. There are many caveats for this type of connection with a big concern being that each terminal unit control valve must be able to close off tightly against the significant system pressure that the DCP distribution pumps will create. This is not only specific to the valve but also its actuator. Typical commercial-grade terminal unit control valves and actuators may only close off against 40 to 50 psid (2.8 to 3.4 bar); therefore, it is up to the building EOR to design the system with this in mind.

As shown in Figures 2.1 and 2.2, most commercial systems have a flowmeter installed as well as temperature sensors and transmitters to calculate the energy used; metering is discussed later in this chapter. Temperature and pressure indicators should always be installed at the location of the consumer interconnection. In addition, the DC provider typically will have access to the ETS equipment and will monitor the operation from the DCP; instrumentation and control at the consumer's location are discussed in more detail later in this chapter.

With a direct connection, particular attention must be paid to connecting high-rise buildings because they induce a static head. Pressure-control devices should be investigated carefully. It is not unusual to have a DCS with a mixture of direct and indirect connections in which heat exchangers isolate the systems hydraulically for the taller buildings, i.e., high-zone/low-zone arrangements.

In a direct system, the pressure in the main distribution system must meet local building codes to protect the customer's installation and the reliability of the district system. To minimize noise, cavitation, and control problems, constant-pressure differential control valves could be installed in the buildings. Because the pressure energy consumed by the control valves can be extreme, noise can become an issue, hence special attention should

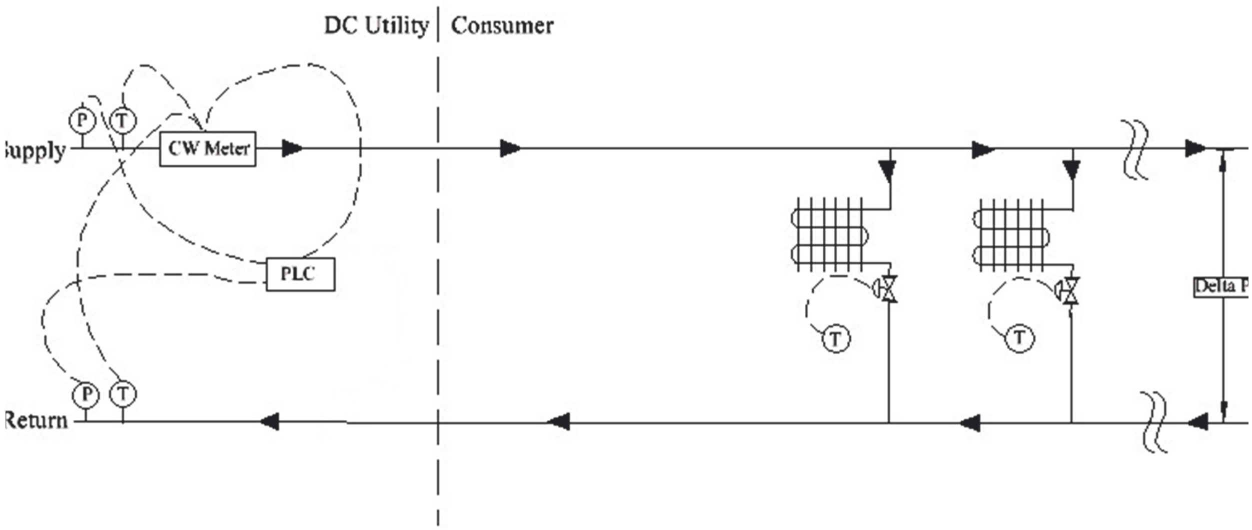



**Figure 2.2** Direct consumer interconnection without in-building pumping or Δ*T* control.

be given to potential noise problems at the control valves. A proper valve selection will reveal the calculated noise level of the valve. These valves must correspond to the design pressure differential in a system that has constantly varying distribution pressures due to load shifts. Similar to steam pressure-reducing stations, multiple valves may be required in order to serve the load under all flow and pressure ranges. Industrial quality valves and actuators should be used for this application due to the high pressure drops, turndown, and controllability requirements.

If the temperature in the main distribution system is lower than that required in the consumer cooling systems, a larger temperature differential between supply and return occurs, thus reducing the required pipe size. In the connection type shown in Figure 2.1, the consumer's desired supply temperature can be attained by mixing the return water with the DC supply water. Depending on the size and design of the main system, elevation differences, and types of customers and building systems, additional safety equipment, such as automatic shutoff valves on both supply and return lines, may be required.

When buildings have separate circulation pumps, primary/secondary piping, and pumping-isolating techniques are used (cross-connection shunts between return and supply piping, decouplers, and bypass lines). This ensures that two-way control valves are subjected only to the differential pressure established by the customer's building (tertiary) pump. Figure 2.3 shows a connection using an in-building pumping scheme with a Δ*T* control valve (i.e., return water temperature control valve).

The use of the Δ*T* control valve is typically controversial during the design of a DCS and contract negotiations with customers because it benefits the DC provider but may negatively affect the customer's chilled-water supply temperature. Ideally, if the customer's chilled-water temperature was returned per contract requirements, the Δ*T* control valve would not be required. Unfortunately, this does not occur often in reality because of the variability in the customer's HVAC system design and operation, hence a control device is required to actively manage the energy transferred. To placate all parties, the Δ*T* control valve has several operating scenarios.

The Δ*T* control valve's primary function is to provide the chilled-water supply temperature that the building requires per the contract and is controlled from a temperature

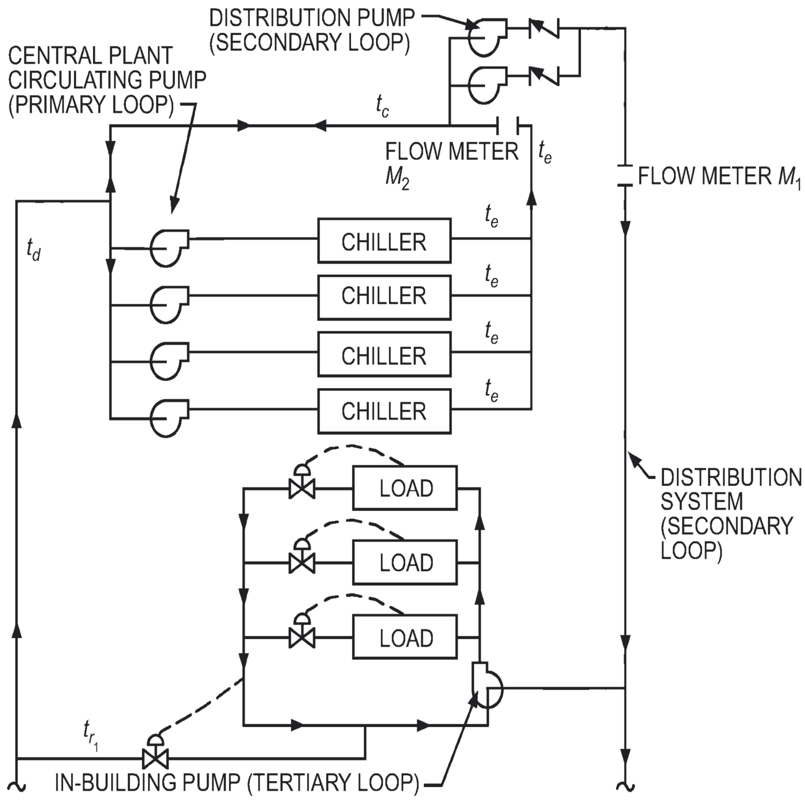



**Figure 2.3** **Direct connection with in-building and primary-secondary pumping of DC chilled water.**

transmitter in the customer's chilled-water supply piping. The secondary control function of the $\Delta T$ control valve is to ensure that the customer's return water temperature is also per the contract. If the chilled water is returned too cool or below the contracted value (as sensed by a temperature transmitter in the customer's chilled-water return pipe), the $\Delta T$ control valve will throttle back and send a portion of the return water into the supply through the decoupler until the district return water temperature reaches the desired set point. This recirculation action will increase the customer's supply water temperature due to the blending of return water. Under most part-load conditions, this is acceptable, but not in hot and humid climates since there is a danger of the supply water temperature being too high to provide dehumidification. Hence, an upper limit set point must be programmed that will only allow the customer's supply water to increase to a point where dehumidification is lost, usually around 48°F (8.9°C).

The bottom line is that the customer must take an active role in ensuring that their water is returned to the DCP per the contract requirements. This means that the terminal unit control valves must be two-way design and of a high quality (pressure independent or characterized ball valve) to be able to throttle the flow through the cooling coils while maintaining control valve authority (see Hegberg 2000; Hegberg and Hegberg 2015). Furthermore, all bypasses within the customer's chilled-water system must be removed, including a three-way valve at the end of the system to prevent pump deadheading. There are several alternative methods to perform the same function, including adding a bypass two-position valve around the pump tied to the pump variable-frequency drive (VFD) low speed signal or adding a constant flow control valve at the end of the system that bypasses

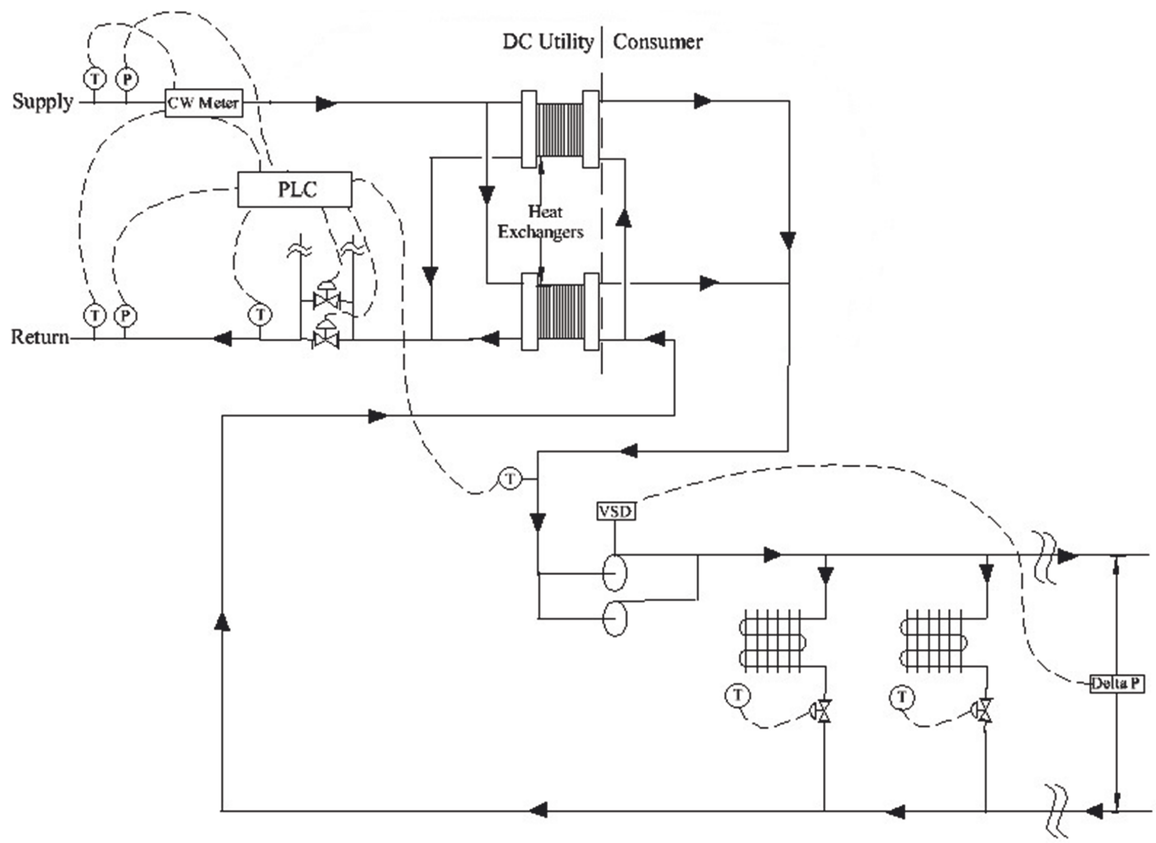



**Figure 2.4** Indirect connection of a building to a DCS.

water at a specific pressure differential at very low load but is closed during all other load conditions. In addition, of course, ongoing vigilance and maintenance needs to be performed as fouling of both air sides (filters as well as finned coil units) and water sides will affect negatively affect temperature conditions of return water, not to mention interfere with proper conditioning of the customer's spaces.

When tertiary (i.e., in-building) pumps are used, all series connections between the district system pumps should be removed or closely scrutinized for proper pump controls. Without proper pump control, a series connection may cause the district system return to operate at a higher pressure than the distribution system supply and disrupt normal flow patterns. Series operation usually occurs during improper use of three-way mixing valves in the primary to secondary connection.

## INDIRECT CONNECTION

Many of the components in an indirect connection are similar to those used in the direct connection applications with the exception that a heat exchanger performs one or more of the following functions: heat transfer, pressure interception, and buffer between potentially different qualities of water treatment. Figure 2.4 shows a form of an indirect connection.

Identical to the direct connection, the rate of energy extraction in the heat exchanger is governed by a control valve that reacts to the building load demand. Once again, the control valve usually modulates to maintain a temperature set point on either side of the heat exchanger, depending on the contractual agreement between the consumer and the producer. In Figure 2.4, the return temperature of the district cooling network water is being controlled primarily; however, an override is provided if the supply temperature of the water on the consumer side becomes too high to provide adequate comfort or dehumidification.

The three major advantages of using heat exchangers are (1) the static head influences of a high-rise building are eliminated, (2) the two water streams are separated, and (3) consumers must make up all of their own lost water and chemicals used in its treatment. The primary disadvantages of using an indirect connection are (1) the additional cost of the heat exchanger, (2) the increase in supply temperature to the consumer due to the heat-exchanger approach temperature limitation, and (3) the increased pumping pressure due to the addition of another heat transfer surface.

Whether direct or indirect connections are used, many times for critical customers that cannot be without cooling, emergency connections are extended from the district side to an outside wall or service yard of the customer's building to provide a quick interconnection to the building to provide chilled water from mobile rental equipment (not shown in the figures).

## COMPONENTS

### Pumps and Pump Control

Unless the customer's building is using a direct connection without a decoupler, building chilled-water circulation pumps will be required. Pumps should be variable flow and all terminal units should use two-way modulating control valves, preferably PICVs for optimum performance. Pumps should have adequate level of redundancy (two at 50%, etc.) so some or all of the peak capacity can be served if one pump or its VFD is out of service for maintenance purposes. Pumps shall ramp up and ramp down based on system differential pressure detected by several (at least a minimum of two) remote-mounted differential pressure transmitters (DPT). Triple duty valves should be avoided because they have no place in a variable-flow system; use proper check valves and isolation valves instead.

A means to compensate for flows below minimum pump flows (typically 25% of design flow) should be provided. This can be accomplished with a strategically located three-way valve or a two-way bypass at the pumps across the suction and discharge header or remotely mounted at the end of the system. The advantage of the two-way valve is that it can be commanded open from a signal from the VFD and they do not bypass supply water back into the return all the time. Another method is to use a constant-flow control valve in a bypass at the end of the system that is typically closed and doesn't open until the system pressure exceeds a set point that equals most of the terminal unit valves closing.

Sometimes, the DC provider desires final control of the customer's circulating pumps in order to manage the building $\Delta T$ better during low return water scenarios.

### Piping

DC service piping to the customer building must be sized to handle the flow rate of the peak demand load but also to handle any future load growth and temperature differentials lower than design. Table 6.5.4.6 in ASHRAE/IES Standard 90.1-2016 dictates pipe sizing parameters once the DC piping is within a building based on flow rate, but there are other devices associated with the connection that can also be sized based on flow and load. Table 2.4 summarizes suggested pipe, control valve, and flowmeter sizes based on peak flow rate. The DC provider should be consulted to determine if their specific ETS criteria are different from the values listed in Table 2.4.

**Table 2.4** Energy Transfer Component Sizing Based on Load and Flow using 16°F (8.9°C) Differential

| Flow Rate Range, gpm (L/s) | Capacity Range, kW (Tons) | Service Line Size, in. (mm) | Control Valve Size, in. (mm) | Minimum Flow Tube Meter Size, in. (mm) |
|---|---|---|---|---|
| 120 to 280 (7 to 18) | 80 to 190 (280 to 670) | 4 (100) | 3 (80) | 3 (80) |
| 280 to 700 (18 to 44) | 190 to 460 (670 to 1620) | 6 (150) | 4 (100) | 4 (100) |
| 700 to 1100 (44 to 69) | 460 to 730 (1620 to 2570) | 8 (200) | 4 (100) | 4 (100) |
| 1100 to 1600 (69 to 101) | 730 to 1060 (2570 to 3730) | 10 (250) | 5 (125) | 5 (125) |
| 1600 to 2300 (101 to 145) | 1060 to 1530 (3730 to 5380) | 12 (300) | 6 (150) | 6 (150) |
| 2300 to 3160 (145 to 200) | 1530 to 2100 (5380 to 7385) | 14 (350) | 8 (200) | 8 (200) |
| 3160 to 4140 (200 to 260) | 2100 to 2750 (7385 to 9670) | 16 (400) | 10 (250) | 10 (250) |

## Heat Exchangers

Heat exchangers, as shown in Figure 2.4, typically act as the line of demarcation between ownership responsibility of the different components of an indirect system. The heat exchangers transfer thermal energy and act as pressure interceptors for the water pressure in high-rise buildings. They also keep fluids from each side (that may have different chemical treatments) from mixing.

Reliability of the installation is increased if multiple heat exchangers are installed. The number selected depends on the types of loads present, the magnitude and shape of the loads, and how those loads vary throughout the year. When selecting all equipment for the building interconnection, but specifically heat exchangers, the designer should:

- Size the unit's capacity to match the given load and estimated load turndown as closely as possible. Oversized units may not perform as desired at maximum turndown; therefore, several smaller units will optimize the installation. However, it must also be considered that multiple units will cost more and require more floor space.
- Verify existing building design chilled-water temperature relative to the planned district chilled-water delivery temperatures. If chilled-water temperature reset will be used, heat exchangers may require rerating at a higher district chilled-water supply temperature at the reduced load during off-peak hours.
- Assess the critical nature of the load/operation/process to address reliability and redundancy. For example, if a building has 24-hour process loads (i.e., computer room cooling, water-cooled equipment, etc.), consider adding a separate heat exchanger for this load. Also, consider operation and maintenance of the units.
- If the customer is a hotel, hospital, casino, or data center, select a minimum of two units of at least 50% load each to allow one unit to be cleaned without interrupting building service. For customers with critical loads, multiple heat exchangers should be considered with each having the capacity to fully meet the load when one unit is out of service. Separate heat exchangers should be capable of automatic isolation during low-load conditions to increase part-load performance. Isolating a redundant heat exchanger at part load will also keep the internal velocity higher and mitigate some internal fouling of the unit.
- Determine the customer's temperature and pressure design conditions. Some gasket materials for plate heat exchangers (PHEs) have low pressure and temperature limits.
- Investigate if existing building pumps and motors can be reused or if new pumps and motors must be installed.

- Consider operation and maintenance requirements. Because PHEs require tilting the plates to remove them, adequate clearance must be available on the sides of the units to remove/add plates for cleaning or capacity growth. Multiple units may share this clearance to economize on the installation footprint. A general rule of thumb is 150% of the PHE width.
- Select the heat-exchanger approach. If the customer requires a low water supply temperature within 2°F (1.1°C) of the district system supply temperature, then select a heat exchanger with such an approach. Otherwise, choose a 3°F or 4°F (1.7°C or 2.2°C) approach because the equipment is smaller and less expensive. In any case, the benefits to the customer and the owner should be compared to the construction cost to determine if the benefits justify the additional cost of a close approach heat exchanger.
- Evaluate customer's water quality (i.e., use appropriate fouling factor).
- Determine available space and structural factors of the mechanical room.
- Recognize that removable insulated housings and drip trays for PHEs are desirable because of the ease of disassembly and reassembly for periodic maintenance requirements.
- Consider using port strainers or automatic back-flushing strainers on PHEs to mitigate clogging plates and extend the duration between full disassembly cleaning events.
- Verify existing building system design pressure classification. If the building is above 20 floors tall, the design pressure may exceed 150 psig (10.34 bar); therefore, the heat exchanger and appurtenances must be designed for this higher pressure class.
- Calculate the allowable pressure drop on both sides of heat exchangers.

The customer's side is usually the most critical for pressure drop. The higher the pressure drop, the smaller and less expensive the heat exchanger. However, the pressure drop must be kept in reasonable limits (15 psig [1.0 bar] or below) if existing pumps are to be reused in retrofit situations. Investigate the existing chiller evaporator pressure drop in order to assist in this evaluation.

The quantity of heat exchangers installed in the ETS room are based on the estimated peak load of the customer. Table 2.5 is a typical breakdown based on load. The values in the table can be increased if there are any oversizing factors used to compensate for either growth or redundancy if one heat exchanger is out of service for some reason.

All heat exchangers should be sized with future expansion in mind. When selecting heat exchangers, be cognizant that closer approach temperatures or low pressure drop require more heat transfer area, and hence cost more and take up more space. Strainers should be installed in front of any heat exchanger and control valve to keep debris from fouling surfaces.

## PHEs

Plate heat exchangers (PHEs) are the most common type of heat exchanger used in DCSs because shell-and-tube or shell-and-coil heat exchanges are not able to achieve lower approach temperatures within space constraints to be suitable for DC applications (Skagestad and Mildenstein 2002). PHEs are available as gasketed units and in two gasket-free designs (brazed and all-welded or semi-welded construction). All PHEs consist of metal plates compressed between two end frames and sealed along the edges. Alternate plates are inverted and the gaps between the plates form the liquid flow channels. Fluids never mix as DCS water flows on one side of the plate and the consumer's water flows countercurrent on

**Table 2.5** Number of Heat Exchangers Based on Customer Peak Load Estimate

| Cooling Capacity, ton (kW) | Heat Exchangers and Capacity (Number and Size), tons (kW) |
|---|---|
| 100 tons (350 kW) | 1 x 100 tons (350 kW) |
| 200 tons (700 kW) | 2 x 100 tons (350 kW each) |
| 300 tons (1055 kW) | 2 x 150 tons (525 kW each) |
| 500 tons (1760 kW) | 2 x 250 tons (880 kW each) |
| 1000 tons (3517 kW) | 3 x 333 tons (1170 kW each) |
| 1500 tons (5275 kW) | 3 x 500 tons (880 kW each) |
| 2000 tons (7035 kW) | 3 x 666 tons (2340 kW each) |
| > 2000 tons (7035 kW) | Consult DC Provider |

the other side. Ports at each corner of the end plates act as headers for the fluid. One fluid travels in the odd-numbered plates and the other in the even-numbered plates.

Because PHEs require turbulent flow for good heat transfer, pressure drops may be higher than that for a comparable shell-and-tube model. High efficiency leads to a smaller package. The designer should consider specifying that the frame be sized to hold 20% additional plates. PHEs require very little maintenance because the high velocity of the fluid in the channels tends to keep the surfaces clean. PHEs generally have a cost advantage and require one-third to one-half of the surface required by shell-and-tube units for the same operating conditions. PHEs normally achieve closer approach temperatures.

Gasketed PHEs (also called plate-and-frame heat exchangers) consist of a number of gasketed embossed metal plates bolted together between two end frames. Gaskets are placed between the plates to contain the two media in the plates and to act as a boundary. Gasket failure will not cause the two media to mix; instead the media will leak to the atmosphere. Gaskets can be either glued on or clipped on. Designers should select the appropriate gasket material for the design temperatures and pressures expected. Plates are typically stainless steel; however, plate material can be varied based on the chemical makeup of the heat transfer fluids. For maintenance recommendations for PHEs, see Tredinnick (2010).

Gasketed PHEs are typically used for DC with water and cooling tower water heat recovery (free cooling). PHEs have three to five times greater heat transfer coefficients than shell-and-tube units and are capable of achieving 1°F (0.56°C) approach temperatures. This type of PHE can be disassembled in the field to clean the plates and replace the gaskets. The gaskets can be either glued on or clipped on.

Brazed PHEs feature a close approach temperature (within 2°F [1.1°C]), large temperature drops, compact size, and a high heat transfer coefficient. Construction materials are stainless steel plates and frames brazed together with copper or nickel. Tightening bolts are not required as in the gasketed design. These units cannot be disassembled and cleaned; therefore, adequate strainers must be installed ahead of an exchanger and it must be periodically flushed clean in a normal maintenance program. Brazed PHEs typically have a peak capacity of under 2,500,000 Btu/h (about 220 plates and 120 gpm [7.6 L/s]).

## Heat-Exchanger Load Characteristics

To provide high $\Delta T$ under multiple load conditions, variable flow is required on both sides of the heat exchanger (Skagestad and Mildenstein 2002; Tredinnick 2007; Perdue and Ansbro 1999). Without variable flow on the customer side, it becomes necessary to flow more water on the DCS side at conditions of reduced load. This condition results in

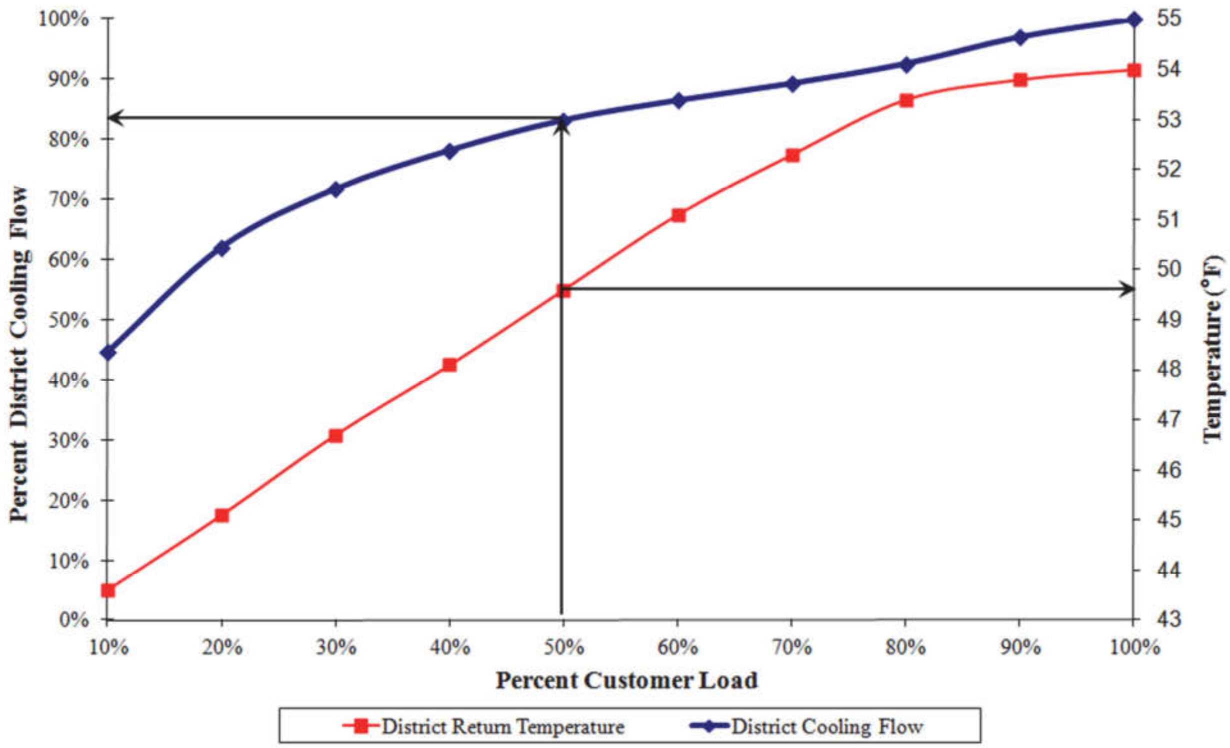



**Figure 2.5** PHE performance with constant flow on the customer side and a customer-side supply temperature of 42°F (5.6°C) (Tredinnick 2007).

both increased pumping for the DC utility as well as reduced $\Delta T$. In addition, the customer side also experiences increased pumping costs without the use of variable flow. The specific degradation in $\Delta T$ and the increases in flow depend on the actual heat exchanger selection and can easily be determined for a specific heat exchanger by use of selection and sizing software available from the heat exchanger manufacturer. An example provided by Skagestad and Mildenstein (2002) is for a 427 ton (1500 kW) design load indicates that at 50% load for constant flow on the consumer side, 75% of the design flow would be required on the DCS side whereas with variable flow on the consumer side, the required flow on the DCS side would be reduced to 45% of the design. In addition, the $\Delta T$ in this case is reduced from the design value of 15°F (8.3°C) to just 10°F (5.6°C) at 50% load with constant flow on the consumers side. For this case, when variable flow is used on the consumer's side of the PHE, the $\Delta T$ is actually increased from the design value of 15°F to 16.7°F (8.3°C to 9.3°C).

Another example of the need for variable-flow pumping on the consumer's side of PHE is provided by Tredinnick (2007) for a 500-ton (1750 kW) application. In Figure 2.5, the consumer side of the heat exchanger has constant flow on the consumer side with the consumer side design supply temperature of 42°F (5.6°C). The PHE has been sized such that at 100% of design load, the district cooling return temperature will be 54°F (12.2°C); thus, a $\Delta T$ of 14°F (7.8°C) will be achieved at this maximum load condition assuming 2°F (1.1°C) approach temperature. However, with constant flow on the consumer side of the PHE at 50% of the design load, over 83% of the peak design flow on the district cooling side will be required and the district cooling return temperature will

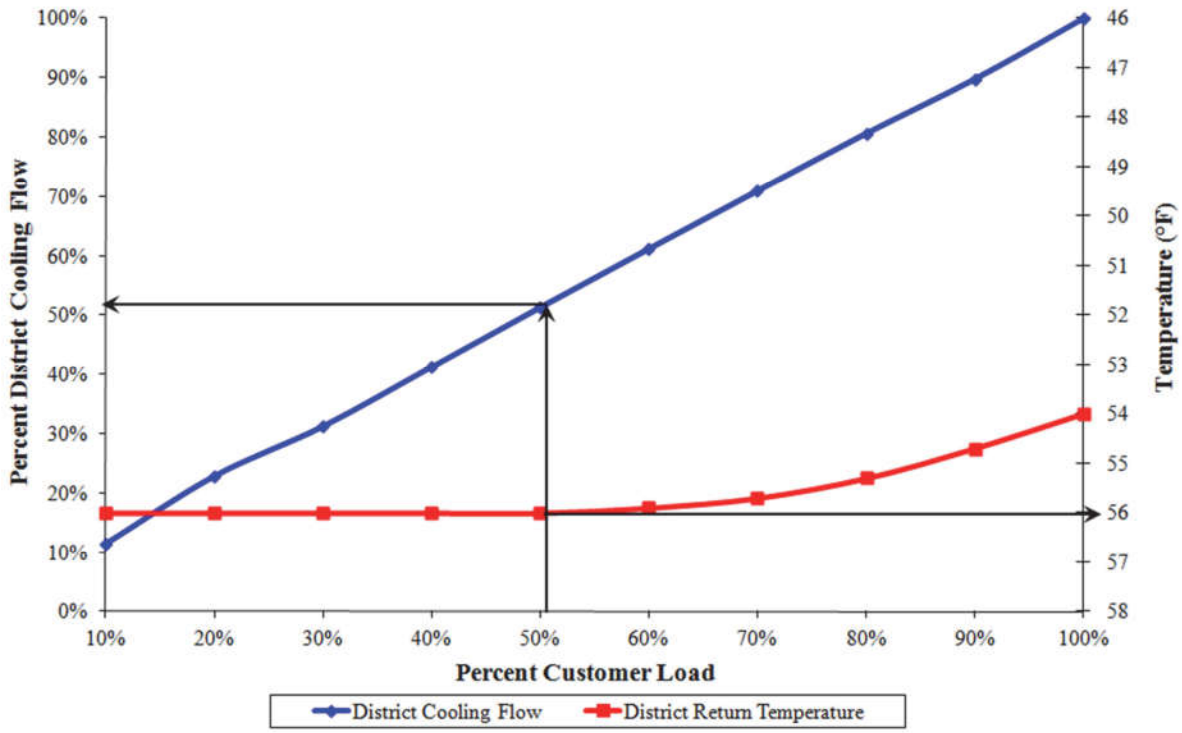



**Figure 2.6** PHE performance with variable flow on the customer side and a customer-side supply temperature of 42°F (5.6°C) (Tredinnick 2007).

have decreased to 49.6°F (9.8°C), and thus the $\Delta T$ on the district cooling side will have decreased to 9.6°F (5.3°C).

Figure 2.6, also from Tredinnick (2007), illustrates the situation under identical conditions but with variable flow on the consumer side of the PHE. As before, with the consumer-side design supply temperature of 42°F (5.6°C) and at 100% of design load, the PHE has been sized to yield a district cooling return temperature of 54°F (12.2°C); thus, at the design condition, the $\Delta T$ will be 14°F (7.8°C), again assuming a 2°F (1.1°C) approach temperature. However, under the 50% load condition with variable flow on the consumer's side, only 51% of the design flow is required on the district cooling side of the PHE and the return temperature has actually increased to 56°F (13.3°C). Thus, the $\Delta T$ for the DCS has increased from 9.6°F (5.3°C) with constant flow under this load condition to 16°F (8.9°C) for variable consumer-side flow.

Variable flow also saves electrical pump energy and aids in controlling comfort. These examples, as well as others (see Perdue and Ansbro 1999) should make clear the need for variable flow on the consumer side of the PHE in an indirect connection of district cooling.

Typical constant-flow systems are found in older buildings and may be converted to simulate a variable-flow system by blocking off the bypass line around the air handler heat exchanger coil three-way control valve. At low operating pressures, this potentially may convert a three-way bypass-type valve to a two-way modulating shutoff valve. Careful analysis of the valve actuator must be undertaken since the shut-off requirements and control characteristics are totally different for a two-way valve compared to a three-way valve. More information on building conversion can be found in Skagestad and Mildenstein (2002).

In theory, a partially loaded cooling coil should have higher return-water temperature than at full load since the coil is oversized for the duty and hence has closer approach temperatures. In many real systems, as the load increases, the return-water temperature tends to rise, and with a low-load condition the supply water temperature rises. Consequently, process or critical humidity control systems may suffer when connected to a system where return-water temperature control is used to achieve high temperature differentials. Other techniques, such as separately pumping each chilled-water coil, may be used where constant supply water temperatures are necessary year-round.

## Flow-Control Devices

In commercial district energy systems, second to the flowmeter, control valves are the most important element within the interface with the district energy system because proper valve adjustment and calibration save energy by ensuring high $\Delta T$ under all load conditions. High-quality, industrial-grade control valves provide more precise control, longer service life, and minimum maintenance.

All control valve actuators should take longer than 60 s to close from full open to mitigate pressure transients or water hammer, which occurs when valves close more rapidly. Actuators should also be sized to close against the anticipated system pressure so the valve seats are not forced open, thus forcing water to bypass and degrading temperature differential. Buildings near the central plant may require additional pressure-reducing valves upstream of the control valves.

The wide range of flows and pressures expected makes the selection of control valves difficult. Typically, only one control valve is required; however, for optimal response to load fluctuations and to prevent cavitation, two or more valves in parallel are often needed, as shown on the DCS side of Figure 2.4. The two valves operate in sequence and for a portion of the load (i.e., one valve is sized for two-thirds of peak flow and the other sized for one-third of peak flow). The designer should review the occurrence of these loads to size the proportions correctly. The possibility of overstating customer loads complicates the selection process, so accurate load information is important. It is also important that the valve selected operates under the extreme pressure and flow ranges foreseen. Because most commercial-grade valves will not perform well for this installation, industrial-quality valves should be specified.

Electronic control valve actuators should remain in a fixed position when a power failure occurs and should be manually operable. Pneumatic control valve actuators should close upon loss of air pressure. A manual override on the control valves allows the operator to control flow with either loss of power or control air. All chilled-water control valves must fail in the closed position. Thus, when any secondary in-building systems are de-energized, the valves close and will not bypass chilled water to the return system. The valves must close slowly as rapid closure at peak load conditions may cause damaging water hammer. Oversizing of control valves reduces valve life and causes valve hunting. Select control valves having a wide range of control, low leakage, and proportional-plus-integral control for close adjustment, balancing, temperature accuracy, and response time. Control valves should have actuators with enough force to open and close under the maximum pressure differential in the system. The control valve should have a pressure drop through the valve equal to at least 10% to 30% of the static pressure drop of the distribution system. This pressure drop gives the control valve the authority it requires to properly control flow. The relationship between valve travel and capacity output should be linear with an equal percentage characteristic.

In chilled-water systems, control valves can be installed either in the supply or return line; typically, however, they are installed in the return line to reduce the potential for condensation on any exposed external surfaces and to minimize any water turbulence upstream of the flowmeter.

## Instrumentation and Control

In many systems where energy to the consumer is measured for billing purposes, temperature sensors assist in calculating the energy consumed as well as in diagnosing performance. Sensors and their transmitters should have an accuracy range commensurate to the accuracy of the flowmeter. In addition, pressure sensors are required for variable-speed/variable-frequency (VS/VF) pump control or valve control for pressure-reducing stations. Detailed recommendations on pressure, temperature, flow, and power transducers may be found in IDEA (2008). Temperature sensors need to be located by the heat exchangers being controlled rather than in the common pipe. Improperly located sensors will cause one control valve to open and others to close, resulting in unequal loads in the heat exchangers.

Table 2.6 provides a list of common measuring points and derivative parameters for remote monitoring and control of an indirect consumer interconnection. Other measuring points and derivative parameters may be required/recommended where the DC utility assumes some responsibility for operation of the building systems or the building and DCSs belong to a single owner.

At the consumer's location, the controller performs several functions, including recording demand and the amount of energy used for billing purposes, monitoring the differential pressure for plant pump control, performing energy calculations, alarming for parameters outside normal, and monitoring and controlling all components.

**Table 2.6** Measuring Points and Derivative Parameters for Remote Monitoring and Control of an Indirect Consumer Interconnection

| Measured Point/Parameter | Location |
|---|---|
| Temperature | DC-side supply |
| | DC-side return (Optional) |
| | Consumer-side supply |
| | Consumer-side return |
| Pressure | DC-side supply |
| | DC-side return |
| | Consumer-side supply |
| | Consumer-side return |
| Differential pressure | DC-side at building entrance |
| | DC-side of heat exchanger(s) |
| | Consumer-side of heat exchanger(s) |
| | DC-side control valve(s) |
| | DC-side strainer |
| Flow rate | DC-side water |
| Energy transfer | DC-side water |
| Position of control valve(s) | DC side |
| Variable-speed drive percentage(s) | Consumer side |

Typical control strategies include regulating district flow to maintain the customer's supply temperature (which results in a fluctuating customer return temperature) or maintaining the customer's return temperature (which results in a fluctuating customer supply temperature). When controlling return flow, the impact on the customer's ability to dehumidify properly with an elevated entering coil temperature should be investigated carefully. As discussed earlier, proper design and control of the customer's heat transfer coils are necessary to ensure adequate $\Delta T$ for efficient operation of the DCS.

### Temperature Measurement

Temperature measurements should be made with sensors located in wells to allow for change-out. Both resistance temperature detector (RTD) and thermistors have been used for temperature sensing elements. Temperature measurement transducers should be specified to provide accuracy of ±1°F (0.56°C) (IDEA 2008). For thermal meters (discussed in more detail in the "Metering" section), temperature measurement may be made by differential methods in order to achieve higher accuracy of the differential temperature, which is the parameter of interest rather than the absolute temperature. Furthermore, a matched pair of RTDs should be used with flowmeters when calculating energy usage to get a more accurate reading.

### Pressure Measurement

Aside from the customary mechanical pressure gauges that should be provided at the end user interface for on-site diagnostics, pressure transducers are normally provided for remote monitoring and in many instances control. Pressure transducers should be specified to provide accuracy of ±1% of full scale and typically resolution of 0.1 psi (6.9 mbar) (IDEA 2008).

### Pressure-Control Devices

If the water pressure delivered to the customer is too high for direct use, it must be reduced. Similarly, pressure reducing or sustaining valves may be required if building height creates a high static pressure and influences the DCS return-water pressure for directly connected buildings. Water pressure can also be reduced by control valves or regenerative turbine pumps. The risks of using pressure-regulating devices to lower pressure on the return line is that if they fail, the entire distribution system (and other directly connected building) will be exposed to their pressure and over pressurization will occur.

In high-rise buildings, all piping, valves, coils, and other equipment may be required to withstand higher design pressures. Where system static pressure exceeds safe or economical operating pressure, either the heat exchanger method or pressure-sustaining valves in the return line may be used to minimize the impact of the pressure. Vacuum vents should be provided at the top of the building's water risers to introduce air into the piping in case the vertical water column collapses.

## METERING

All chilled water delivered by a commercially operated DCS to customers or end users for billing or revenue must be metered. For such systems, the meter is the most important component of the end-user interface; it is often referred to as the "cash register" of the system as it forms the basis for billing the customers. For DCSs under common ownership for the buildings, distribution system, and central plant, metering is also advisable for diagnostic and optimization purposes. The type of meter selected depends on the accuracy required and the expected turndown of flow to meet the low flow and maximum flow conditions. Typically, the higher the accuracy desired, the higher the cost of the meter. Hence, it is important that the meter is sized accurately for the anticipated loads

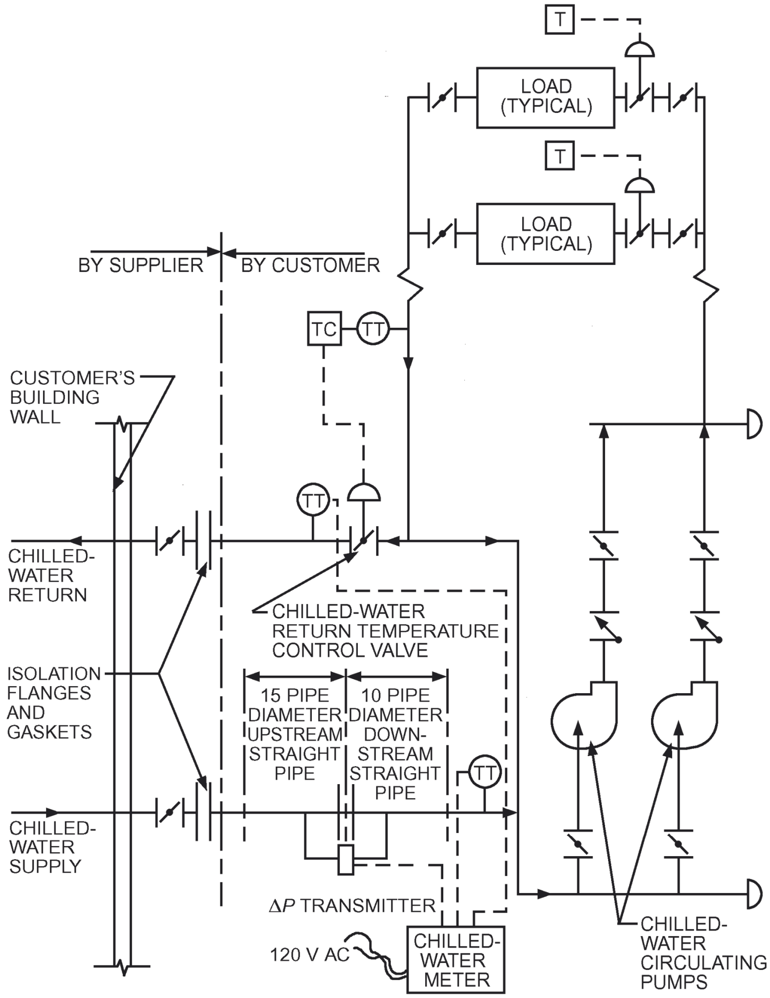



**Figure 2.7** Typical chilled-water piping and metering diagram.

and not oversized since this will lead to inaccuracies. Historical metered or otherwise benchmarked data should be used when available if the actual load is not accurately known.

For DCSs, energy is calculated by measuring the temperature differential between the supply and return lines and the flow rate. Thermal (Btu or kWh) meters compensate for the actual volume and heat content characteristics of the chilled water. Thermal transducers, resistance thermometer elements, or liquid expansion capillaries are usually used to measure the differential temperature of the energy transfer medium in supply and return lines. Figure 2.7 shows a typical DC connection with metering.

Water flow can be measured with a variety of meters, usually pressure differential, turbine or propeller, or displacement meters. Chapter 37 of *ASHRAE Handbook—Fundamentals* (ASHRAE 2017a) and Skagestad and Mildenstein (2002) have more information on measurement. Ultrasonic meters are sometimes used to check performance of installed meters. Various flowmeters are available for district energy billing purposes. Critical characteristics for proper installation include clearances and spatial limitations as well as the attributes presented in Table 2.7. Note that the data in the table only provide general

**Table 2.7** Flowmeter Characteristics

| Meter Type | Accuracy | Range of Control | Pressure Loss | Straight Piping Requirements (Length in Pipe Diameters) |
|---|---|---|---|---|
| Orifice plate | ±1% to 5% full scale | 3:1 to 5:1 | High (>5 psi) | 10 D to 40 D upstream; 2 D to 6 D downstream |
| Electromagnetic | ±0.15% to 1% rate | 30:1 to 100:1 | Low (<3 psi) | 5 D to 10 D upstream; 3 D downstream |
| Vortex | ±0.5% to 1.25% rate | 10:1 to 25:1 | Medium (3 to 5 psi) | 10 D to 40 D upstream; 2 D to 6 D downstream |
| Turbine | ±0.15% to 0.5% rate | 10:1 to 50:1 | Medium (3 to 5 psi) | 10 D to 40 D upstream; 2 D to D downstream |
| Ultrasonic | ±1% to 5% rate | >10:1 to 100:1 | Low (<3 psi) | 10 D to 40 D upstream; 2 D to 6 D downstream |

guidance, and the manufacturers of meters should be contacted for data specific to their products.

High-accuracy meters are desired for more accurate billing of the customers, which benefits both the DC provider and the customer by reducing the incidences of disputes over billing.

Flowmeters also have a specific flow rate range where they are most accurate. Table 2.4 reflects that sometimes the flow tube should be less than service line size to maintain accuracy at lower flow rates.

Displacement meters are more accurate than propeller meters, but they are also larger. They can handle flow ranges from less than 2% up to 100% of the maximum rated flow with claimed ±1% accuracy. Turbine-type meters require the smallest physical space for a given maximum flow. However, like many meters, they require at least 10 diameters of straight pipe upstream and downstream of the meter to achieve their claimed accuracy.

The meter should be located upstream of the heat exchanger and the control valve(s) should be downstream from the heat exchanger. This orientation minimizes the possible formation of bubbles in the flow stream and provides a more accurate flow indication. The transmitter should be calibrated for zero and span as recommended by the manufacturer.

Wherever possible, the type and size of meters selected should be standardized to reduce the number of stored spare parts, technician training, etc.

Until recently the United States had no performance standards for thermal meters. Recently, however, ASTM Standard E3137 was enacted (ASTM 2018). Where DC utilities are regulated by a public utilities commission, many are required to meet an accuracy standard of ±2% with periodic testing for continued assurance. ASHRAE Standard 125 describes a test method for rating liquid thermal meters (ASHRAE 1992). Several European countries have developed performance standards and/or test methods for thermal meters, and EN 1434 (CEN 2007), developed by the European Community, is a performance and testing standard for heat meters.

District energy plant meters intended for billing or revenue, require means for verifying performance periodically. Major meter manufacturers, some laboratories, and some district energy companies maintain facilities for this purpose. In the absence of a single performance standard, meters are typically tested in accordance with their respective manufacturers' recommendations. Primary measurement elements used in these laboratories frequently obtain calibration traceability to the National Institute of Standards and Technology (NIST).

## REFERENCES


ASHRAE. 1992. ANSI/ASHRAE Standard 125-1992 (RA 2011), *Method of testing thermal energy meters for liquid streams in HVAC systems*. Atlanta: ASHRAE.

ASHRAE. 2014. *Load calculation applications manual,* 2d ed. Atlanta: ASHRAE.

ASHRAE. 2016. ANSI/ASHRAE/IES Standard 90.1-2016, *Energy standard for buildings except low-rise residential buildings*. Atlanta: ASHRAE.

ASHRAE. 2017a. *ASHRAE handbook—Fundamentals*. Atlanta: ASHRAE.

ASHRAE. 2017b. ANSI/ASHRAE/ACCA Standard 183-2007 (RA 2017)*, Peak cooling and heating load calculations in buildings except low-rise residential buildings*. Atlanta: ASHRAE.

ASTM. 2018. ASTM E3137/E3137M-18, *Standard specification for heat meter instrumentation*. West Conshohocken, PA: ASTM International.

CEN. 2007. EN 1434, *Heat meters, Parts 1–6*. Brussels, Belgium: European Committee for Standardization.

Hegberg, M.C. 2000. Control valve selection for hydronic systems. *ASHRAE Journal* (November):33-39.

Hegberg, M.C., and R.A. Hegberg. 2015. *Fundamentals of water system design*, 2d ed. Atlanta: ASHRAE.

IDEA. 2008. *District cooling best practices guide*. Westborough, MA: International District Energy Association.

Perdue, S. and J. Ansbro. 1999. Understanding design implications of constant vs. variable flow pumping on plate heat exchangers, IDEA District Cooling Conference, October 7. Atlantic City, NJ.

Phetteplace, G., S. Abdullah, J. Andrepont, D. Bahnfleth, A. Ghani, B. Kirk, V. Meyer, and S. Tredinnick. 2019. *District cooling guide*, 2d ed. Atlanta: ASHRAE.

Rishel, J. 2007. Connecting buildings to central chilled water plants. ASHRAE Journal 49(11).

Skagestad, B., and P. Mildenstein. 2002. *District heating and cooling connection handbook*. Sittard, Netherlands: Netherlands Agency for Energy and Environment (NOVEM), operating agent for International Energy Agency.

Sperko, W. 2009. Dissimilar metals in heating and ac piping systems. *ASHRAE Journal* 51(4):28–32.

Tredinnick, S. 2007. Tales of the paranormal: PHE's variable countercurrent flows, and the "X-Files." "Inside Insights." *District Energy*: Fourth quarter. Westborough, MA: International District Energy Association.

Tredinnick, S. 2008. Opposites Attract: The color purple and galvanic corrosion. "Inside Insights." *District Energy*: Second quarter. Westborough, MA: International District Energy Association.

Tredinnick, S. 2010. Maintaining plate heat exchangers: No need for an idiot light here. *District Energy*: First quarter. Westborough, MA: International District Energy Association.

# 3

# Existing Buildings: When Design Deficiencies or Other Constraints Prevent Achieving Acceptable $\Delta T$

While buildings experiencing chilled-water temperature differentials ($\Delta T$s) lower than design is not uncommon, why is it that buildings that do produce $\Delta T$s that are at or above design seem to be uncommon? Why can't buildings achieve their design $\Delta T$ at peak or part load? It is such a common occurrence that the industry has coined the phrase low-$\Delta T$ syndrome to diagnose the problem. The impacts of the syndrome are mostly felt in the DCP, where additional energy is required to meet the increased chilled-water flow demands and reduced chiller efficiency (see Chapter 1 for an example), but also could impact the comfort of the DC-connected building's occupants, depending on the contract when a chilled-water return temperature control valve is present on the district side of the connection.

First and foremost, the simplest and most common cause of low $\Delta T$ is the selection and operation of the terminal cooling units and control valves. Other than bypasses or shunts between the chilled-water supply and return piping, almost all low chilled $\Delta T$ occurrences can be pinpointed to these devices. Hence, quite a bit of space in this chapter is devoted to cooling coils, control valves, and actuators. The prevalent use of lower quality, two-position control valves and three-way control valves also are detrimental to keeping chilled-water return temperatures high.

It is noteworthy to state that at part load the leaving chilled-water temperature should be higher than design since the coil is oversized for the load and the approach temperature between the coil entering air and leaving water temperature should be smaller. So why is low-$\Delta T$ syndrome prevalent? This chapter covers the reasons behind the phenomena and suggests corrective actions that can be taken to improve a building's $\Delta T$.

## CAUSES OF LOW-$\Delta T$ SYNDROME

Taylor (2002) covers the causes and mitigation in detail and is a good desk reference but most of the issues can be categorized as either improper design or improper operation and maintenance. Many of the causes will be covered in detail in this chapter, but some of the common causes that can be addressed and mitigated with proper design and operation practices are from:

- **Improper Design and Installation**
    - Secondary pump differential pressure transmitters are not located properly.
    - Coils and valves are not properly sized or selected.
        - Three-way valves are used (water is being bypassed at all times except peak load).

    - Correctly select (see below for guidance) coils and valves; do not oversize.
    - Actuators undersized and cannot close off against system pressure.
  - Cooling coil leaving air set points set lower than design.
    - Control valves go fully open—bypassing water and having no control
  - Valves not closed when not required.
    - Valves should close when the air-handler unit (AHU) is not in use
  - Coils piped "backwards."
    - Coils must be piped so water flow is counterflow to airflow.
- **Improper Operation and Maintenance**
  - Instruments are out of calibration, which causes improper control signals
    - Clean, calibrate, or replace sensors.
  - There are "leaks" in the chilled-water system (from supply side to return side).
    - Actuators are undersized and cannot close off against system pressure.
    - Control valves do not close when AHU or terminal unit is de-energized.
    - Fouled coils—keep interior and exterior of coils and fins clean.
    - Dirty filters—change filters regularly.
    - Bypassing air around coil—seal AHU sections tight.
    - Uncontrolled process loads.
    - Bypasses around equipment installed for commissioning or startup (flushing and cleaning) purposes that are still open.

Low-$\Delta T$ syndrome is not a new phenomenon and has been well documented for several decades in trade journals and conference presentations. One such presentation by now-retired Trane applications engineer Don Eppelheimer discussing the above symptoms of low-$\Delta T$ syndrome coined the phrase "$\Delta T$ Pirates," since the symptoms are thieves which rob and plunder the efficiency from the chilled-water systems. While the comment was light hearted, it does effectively describe the topic.

## BEST PRACTICES FOR SELECTING COOLING COILS

Following some general coil selection guidelines early in the design process will solve a great deal of the operational issues later. Coils should be selected to keep water velocity in the turbulent range for both full load and low load conditions. Hence, at low load, the water velocity in the tubes must remain above 0.8 to 1.0 fps (0.24 to 0.3 m/s) for a standard 5/8 in. (16 mm) tube to stay in the turbulent range.[2] Laminar flows drastically reduce the heat transfer

2. The Reynolds number Re is a dimensionless measure of the level of turbulence; increased turbulence enhances heat transfer. If $Re \geq 10{,}000$, then the flow is deemed fully turbulent. If $Re \leq 2100$, the flow is laminar. The value between these two extremes is the transition area. ARI 410 (AHRI 2001) only certifies cooling coils above Re 3100 (turbulent and transition ranges), thus avoiding laminar flow range.

$$Re = D \rho v / \mu$$

where

| | | |
|---|---|---|
| D | = | diameter |
| $\rho$ | = | density |
| v | = | velocity |
| $\mu$ | = | viscosity |

Any consistent set of units may be used.

characteristics. Heat *is* still transferred at laminar flow; however, the actual heat transfer rate is unpredictable and control valve "hunting" may occur due to the low flow and valve hysteresis.

Therefore, for good part-load performance, the full-flow tube velocity must be a minimum of 4 fps (1.2 m/s) and ideally in the 5 to 6 fps range (1.5 ton 1.8 m/s) and coils should not be oversized for capacity. In a grossly oversized coil, this usually means a lower leaving water temperature and, hence, a lower system $\Delta T$, and the problem is compounded. As seen from Figure 3.1, coil output capacity of 25% of full load may correspond to a flow requirement less than 15 to 20%, therefore, illustrating:

- Coil flow rates are not linear with load
- Part-load conditions are critical in the proper selection of coils and accompanying control valves

The basic concept is for the chilled water to stay in the coil longer, thus requiring more circuits, more rows, and more of a pressure drop. The longer the water stays in the coil, the more opportunity it has to transfer heat and leave warmer. If coil selection results in an eight-row unit, investigate using two independently controlled four-row units in series. While this increases installation costs, it should improve dehumidification and heat transfer characteristics at part-load conditions since one coil slab can be controlled independently while the other has no flow. Similarly, and probably more successful, is using the return water from the main cooling coil as the inlet water to an outside air precooling coil to increase overall system $\Delta T$. Another method of increasing system return water temperature is by using stacked cooling coils that are controlled independently.

Select a coil for higher entering water temperature (EWT) than expected and highest practical $\Delta T$. This will result in the selected coil being slightly larger with a higher pressure drop. For example, if building circuit is designed for 42ºF (5.6°C)

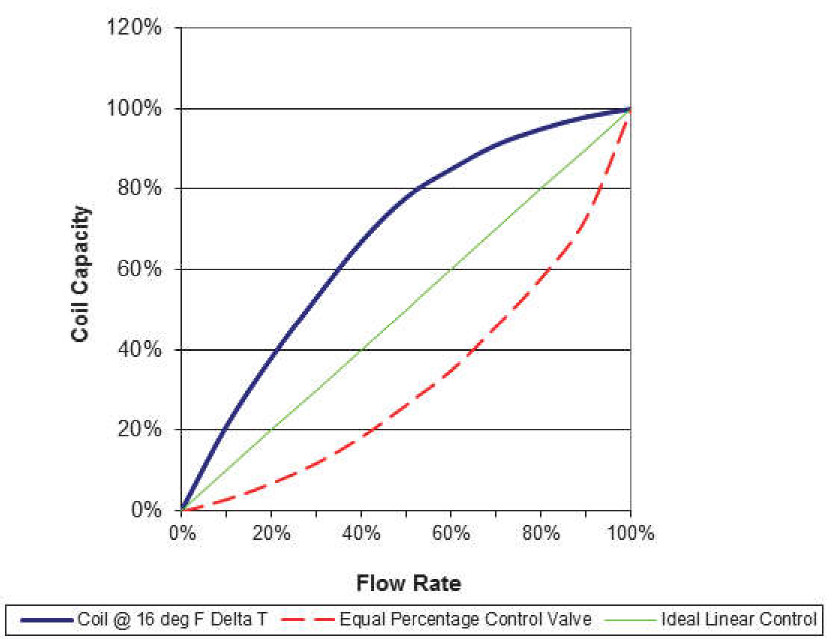



**Figure 3.1** AHU cooling coil performance based on varying flow rates.

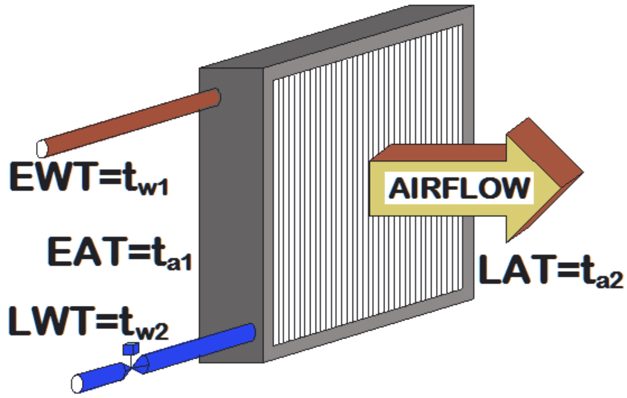



**Figure 3.2** Cooling coil performance heat transfer.

then size coil for 44ºF (6.7°C).[3] While grossly oversizing coils is not recommended, this practice of slightly oversizing coils allows for aging and fouling of heat transfer surfaces as well.

The performance of heat exchangers is characterized with the logarithmic mean temperature difference (LMTD), if a coil is supplied colder water $t_{w1}$ than that which it was designed, the leaving water $t_{w2}$ will be higher than what it was designed for if the load is the same, per Equation 3.1 and Figure 3.2. This process increases the $\Delta T$, which naturally decreases the flow, water velocity, and pressure drop through the coil and is effective both for an existing or a new coil. When applying the method to an existing coil, the designer must pay attention to whether the water velocity falls below laminar flow during part-load conditions as shown in Figure 3.3.

$$\text{LMTD} = \Delta T_m = (T_{a1} - T_{w1})/(\ln_e[(T_{a1} - T_{w2})/(T_{a2} - T_{w1})]) \qquad (3.1)$$

where

| | | |
|---|---|---|
| LMTD | = | $\Delta T_m$ = log mean temperature difference, °F (°C) |
| $T_{a1}$ | = | entering air temperature, °F (°C) |
| $T_{a2}$ | = | leaving air temperature, °F (°C) |
| $T_{w1}$ | = | entering water temperature, °F (°C) |
| $T_{w2}$ | = | leaving water temperature, °F (°C) |

For example, by using a coil manufacturer's selection software program, if a coil was selected for 10,000 CFM (283 $m^3$/min), 44°F (6.7°C) entering water temperature (EWT) and 54°F (12.2°C) leaving water temperature (LWT) with a flow of 100 gpm (378.5 liter/min), the LWT will rise as the EWT falls. Table 3.1 summarizes the results.

As previously mentioned, it is recommended to select coils with higher fluid velocities in the tubes to keep flows in the turbulent region of heat transfer at lower loads. Figure 3.3 illustrates a typical coil characteristic for coils selected 4.0 fps (1.2 m/s) and above

3. To avoid confusion regarding flows and "balancing" issues because the $\Delta T$s do not match and flow rates will not "add up," it may be prudent to schedule the cooling coils for both coil selection flow rate and design flow rate while maintaining the same $\Delta T$.

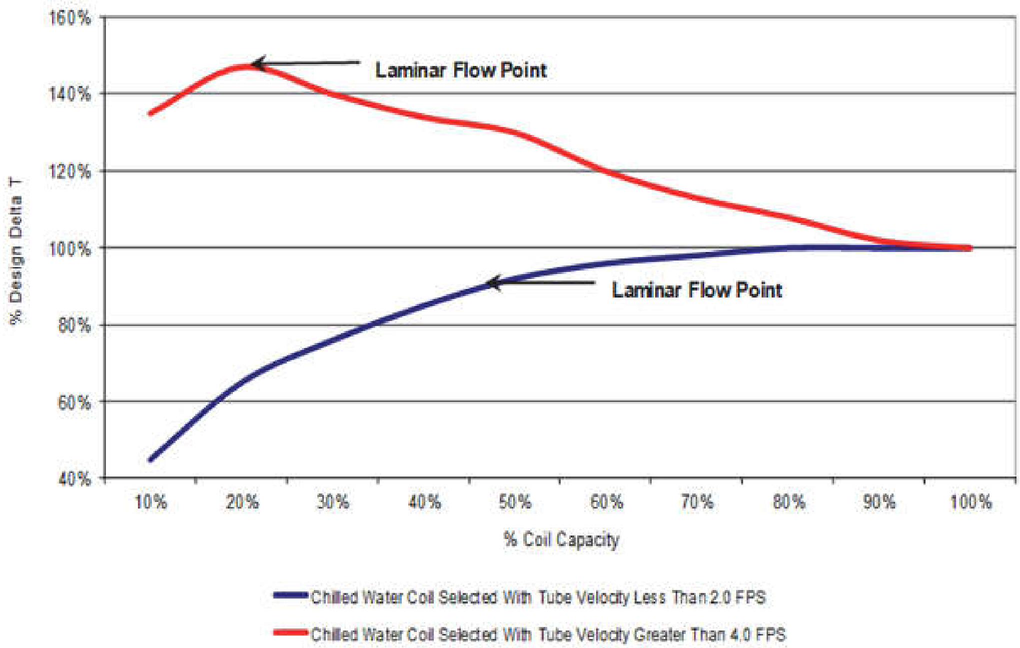



**Figure 3.3** Cooling coil performance due to water velocity.

**Table 3.1** Coil Performance Based on Lowering EWT

| Case | EWT, °F (°C) | LWT, °F (°C) | Flow, gpm (L/s) |
|---|---|---|---|
| Base | 44°F (6.7°C) | 54°F (12.2°C) | 100 (378.5) |
| 1 | 40°F (4.4°C) | 57.2°F (14.0°C) | 56.5 (213.9) |
| 2 | 41°F (5.0°C) | 57.2°F (14.0°C) | 59.4 (224.9) |
| 3 | 42°F (5.6°C) | 56.5°F (13.6°C) | 68 (257.4) |

and coils selected less than 2.0 fps (0.6 m/s) where coils selected with higher water velocities can turn down much lower and serve part loads more effectively (~20% turndown compared to ~50% turndown).

Coils in AHUs and fan coils should be air tight and must not have thermal leaks (air or water by-passing coil) or an accumulation of dirt, rust, slime or scaling (air or water side fouling) which effects the heat transfer coefficient and obstructs airflow, thus effecting the coil output. Similarly, dirty filters decrease coil performance by reducing airflow, which requires a lower discharge air temperature to satisfy the load. The reduction in discharge air temperature will cause the return water temperature to drop due to a shortage of coil surface area and the control valve going wide open.

Many coils were selected for discharge air temperatures higher than what the conditions they are forced to operate. Lowering the discharge air set point has the unwanted effect of reducing the leaving water temperature. The cooling coil discharge air temperature should be increased back to design values. Furthermore, control instrumentation should be calibrated regularly. If not working properly they should be modified, repaired or replaced.

In summary, the best practices for coil selection are:

- Select a coil for higher EWT than available from the DCS supply temperature.

- Size coils for a high initial tube water velocity (over 5 fps or 1.5m/s) using partially circuited (½ or ¼ circuits) coils resulting in coil pressure drops of 15 to 20 ft (4.5 to 6.1 m).
- Use turbulators to delay the onset of laminar flow.
- Keep coils clean, inside and out.
- Change air-side filters regularly.
- Blowdown water-side strainers regularly.
- Provide direct digital controls.

## BEST PRACTICES FOR SELECTING CONTROL VALVES

Like proper cooling coil selection, proper selection of the control valve and actuator combination permit the coil to operate efficiently. The valve type must be a marriage between the non-linear coil capacity profile and the valve operation to create an equal percentage profile for controllability. All control valves should be modulating two-way valves and there is no reason to use three-way valves at all in a building especially for pump protection since there are other methods to provide that feature without bypassing water 99% of the time.

Figure 3.1 illustrates this graphically with the coil characteristic curve and the equal percentage valve characteristic curve combine to create a linear operating characteristic for coil output compared to control valve stem travel or percent open. Hegberg (2000) discusses this combined characteristic curve in detail as to how to size a control valve based on valve authority. Having an improperly sized 2-way control valve (i.e., too large) has several negative impacts such as controllability and is almost as bad as using 3-way valves.

Selecting a control valve properly for a large air handling unit (AHU) can be involved, however, a great deal of the cooling systems in the Middle East center around the use of two pipe fan coils. Fan coil units generally come with inexpensive quick opening, two-position (open/close), flapper operated, not modulating, control valves. The prevalence of these valves is the bane of high return water design since they allow more flow through the coil than what the design requires to meet the cooling load. This is due to the open-close function (not modulating) and the low close-off pressure rating associated with this type of control valve, allowing the valves to leak through the commanded closed valve due to high system differential pressure. If possible, the fan coil control valve should be upgraded to a modulating control valve (globe, characterized ball or pressure independent valves) with an equal percentage flow characteristic and the required close-off pressure rating (which differs from the valve body pressure rating). This will contribute to a higher leaving coil water temperature.

Pressure independent control valves (PICV) are designed to eliminate the effects of pressure variations on the control valve which contribute to the overflow through the associated cooling coils. PICV are basically two devices in one serving as the control valve and the balancing device. PICVs are more expensive than a standard control valve, however, after factoring the cost of a separate balancing device and its associated labor, the additional cost is minimal (and in some cases less expensive) than a traditional control valve and balancing device. Options to the PICV are pressure dependent equal percentage valves such as characterized ball valves with an automatic flow control valve (AFCV).

For example, in high rise buildings the pressure differential will vary floor by floor the further away the connections get from the building chilled water pumps. PICVs can be added at each device or an AFCV can be added on each floor to reduce the pressure drop and provide equivalent pressure differentials for each floor. However, this solution is only as effective as a PICV if the building is at design condition. Peterson (2017) states that a flow-limiting assembly does nothing when the flow is below design as it does nothing to limit the differential pressure variations across the control valve, thus, the control valve is seeing the effects of poor valve authority while throttling.

It is also recommended that the coil control valves be direct acting actuator with PID control, and good rangeability. Globe style valves are usually superior to butterfly valves based on percent open and amount of flow. If control valves are undersized then maximum load conditions cannot be obtained and if oversized, then adequate control cannot be maintained especially at low load conditions. Designers typically walk this tightrope every day and apply safety factors to loads and valve sizes to avoid any under sizing ramifications.

## CONTROL VALVE AUTHORITY

Valves will be exposed to the lowest differential pressure when wide open and the highest DP when throttling and near closed. Pressure dependent valves are sized by flow capacity, which is measured by the flow coefficient ($C_v$). The goal is to select a valve with a $C_v$ that will offer good control and capacity. Once the correct $C_v$ is found (see Equation 3.2), the appropriate valve may be chosen from a manufacturer's catalog. Good control $C_v$ should fall between 70% and 90% of the control valve's maximum $C_v$ capability. The ratio of the maximum $C_v$ of the valve per the maximum $C_v$ required should never exceed 1.6. Severe oversizing can accelerate wear of the stem packings and the erosion of the valve trim due to control valve cycling and continuous valve operation near the seat thus shortening the life of the valve.

$$Q = C_v\sqrt{P_1 - P_2} \quad \text{or} \quad C_v = Q/\sqrt{\Delta P} \tag{3.2}$$

where

$Q$ = volumetric flow rate, gal/min (m$^3$/hr)
$P_1$ = upstream pressure at design flow, lbf/in$^2$, (bar)
$P_2$ = downstream pressure at design flow, lbf/in$^2$, (bar)
$\Delta P$ = $P_1 - P_2$ at design flow, lbf/in$^2$, (bar)
$C_v$ = flow coefficient = 1 gal/min with 1 lbf/in$^2$ pressure drop
(For SI units, $K_v$ = 1 m$^3$/hr at 1 bar pressure drop)

Pressure dependent control valves must have high "authority" characteristics, so they influence the flow in their circuit. Control valve pressure drop should be at least equal to the cooling coil pressure drop (5 to 10 psi) when wide open. The old rule of thumb sizing the control valve pressure drop to equal the cooling coil pressure drop should not be followed since many cooling coils today have very low pressure drops. As stated, it is best to keep the control valve pressure drop between 5 to 10 psi (wide open) and greater than the coil pressure drop. Equal percentage globe valves or characterized or reduced port ball valves are the best for this use. Globe and ball valves are used successfully due to their high valve authority. Double seated valves should not be used since they do not provide tight shut-off. ASHRAE (2016) defines control valve authority as:

$$\text{Authority} = \frac{\text{Differential pressure of the control valve}}{(\text{Differential pressure of the control valve/Differential pressure of the branch})}$$

where pressure drop across the control valve is defined from:

$$\sqrt{\Delta}P = Q/C_v \tag{3.3}$$

The maximum value for Authority is 1.0, however, that unrealistically assumes that the coil has zero pressure drop and the control valve is taking the entire pressure drop of the branch. Therefore, optimal values for Authority are typically around 0.5 or 50%. Hegberg (2000) also highlights the importance of high valve Authority by pointing out that valves selected with a smaller Authority result in larger valves, and vice versa. Furthermore, valves selected with lower Authorities provide less linear control output with coil heat transfer characteristics making control more difficult. One of the best features of a PICV is that it mimics the flow characteristics of a valve with an Authority of 1.0 linearizing the control response with the coil characteristic curve.

Similarly, Taylor (2002) recommends eliminating balancing valves on all terminal units in variable flow systems since they are detrimental to the performance of the system since they reduce the valve authority of the control valve and add permanent restrictions in every branch.

## ACTUATOR SIZING

Improperly-sized control valve actuators may lack the strength to close off the valve completely at the higher differential pressures experienced for circuits close to the pump when the load on that circuit drops to zero, or even to sufficiently close for low loads. Properly selected control valves should have a properly selected actuator in order for correct operation. The system designer should also take into account the pressure of the compression tank as well as pump shutoff pressure. Generally, valves and actuators should be sized to close against 1.25 to 1.5 times the design pump head or the pump dead head pressure as recommended in Siemens (2006).

Unfortunately, typical 2-way, two position control valve actuators used on fan coils have a maximum close-off pressure of 40 psig (276 kPa) or less and a high close-off pressure of 50 psig (345 kPa) or less. The low close off pressures lead to valves not being able to tightly seal against the system pressure. Again, in order to have higher leaving water temperatures, the control valve must modulate to vary the flow based on load demands. Lower quality valves with lower quality actuators should not be used.

As stated earlier, wherever possible, valves and actuators should be specified and purchased as a single unit so that the quality can be controlled, and any trouble shooting will minimize finger pointing of responsibilities. Additionally, purchasing as a single unit ensures the proper testing of the valve assembly for the close-off pressure rating. Both valve and actuator shall have high levels of turn down in order to control flows at lower loads. Furthermore, proportional–integral–derivative (PID) control capability on coil leaving air temperatures can overcome substantial oversizing where equal percentage valves are utilized since a slower operating speed can be programmed into the controller.

## BEST PRACTICES FOR INCREASING CHILLED-WATER RETURN TEMPERATURES AND SYSTEM $\Delta T$

In summary, the best practices for control valve selection to improve low chilled water $\Delta T$ are:

- Use only two-way modulating control valves and eliminate all 3-way control valves.
    - For low flow control for pumps, add 2-way control bypass valves around pump or a automatic flow control valve at the end of the system to open when system pressure gets high signifying closing of all control valves.
- Avoid two-position (open/close) control valves for fan coils.
- Use valves with equal percentage characteristics (equal percentage of valve travel produce equal percentage changes in flow).

  - Linear plug valves do not control as well as equal percentage valves.
  - An equal percentage valve has a near linear performance at high flow rates. At low flow rates, the valve travel is greater than flow change percentage, hence giving better controllability.
  - Use a high quality PICV or pressure dependent characterized ball valve.
- Select a valve with high rangeability to ensure predictable minimum flow through the control valve at low loads (minimum of 40 to 100:1 is preferred).[4]
- Each coil control valve should be sized for high-pressure drop in relation to the coil and the branch piping to maintain good control authority.
  - Rules-of-thumb use a minimum of 10 to 30% of the total system static pressure and the full open pressure drop should be at least 50% of[5] the total available branch circuit differential pressure under all operating conditions.
  - This will insure a high control authority in order to influence the flow in the circuit.
  - If the pressure differential is large, then multiple units should be considered.
- Don't undersize valve actuators. The valve actuator should be sized large enough to overcome this large system pressure differential to provide adequate shut-off, so valves are not lifted from seats, thus degrading the return temperature.
- Control valves should not be oversized. Unless control valves are PICV, they are typically smaller than line size.
  - At low load conditions an oversized valve will be cracked (i.e., off their seat) a fraction of an inch. This small control range is very hard to control.
  - It is better to undersize the valve so that the throttle range is larger for low load performance and even provide multiple valves for low flow and high flow conditions if warranted.
- All control valves should fully close upon de-energizing of air handling unit to minimize water bypass.
- Valves should be controlled with PID loop control sequence. If possible, monitor leaving water temperature at all units to ensure that all system components are functioning properly (dirty coil, fouling, air leakage, etc.).
- Vent coils to make sure they are not air bound.
- Clean coils and keep filters clean.
- Blow down strainers at each terminal unit to confirm the control valve and strainer flows are not obstructed preventing the valve to close all the way or a plugged strainer is blocking water flow.

## ALTERNATIVE METHODS TO INCREASE RETURN WATER TEMPERATURES

While chances are high that buildings will have a low chilled water return temperature due to issues discussed above, there are several things HVAC designers can implement to mitigate this occurrence by making the chilled water "work harder" within the building. While many of these design additions only have a minor impact on increasing the return water temperature, by operating together they may have more of an impact.

4. Rangeability is the ratio of its maximum and minimum controllable flow rates.
5. Control valve must have a high enough pressure drop (Authority) to influence fluid flow in the branch circuit. Authority is the ratio of the minimum to maximum differential pressures of the valve's control range.

In lieu of using chilled-water supply piped to cooling coils, chilled water return can be used to condition areas that are permitted be above the temperature thresholds of occupied areas such as electric rooms and mechanical rooms. A small circulation pump should be used to pump the return water through fan coils to compensate for the added pressure drop.

In addition, before return water is returned to the DCS provider it may be used to cool incoming outside air being used to meet ventilation requirements in, for example, buildings with dedicated outdoor air systems. In low humidity climates like Riyadh, the return temperatures are sufficient to bring the temperature of the outside air to around 72°F (22°C). Return water can be used as a first stage cooling in areas where high humidity is prevailing in summer, for example in climates similar to Dubai.

Similarly, chilled water return can be used to

- cool domestic water supply,
- cool swimming pools that are hot enough to be jacuzzi tubs,
- cool process load such as the compressors for walk in freezers and refrigerators in large kitchens, and
- cool remote spaces or other process loads with a water source heat pump using the chilled-water return piping as the heat sink.

## REFERENCES


AHRI. 2001. AHRI Standard 410-2001, *Standard for forced-circulation air-cooling and air-heating coils*. Arlington, VA: Air-Conditioning, Heating, and Refrigeration Institute.

ASHRAE. 2016. Chapter 47, Values. In *ASHRAE Handbook—Systems and equipment*. Atlanta: ASHRAE.

Hegberg, M.C. 2000. Control valve selection for hydronic systems. *ASHRAE Journal* (November):33–39.

Peterson, K. 2017. Considerations for selecting modulating control valves. *ASHRAE Journal* (February):56–61.

Taylor, S.T., 2002. Degrading chilled water plant Delta-T causes and mitigation. ASHRAE Symposium AC-02-6-1.


## BIBLIOGRAPHY


Fiorino, D.P. 1996. Twenty-five ways to raise your chilled water temperature differential. *ASHRAE Transactions* 102 (1).

Fiorino., D.P. 1999. Achieving high chilled water Delta Ts. *ASHRAE Journal*, November.

Hegberg, R. A. 1998. Application of control valves and balancing valves in a variable-flow hydronic system. *ASHRAE Transactions* 104(1).

Hyman, Lucas B., 2004. Overcoming Low Delta T, Negative Delta P at Large University Campus. ASHRAE Journal February.

Lizardos, E.J. 1994. Optimizing delta-T. *Engineered Systems*, September.

Sauer, J.M. 1989. Diagnosing low temperature differential—Does your chilled water system have this disease? *ASHRAE Journal*, June.

Siemens, 2006. Pressure-Independent Control Valves vs. High-Performance Control Valves. Balancing and Control Valve Sizing for Direct-Return Variable-Flow Hydronic Systems. November Technology Report.

Spalding, D.J. Causes and cures of low differential temperature in chilled water systems. International District Heating & Cooling Association, 7th Annual Cooling Conference.

# Appendix A
# Plant Efficiency Impacts from Low $\Delta T$ at Customers

There are a number of impacts of low $\Delta T$ at the customers' buildings that are felt at the DCS plant, including:

- extra pumping energy,
- removal of the extra pumping energy from the chilled water,
- chiller efficiency degradation at lower average operating temperature,
- increased condenser water pumping and cooling tower fan energy,
- increased cooling tower water usage, and
- increased heat gain to the chilled-water distribution system due to the lower chilled-water return temperature.

It's possible to quantify these impacts for a typical DCS by making a few assumptions on the performance of the central plant and the distribution system. For a modern, efficient, electric-driven compression refrigeration chiller plant, the overall specific energy consumption of the plant is normally around 1.0 kW of electric energy use per ton of delivered chilling effect, or in the common shorthand 1.0 kW/ton (0.28 $kW_e/kW_t$). The breakdown of the energy consumption of the major components in the plant will vary given system specifics, but a typical system might have the following breakdown:

| | | |
|---|---|---|
| Chiller energy input | = | 0.70 kW/ton (0.199 $kW_e/kW_t$) |
| Chilled-water distribution pumps | = | 0.15 kW/ton (0.043 $kW_e/kW_t$) |
| Condenser water pumps | = | 0.075 kW/ton (0.0213 $kW_e/kW_t$) |
| Cooling tower fans | = | 0.075 kW/ton (0.0213 $kW_e/kW_t$) |

With these assumptions, first let's quantify the impact of low $\Delta T$ that arises from additional pumping energy that must be expended to circulate the additional volume of water. For our "Low Chilled-Water Return Temperature" example in Chapter 1, twice the amount is needed. As noted above, pumping of the chilled water might represent about 0.15 kW/ton (0.043 $kW_e/kW_t$) in a typical DCS, thus simply doubling the pumping power would increase this to 0.30 kW/ton (0.0853 $kW_e/kW_t$). Unfortunately, the penalty for doubling the flow is significantly worse when the pipe diameters are fixed, since the frictional pressure drop is roughly proportional to the square of the flow velocity and that pressure drop is multiplied by the flow rate to find the frictional pumping power; thus, pumping power varies with the cube of the flow rate. If the flow rate is doubled the pumping power will be eight times greater. For the example in Chapter 1, it would increase from 0.15 kW/ton to 1.2 kW/ton (0.043 $kW_e/kW_t$

to 0.34 $kW_e/kW_t$)—an increase of 1.05 kW/ton (0.299 $kW_e/kW_t$). A major portion of this increase will end up in the chilled water as heat that must be removed by the chillers, thus furthering this impact, as discussed below.

To quantify the low $\Delta T$ impacts on chiller efficiency degradation, Phetteplace et al. (2019) indicates that for each 1°F (0.6°C) that the chilled-water temperature is decreased, a 2% increase in chiller energy consumption can be expected. Thus, for the building with low-$\Delta T$ syndrome in the Chapter 1 example, the *average* chiller operating temperature is decreased from 8°F (4.4°C) above chilled-water supply temperature to only 4°F (2.2°C), a 4°F (2.2°C) reduction in average chiller operating temperature. At the 2% increase per 1°F (0.6°C) drop in temperature, this would mean an 8% increase in chiller energy consumption. This is an enormous chiller energy use penalty for the DC provider. The amount of chiller energy use penalty can be estimated by looking first at the energy consumption of the chiller per ton of cooling generated. For our assumed chiller plant parameters above of 0.70 kW/ton (0.199 $kW_e/kW_t$), this low $\Delta T$ of only 4°F (2.2°C) would represent an additional 0.056 kW/ton (0.016 $kW_e/kW_t$) of chiller energy consumption, bringing total chiller energy consumption to 0.756 kW/ton (0.215 $kW_e/kW_t$).

As noted above, a portion of the extra pumping power input into the system due to increasing the flow rate ends up as extra heat that the chillers must remove. However, some portion of the additional pumping energy is also lost to electric motor inefficiencies and does not end up in the chilled water. Thus, the additional pumping power must be multiplied by the efficiency of the electric motor driving the pump; 95% would be a typical value for motors of the scale used to drive chilled-water pumps of a DCS. Thus, the chillers, now operating at the lower efficiency due to the low $\Delta T$, will need to remove an additional 1.14 $kW_t$ (1.20 kW/ton × 0.95) of heat from the chilled water for every ton of chilling effect that the plant delivers. For simplicity, this thermal energy is expressed as $kW_t$ rather than in refrigeration tons, thus the electric energy impact is easily arrived at by multiplying this thermal energy amount by the chiller specific performance in SI units of 0.215 $kW_e/kW_t$, yielding 0.245 kW/ton (0.0697 $kW_e/kW_t$) of additional energy consumption by the chillers, bringing their total energy consumption to 1.00 kW/ton (0.285 $kW_e/kW_t$).

Yet another impact of the low $\Delta T$ will be felt in the energy consumption on the condenser water side of the chillers, i.e., the condenser water pumps and the cooling tower fans. The increase in their energy consumption is easily determined by looking at the increase in chiller-specific energy consumption from that of the design situation of 0.70 kW/ton to 1.00 kW/ton (0.199 $kW_e/kW_t$ to 0.284 $kW_e/kW_t$) for the DCS operating with low $\Delta T$, a 43% increase. Thus, the specific energy consumption of both the condenser water pumps and the cooling tower fans will be increased by a similar percentage from 0.075 kW/ton to 0.107 kW/ton (0.0213 $kW_e/kW_t$ to 0.0304 $kW_e/kW_t$).

Now we can compute the total specific energy consumption of the central plant would be for the DCS operating with low $\Delta T$, using these parameters:

| | | |
|---|---|---|
| Chiller energy input | = | 1.00 kW/ton (0.285 $kW_e/kW_t$) |
| Water distribution pumps | = | 1.20 kW/ton (0.34 $kW_e/kW_t$) |
| Condenser water pumps | = | 0.107 kW/ton (0.0304 $kW_e/kW_t$) |
| Cooling tower fans | = | 0.107 kW/ton (0.0304 $kW_e/kW_t$) |

The total specific energy consumption of the chiller plant now increases from the 1.0 kW/ton (0.28 $kW_e/kW_t$) of the design case to 2.41 kW/ton (0.685 $kW_e/kW_t$), a 141% increase. However, there is yet one more impact of a DCS operating at low $\Delta T$ because of the lower return water temperature in the distribution system and that is increased heat gain from the ground to the buried return line piping system. Obviously, this will be much greater where return lines are uninsulated. Furthermore, for systems that have low load densities (e.g., smaller buildings that are spread out (for example, a college campus) or large systems that are not "built out" yet), heat gains will also be a much larger percentage of the central plant output. For the purposes of this example, we'll assume a well-insulated system with a typical load density might have 5% of plant output in distribution system heat gains. While heat gains into the system will not vary significantly with loading on the system, i.e., times of peak cooling load versus times of lower loads, there will be some seasonal variations due to varying soil temperatures; see Phetteplace et al. (2019) for a complete treatment of heat gains to DCS piping systems. If the average annual soil temperature is assumed to be 70°F (21°C), the relative relationship between the supply line and return line heat gains for the design condition would be

Supply line fraction = (70 – 40)/([70 – 40] + [70 – 56]) = 68%
Return line fraction = (70 – 56)/([70 – 40] + [70 – 56]) = 32%

For the case where the DCS is suffering from low $\Delta T$, the return line temperature is decreased and the fractions become

Supply line fraction = (70 – 40)/([70 – 40] + [70 – 48]) = 58%
Return line fraction = (70 – 56)/([70 – 40] + [70 – 48] = 42%

In the low-$\Delta T$ case, the return line heat gain as a percentage of the supply line heat gain is

Return heat gain, percent of supply = 42/58 = 72%

The supply line heat gain is assumed constant for both cases (it will perhaps be slightly lower in the case of low $\Delta T$ due to its proximity to the now cooler return line) and so the ratio of the total heat loss with low $\Delta T$ compared to that for the design condition will be:

Heat gain with low $\Delta T$ relative to design = (0.68 × (1 + 0.72))/(0.68 +0.32) = 1.17

And thus, the total heat gain to the distribution system of a DCS operating with low $\Delta T$ will be 1.17 times that of the design case, or 5.85%. And so finally if we look at the net specific energy consumption for each unit of cooling effect actually delivered, we have

For the design case = (1.0 kW/ton)/0.95 = 1.053 kW/ton (0.299 $kW_e/kW_t$)

For the case of the DCS operating with low $\Delta T$ = (2.41 kW/ton)/0.9415 = 2.56 kW/ton (0.728 $kW_e/kW_t$)

In summary, for this example we find that the DCS operating with low $\Delta T$ consumes 143% ([2.56 – 1.053]/1.053) more energy per net unit of cooling delivered than it would if operating as designed. A plot of the data from this example is provided as Figure 1.3.

Obviously, this result is specific to the assumptions of this example case, assumptions that we feel are representative for the conditions noted. However, the methodology that has been laid out could easily be used for a DCS operating under much different conditions.

## REFERENCE

Phetteplace, G., S. Abdullah, J. Andrepont, D. Bahnfleth, A. Ghani, B. Kirk, V. Meyer, and S. Tredinnick. (2019). *District cooling guide,* 2d ed. Atlanta: ASHRAE.

# Appendix B
# Case Study in Mitigation of Low $\Delta T$

*The following case study was contributed by Hassan Younes, Griffin Project Development Consultant, Dubai, United Arab Emirates. It is repeated nearly verbatim, with minor editorial changes to ensure conformity with the style of this book, from a case study performed for a client of the author's firm.*

## CHILLED-WATER SYSTEM AND LOW-$\Delta T$ SYNDROME DESCRIPTION

The building cooling load is catered by a district cooling plant (DCP). The building is connected to the chilled-water network through an energy transfer station (ETS). The ETS is located in the basement where six plate heat exchangers (PHEXs) connected to the chilled-water distribution circuit feed the building with chilled water and one additional PHEX feeds the condenser circuit of the cold room.

Three duty, variable-speed, chilled-water pumps and one standby pump circulate chilled water to serve outdoor air-handling units (OAHU), air-handling units (AHUs), fan coil units, pool heat exchangers, and cold potable water heat exchangers. In addition, one duty and one standby chilled-water pump feed chilled water to the condensing unit (CU) in the kitchen cold rooms.

All AHUs/OAHUs are provided with two-way modulating control valves that modulate the chilled-water flow through the coils to maintain space temperature/supply air temperature. In response to the varying differential pressure, the speed of the chilled-water pumps modulates to maintain a differential pressure set point between the chilled-water supply and return risers.

The building fan coil units (FCUs) are controlled via a room thermostat that controls the cooling coil's two-way valve and fan speed. The thermostat is also connected to the building monitoring system (BMS) for control and monitoring.

The chilled-water system pressure is maintained by the pressurization unit for the secondary circuit. The unit also provides necessary makeup water required for the chilled-water system.

The total capacity for heat exchangers 1 to 6 is 16,092 kW (4575 tons). The heat exchangers design $\Delta T$ is 16°F (8.9°C) and the flow to load ratio is 1.5 gpm/ton. The flow at maximum capacity is 6830 gpm (25,670 lpm), including the standby heat exchanger, which was operational during our first visit to the hotel.

The contracted value from the DC service provider is 3652 tons (12,843 kW).

The chilled-water pumps are all identical with a design flow of 2546 gpm (9638 lpm) each. The cooling capacity based on the pumps flow is 5830 tons (20,503 kW).

Because of the oversizing of every element in the system and other issues that will be indicated in the following paragraphs, the $\Delta T$ on the DC provider side and the building side is lower than design conditions.

As per the contract between the DC service provider and the hotel, for a $\Delta T$ below 7°C (12.6°F), the DC service provider will impose a penalty on the bill that is equal to 10% of the consumption at the time where $\Delta T$ was below 7°C (12.6°F). Below 5.5°C, (9.9°F) the penalty increases to 15% as per the table in Figure B.1, which was extracted from the contract.

As per the $\Delta T$ log for the last eight months and the hourly readings taken during our investigations, $\Delta T$ < 7°C (12.6°F) occurs 16% of the time and affects almost 12% of the load. The penalty estimation per year = yearly consumption charges × 12% × 10% = 5,000,000 × 12% × 10% = 60,000 AED/year (~16,333 USD/year).

Based on the multiple visits we made to site and the daily and hourly readings of the DC service provider's BTU meter, the peak cooling load recorded was 1860 ton. Note that this load includes a 128 ton cold-store load, which is being fed by a separate pump and a separate heat exchanger.

The peak load excluding the cold store is thus around 1750 tons (6154 kW).

At 1750 tons and with a $\Delta T$ of 16°F (8.9°C), the correspondent flow from the pumps should be 2625 gpm (9940 lpm).

One pump should be able to cater for the load. Note that in parallel pumping when only one pump is running its correspondent flow is 10% more than when it is running with the other two pumps because of lower pressure drop.

The main reason of low $\Delta T$ aside from oversizing is the wrong fan coil unit selection. The fan coil units are selected for a DT of 7.22°C (13°F) and a chilled-water velocity of 0.73 m/s (2.4 ft/s). The total FCU flow and load is 249 L/s (3944 gpm) and 2135 tons, respectively.

The AHUs' chilled-water design flow is 71 L/s (1126 gpm) for a correct design $\Delta T$ of 9°C (16°F). The OAHUs' chilled-water design flow is 130.9 L/s (2078 gpm) for a design $\Delta T$ of 12°C (21.6°F).

2. DELTA T:

| Delta T Range | Penalty* |
|---|---|
| 7.0 > delta T > or = 5.5 | 10% of consumption |
| 5.5> delta T >or = 4.4 | 15% of consumption |
| 4.4 > delta T > or = 3.3 | 25% of consumption |
| 3.3 > delta T | 40% of consumption |

*during period of default

**Figure B.1**

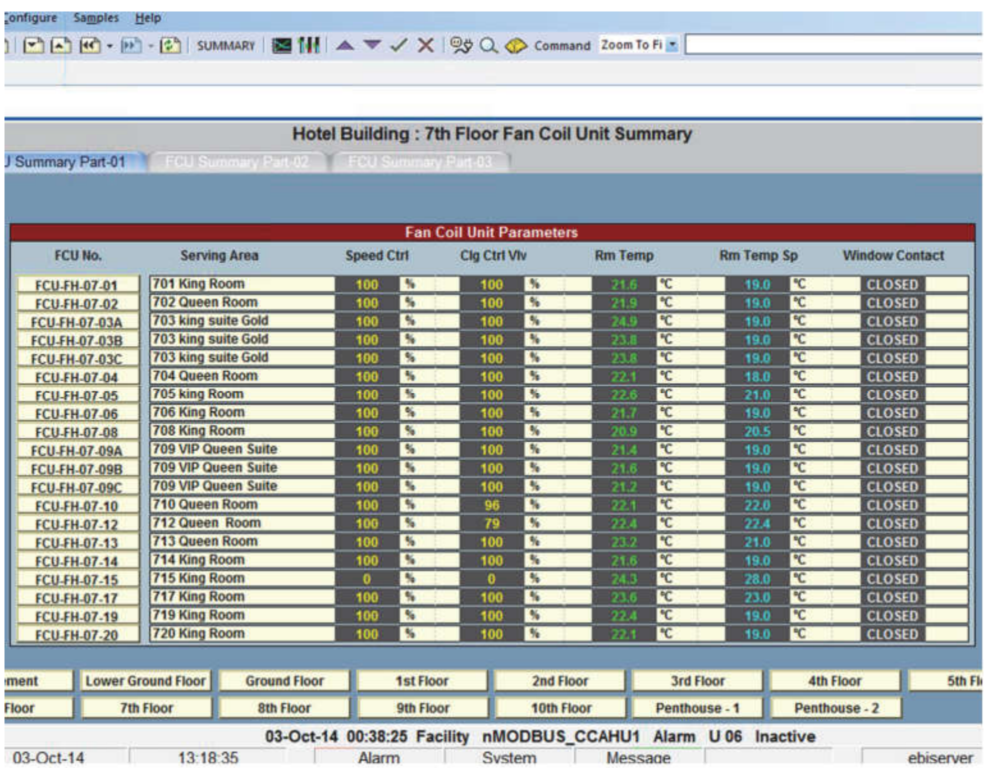
Total design flow for the FCUs, AHUs, and OAHUs is 450.9 L/s (7147 gpm).

For cooling coils and heat exchangers to perform properly, the flow should not be in laminar flow condition. The velocity of the chilled water inside the coil should not go below 0.3 m/s (1.0 ft/s), i.e., the flow to the FCUs should not go below 1645 gpm (assuming all FCU cooling valves are fully open, which is the case for most of the time).

Due to the incorrect set points of 19°C (66°F) in most of the hotel rooms, the FCUs valves are at 100% most of the time (see Figures B.1 and B.2). In Figures B.2 and B.3, most cooling valves are at 100%. The temperatures of rooms 901, 902, and 903 are relatively high. The reason that those rooms at that particular time have high temperatures could be due to many reasons, including to but not limited to low flow of chilled water, clogged strainers, air entrapment, etc.

Whenever the engineering team receives complaints from those rooms' guests, they tend to increase the speed of the pumps, leading to a lower $\Delta T$. The temperature of the room might not necessarily increase or decrease. It was requested that building automation system supplier train the supervisors on trending analysis, which means that the supervisors would look at the history of the room temperature and determine if the reason is low flow or some other reason and come up with the correct measure instead of relying on one measure which would be increasing the chilled-water pumps' speed.

In Figures B.2 and B.3, it is clear that most of the set points are below 23°C (73°F) and that no room temperature reaches below 21.4°C (70.5°F). In Figure B.2, most of the set points are 19°C (66°F) and all occupied rooms' cooling valves are at 100%. Table B.1 shows the effect of lower set points on $\Delta T$ (EDR 2009). With unrealistic set points, $\Delta T$ degrades quickly.




| FCU No. | Serving Area | Speed Ctrl | | Clg Ctrl Vlv | | Rm Temp | | Rm Temp Sp | | Window Contact |
|---|---|---|---|---|---|---|---|---|---|---|
| FCU-FH-07-01 | 701 King Room | 100 | % | 100 | % | 21.6 | °C | 19.0 | °C | CLOSED |
| FCU-FH-07-02 | 702 Queen Room | 100 | % | 100 | % | 21.9 | °C | 19.0 | °C | CLOSED |
| FCU-FH-07-03A | 703 king suite Gold | 100 | % | 100 | % | 24.9 | °C | 19.0 | °C | CLOSED |
| FCU-FH-07-03B | 703 king suite Gold | 100 | % | 100 | % | 23.8 | °C | 19.0 | °C | CLOSED |
| FCU-FH-07-03C | 703 king suite Gold | 100 | % | 100 | % | 23.8 | °C | 19.0 | °C | CLOSED |
| FCU-FH-07-04 | 704 Queen Room | 100 | % | 100 | % | 22.1 | °C | 18.0 | °C | CLOSED |
| FCU-FH-07-05 | 705 king Room | 100 | % | 100 | % | 22.6 | °C | 21.0 | °C | CLOSED |
| FCU-FH-07-06 | 706 King Room | 100 | % | 100 | % | 21.7 | °C | 19.0 | °C | CLOSED |
| FCU-FH-07-08 | 708 King Room | 100 | % | 100 | % | 20.9 | °C | 20.5 | °C | CLOSED |
| FCU-FH-07-09A | 709 VIP Queen Suite | 100 | % | 100 | % | 21.4 | °C | 19.0 | °C | CLOSED |
| FCU-FH-07-09B | 709 VIP Queen Suite | 100 | % | 100 | % | 21.6 | °C | 19.0 | °C | CLOSED |
| FCU-FH-07-09C | 709 VIP Queen Suite | 100 | % | 100 | % | 21.2 | °C | 19.0 | °C | CLOSED |
| FCU-FH-07-10 | 710 Queen Room | 100 | % | 96 | % | 22.1 | °C | 22.0 | °C | CLOSED |
| FCU-FH-07-12 | 712 Queen Room | 100 | % | 79 | % | 22.4 | °C | 22.4 | °C | CLOSED |
| FCU-FH-07-13 | 713 Queen Room | 100 | % | 100 | % | 23.2 | °C | 21.0 | °C | CLOSED |
| FCU-FH-07-14 | 714 King Room | 100 | % | 100 | % | 21.6 | °C | 19.0 | °C | CLOSED |
| FCU-FH-07-15 | 715 King Room | 0 | % | 0 | % | 24.3 | °C | 28.0 | °C | CLOSED |
| FCU-FH-07-17 | 717 King Room | 100 | % | 100 | % | 23.6 | °C | 23.0 | °C | CLOSED |
| FCU-FH-07-19 | 719 King Room | 100 | % | 100 | % | 22.4 | °C | 19.0 | °C | CLOSED |
| FCU-FH-07-20 | 720 King Room | 100 | % | 100 | % | 22.1 | °C | 19.0 | °C | CLOSED |

**Figure B.2**


**Figure B.3**

**Table B.1** Coil Performance (at Full Load) for Low LAT Set Points[1]

| Leaving Air Temperature (LAT) Set Point | gpm | $\Delta T$ | % of Design gpm |
|---|---|---|---|
| 54 | 80 | 13 | 100% |
| 53 | 104 | 11 | 130% |
| 52 | 143 | 8.5 | 197% |
| 51 | 208 | 6.5 | 260% |
| 50 | 327 | 4.3 | 409% |
| 49 | | Cannot be attained | |

*1 EDR (2009)*

During the peak load where the load is 1750 tons (6150 kW) and taking into account the 13°F (7.2°C) $\Delta T$ of the FCUs, i.e., 1.84 gpm/ton, the flow required is 3220 gpm (12,190 lpm). The flow fed to the FCUs is assumed to be 50% of the total flow, where 30% is feeding the FAHUs and 10% is feeding the AHUs. Considering most of the FCU cooling valves are fully open, at 1610 gpm (6094 lpm) the velocity in the FCUs cooling coils is 0.3 m/s (1 ft/s), i.e., at the laminar flow condition boundary. When the building cooling load becomes lower, i.e., 800 tons at 2:00 a.m., the flow fed to the FCUs would be 736 gpm (2790 lpm) and with most of the cooling valves open due to fully open cooling coil valves the velocity would be 0.136 m/s (0.446 ft/s). At these laminar conditions, the $\Delta T$ is degrading further as well. Thus it is imperative to increase the FCU set points that will lead to more closed cooling valves and laminar conditions avoided.

The main heat exchangers laminar condition is occurring below 30% of design flow (as per the email received from the supplier).

**Table B.2**

| Pump Speed % | Time | Pump Speed % | Time |
|---|---|---|---|
| 45 | 0:05 | 72 | 12:00 |
| 45 | 1:00 | 52 | 13:00 |
| 42 | 2:00 | 63 | 14:00 |
| 38 | 3:00 | 72 | 15:00 |
| 37 | 4:00 | 78 | 16:00 |
| 38 | 5:00 | 74 | 17:00 |
| 43 | 6:00 | 59 | 18:00 |
| 48 | 7:00 | 85 | 19:00 |
| 52 | 8:00 | 69 | 20:00 |
| 49 | 9:00 | 73 | 21:00 |
| 52 | 10:00 | 49 | 22:00 |
| 63 | 11:00 | 49 | 23:00 |

At a flow below 2050 gpm (7760 lpm), the heat exchangers' flow might go into laminar flow conditions, and heat transfer might deteriorate.

## ADDITIONAL NOTES

A bypass valve is located in the plant room to bypass return chilled water into supply chilled water. Previously it was recommended to close the bypass valve, which was the correct decision, because of its detrimental effect on chilled water and higher pumping power. In additional to the motorized shutoff valve, we requested that the hotel engineering team shut off the two manual butterfly valves to ensure no bypass occurs.

It was noticed that the room temperature reading on the room thermostats is different from the readings on the BMS. The AHUs chilled-water sensors were giving wrong\ readings and need to be attended to.

In addition to that, operations manual mentions that all chilled-water flowmeters are the immersion type, while on site most of the chilled-water temperature sensors are the strap-on type. Strap-on-type sensors have much lower accuracy than immersion type sensors, and most of those sensors on site are not properly fixed and read the room temperature instead of the chilled-water pipe temperature.

Some sensors in the FAHUs have wrong locations. One case is the temperature sensor shown on the BMS as located before the heat recovery wheel, when in reality it is located after the heat recovery wheel.

## TESTS

The consultant conducted many tests and, in coordination with the hotel engineering team, took records of $\Delta T$ and other parameters for several days following each measure. The BMS was also monitored daily for a period of two months.

The first prominent experiment was to produce a load schedule where the pumps' speed would modulate based on a prefixed schedule. The schedule was based on hourly readings for the building cooling load, based on seven days of cooling load hourly readings.

The percentage profile was based on the maximum seven days load of each hour divided by the peak load as per Table B.2.

When this measure was applied, the $\Delta T$ was always above 7.7°C (14°F) as per Table B.3.

**Table B.3**

| Date | Time | Pumps Speed | | | | DC ST | DC RT | ΔT |
|---|---|---|---|---|---|---|---|---|
| 5/9/2014 | 1:00 | 80 | 81 | 82 | off | 4.2 | 12.9 | 8.6 |
| 5/9/2014 | 2:00 | 80 | 81 | 82 | off | 4 | 12.7 | 8.6 |
| 5/9/2014 | 3:00 | 80 | 81 | 82 | off | 4 | 12.1 | 8.1 |
| 5/9/2014 | 4:00 | 80 | 81 | 82 | off | 4.1 | 11.7 | 7.6 |
| 5/9/2014 | 5:00 | 80 | 81 | 82 | off | 4 | 11.7 | 7.6 |
| 5/9/2014 | 6:00 | off | 44 | 81 | 44 | 3.9 | 11.8 | 7.8 |
| 5/9/2014 | 7:00 | off | 49 | 49 | 50 | 3.87 | 12.15 | 8.28 |
| 5/9/2014 | 8:00 | off | 52 | 53 | 53 | 3.7 | 11.9 | 8.2 |
| 5/9/2014 | 9:00 | off | 51 | 52 | 50 | 3.6 | 12.2 | 8.55 |
| 5/9/2014 | 10:00 | off | 52 | 53 | 53 | 3.7 | 12.3 | 8.5 |
| 5/9/2014 | 11:00 | off | 64 | 64 | 64 | 3.6 | 12.4 | 8.5 |
| 5/9/2014 | 12:00 | off | 64 | 64 | 64 | 3.9 | 12 | 8 |
| 5/9/2014 | 13:00 | 78 | 78 | 79 | off | 4.5 | 13.1 | 8.6 |
| 5/9/2014 | 14:00 | 73 | 74 | 74 | off | 4.6 | 12.75 | 8 |
| 5/9/2014 | 15:00 | 74 | 73 | 75 | off | 4.8 | 13.13 | 8.29 |
| 5/9/2014 | 16:00 | 85 | 85 | 85 | off | 4.6 | 13.3 | 8.68 |
| 5/9/2014 | 17:00 | 85 | 85 | 85 | off | 4.8 | 13.97 | 9.09 |
| 5/9/2014 | 18:00 | 59 | 59 | 59 | off | 5.23 | 14.39 | 9.17 |
| 5/9/2014 | 19:00 | 86 | 86 | 86 | off | 4.94 | 14.22 | 9.28 |
| 5/9/2014 | 20:00 | 69 | 69 | 69 | off | 4.93 | 14.3 | 9.37 |
| 5/9/2014 | 21:00 | 76 | 73 | 73 | off | 4.89 | 14.06 | 9.17 |
| 5/9/2014 | 22:00 | 49 | 49 | 49 | off | 4.86 | 13.64 | 8.77 |
| 5/9/2014 | 23:00 | 49 | 49 | 49 | off | 4.74 | 12.46 | 7.71 |
| 6/9/2014 | 0:00 | 50 | 50 | 51 | off | 4.41 | 12.37 | 7.97 |
| 6/9/2014 | 1:00 | 46 | 46 | 47 | off | 3.95 | 12.83 | 8.88 |
| 6/9/2014 | 2:00 | 43 | 46 | 43 | off | 3.94 | 12.25 | 8.31 |
| 6/9/2014 | 3:00 | 39 | 40 | 40 | off | 3.82 | 11.9 | 8.09 |
| 6/9/2014 | 4:00 | 38 | 38 | 38 | off | 3.77 | 11.49 | 7.73 |
| 6/9/2014 | 5:00 | 40 | 40 | 40 | off | 3.97 | 11.15 | 7.18 |
| 6/9/2014 | 6:00 | 39 | 44 | 44 | off | 3.94 | 12.06 | 8.12 |
| 6/9/2014 | 7:00 | 49 | 49 | 49 | off | 3.96 | 12.73 | 8.77 |
| 6/9/2014 | 8:00 | 49 | 49 | 49 | off | 3.8 | 12.8 | 9 |
| 6/9/2014 | 9:00 | 50 | 50 | 52 | off | 4.29 | 13.19 | 8.89 |
| 6/9/2014 | 10:00 | 52 | 52 | 52 | off | 4.73 | 12.78 | 8.04 |
| 6/9/2014 | 11:00 | 63 | 63 | 63 | off | 4.67 | 12.69 | 8.02 |
| 6/9/2014 | 12:00 | 72 | 72 | 72 | off | 4.42 | 12.9 | 8.47 |
| 6/9/2014 | 13:00 | 78 | 78 | 78 | off | 4.9 | 13.45 | 8.55 |
| 6/9/2014 | 14:00 | 74 | 74 | 74 | off | 4.87 | 14.14 | 9.27 |
| 6/9/2014 | 15:00 | 74 | 74 | 75 | off | 4.65 | 14.27 | 9.62 |
| 6/9/2014 | 16:00 | 85 | 85 | 85 | off | 4.72 | 14.27 | 9.55 |
| 6/9/2014 | 17:00 | 85 | 86 | 87 | off | 4.78 | 14.63 | 9.85 |
| 6/9/2014 | 18:00 | 79 | 79 | 80 | off | 4.9 | 14.6 | 9.7 |
| 6/9/2014 | 19:00 | 60 | 61 | 61 | off | 5 | 14.6 | 9.6 |
| 6/9/2014 | 20:00 | 70 | 70 | 71 | off | 4.9 | 14.5 | 9.6 |

**Table B.3** (continued)

| Date | Time | Pumps Speed | | | | DC ST | DC RT | $\Delta T$ |
|---|---|---|---|---|---|---|---|---|
| 6/9/2014 | 21:00 | 73 | 73 | 74 | off | 4.9 | 14 | 9.1 |
| 6/9/2014 | 22:00 | 51 | 50 | 52 | off | 4.7 | 13 | 8.3 |
| 6/9/2014 | 23:00 | 50 | 50 | 48 | off | 4.5 | 12.5 | 8.3 |
| 7/9/2014 | 6:00 | 47 | 44 | 47 | off | 4.21 | 12.7 | 8.49 |
| 7/9/2014 | 7:00 | 49 | 49 | 53 | off | 4.5 | 12.7 | 8.1 |
| 7/9/2014 | 8:00 | 53 | 52 | 56 | off | 4.9 | 12.8 | 7.9 |
| 7/9/2014 | 9:00 | 53 | 52 | 56 | off | 4.6 | 13.2 | 8.5 |
| 7/9/2014 | 10:00 | 52 | 52 | 56 | off | 4.7 | 13.2 | 8.5 |
| 7/9/2014 | 11:00 | 64 | 64 | 66 | off | 4.8 | 13.5 | 8.7 |
| 7/9/2014 | 12:00 | 72 | 72 | 74 | off | 4.87 | 13.4 | 8.6 |
| 7/9/2014 | 13:00 | 78 | 78 | 80 | off | 4.5 | 14 | 9.5 |
| 7/9/2014 | 14:00 | 75 | 74 | 76 | off | 4.55 | 14 | 9.4 |
| 7/9/2014 | 15:00 | 74 | 73 | 75 | off | 4.5 | 14.1 | 9.6 |
| 7/9/2014 | 16:00 | 85 | 85 | 87 | off | 4.55 | 14.38 | 9.03 |
| 7/9/2014 | 17:00 | 78 | 78 | 78 | off | 4.58 | 14.4 | 9.82 |
| 7/9/2014 | 18:00 | 59 | 59 | 59 | off | 4.7 | 14.6 | 9.9 |
| 7/9/2014 | 19:00 | 61 | 61 | 61 | off | 4.5 | 13.3 | 8.7 |
| 7/9/2014 | 20:00 | 69 | 69 | 69 | off | 4.2 | 13.3 | 8 |
| 7/9/2014 | 21:00 | 73 | 73 | 73 | off | 4.2 | 13.3 | 9 |
| 7/9/2014 | 22:00 | 50 | 50 | 51 | off | 4.3 | 12.8 | 8.4 |
| 7/9/2014 | 23:00 | 49 | 49 | 49 | off | 4.42 | 12.81 | 8.39 |
| 8/9/2014 | 0:00 | 47 | 47 | 48 | off | 4.34 | 13.1 | 8.76 |
| 8/9/2014 | 1:00 | 46 | 45 | 48 | off | 4.15 | 13.02 | 8.87 |
| 8/9/2014 | 2:00 | 46 | 46 | 48 | off | 3.74 | 12.79 | 9 |
| 8/9/2014 | 3:00 | 40 | 39 | 40 | off | 4 | 12.42 | 8.42 |
| 8/9/2014 | 4:10 | 38 | 38 | 39 | off | 4.23 | 12.43 | 8.2 |
| 8/9/2014 | 5:00 | 39 | 39 | 40 | off | 4.63 | 12.59 | 7.96 |
| 8/9/2014 | 6:00 | 39 | 39 | 40 | off | 4.52 | 12.58 | 8.04 |
| 8/9/2014 | 7:00 | 48 | 49 | 50 | off | 4.21 | 12.82 | 8.62 |
| 8/9/2014 | 8:00 | 49 | 49 | 51 | off | 4.11 | 12.3 | 8.2 |
| 8/9/2014 | 9:00 | 49 | 50 | 51 | off | 4.2 | 13.2 | 9 |
| 8/9/2014 | 10:00 | 50 | 50 | 52 | off | 4 | 12.7 | 8.7 |
| 8/9/2014 | 11:00 | 53 | 52 | 55 | off | 3.8 | 13 | 9.1 |
| 8/9/2014 | 12:00 | 63 | 64 | 65 | off | 3.8 | 12.2 | 8.4 |
| 8/9/2014 | 13:00 | 73 | 72 | 75 | off | 3.8 | 13.1 | 9.3 |
| 8/9/2014 | 14:00 | 75 | 75 | 77 | off | 4.5 | 13.2 | 8.7 |
| 8/9/2014 | 15:00 | 74 | 74 | 75 | off | 4.65 | 14.27 | 9.62 |
| 8/9/2014 | 16:00 | 85 | 85 | 85 | off | 4.72 | 14.27 | 9.55 |
| 8/9/2014 | 17:00 | 85 | 86 | 87 | off | 4.78 | 14.63 | 9.85 |
| 8/9/2014 | 18:00 | 79 | 79 | 80 | off | 4.9 | 14.6 | 9.7 |
| 8/9/2014 | 19:00 | 60 | 61 | 61 | off | 5 | 14.6 | 9.6 |
| 8/9/2014 | 20:00 | 70 | 70 | 71 | off | 4.9 | 14.5 | 9.6 |
| 8/9/2014 | 21:00 | 73 | 73 | 74 | off | 4.9 | 14 | 9.1 |
| 8/9/2014 | 22:00 | 51 | 50 | 52 | off | 4.7 | 13 | 8.3 |

The other measure that was implemented is linking the pumps speed to the return temperature. The return temperature set point chosen was 14°C (57.2°F). With this measure the $\Delta T$ was above 7°C (13°F) most of the time except 2:00 a.m. to 6:00 a.m., where the $\Delta T$ reduced to below 7°C (13°F) but above 6°C (11°F). With changing the room set points it is anticipated that $\Delta T$ will not reduce below 7°C (13°F) using this measure.

It was also noted that the BMS operators were not very familiar with the BMS and have little technical knowledge in HVAC in general.

A presentation on how to use the trending function in the BMS was prepared by the building automation system (BAS) supplier and distributed to the BMS supervisors.

Final recommendation is to:

1. Set all room minimum set points to 23°C (73.4°F). The guest would still be able to reduce the thermostat set point to below 23°C (73.4°F) but in effect 23°C (73.4°F) will be the lowest set point as it is overridden by the BMS. This can be easily programmed on the BMS. 23°C (73.4°F) and 50% rh should produce a very high satisfaction rate between guests.
2. Once point 1 is achieved, any of the previous arrangements of differential pressure, return temperature, or a preset profile control can be used. It is to be noted that the BMS should be programmed so that at any time where differential pressure sensor reading reads above 35 psi (2.4 bar), pump speed shall not increase beyond the current speed.
3. Shutdown of additional heat exchangers during low cooling months to avoid laminar conditions.
4. A BTU meter on the main chilled-water pipes should be installed to update the profile seasonally. The BTU meter can also be used to check against the DC service provider's cooling consumption bills. The estimated cost for adding the BTU meter as per the building automation system supplier's service team is 30,000 AED (~8,200 USD).

Additional recommendations:

1. Renegotiation of the contract with DC service provider to reduce the capacity from 3652 to 2652 tons. The 2652 ton load is 40% above the highest peak recorded, which keeps a cushion of safety in case of any planned or unplanned load addition or hotel expansion. Reducing 1000 tons of capacity charges can lead to 750,000 AED (204,000 USD) savings a year.
2. Request that the hotel's management team employ a qualified engineer to operate the BMS and the HVAC systems in the hotel. The current staff simply do not have the required qualifications.
3. Another solution is to assign a company that can remotely oversee the BMS.
4. Perform chilled-water testing adjusting and balancing for all rogue rooms. Rogue rooms comprise of any room showing relatively high temperatures compared to other rooms, even when the pumps are at high speed.

## REFERENCE

EDR. 2009. *Design guidelines: CoolTools chilled water plant*. San Francisco, CA: Energy Design Resources, under the auspices of California Public Utilities Commission.

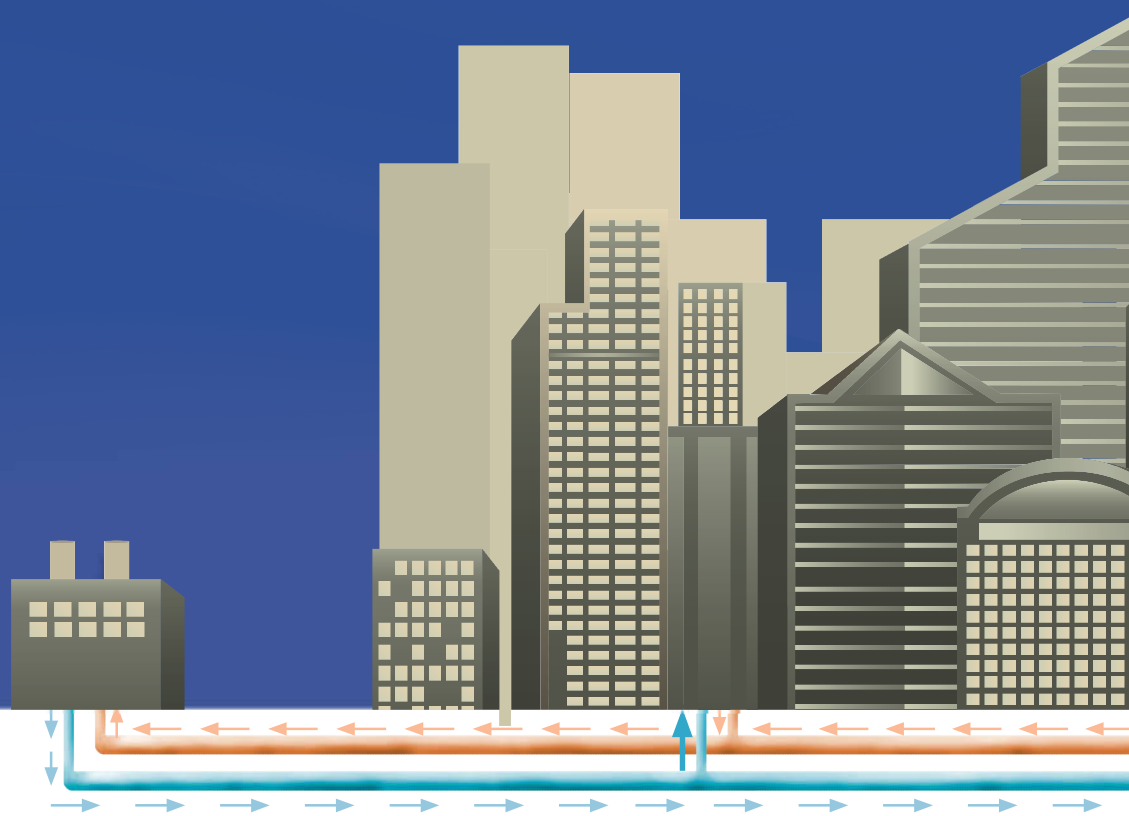
## Essential Guidance on District Cooling for Building Owners and Designers

*Owner's Guide for Buildings Served by District Cooling*, a companion guide to *District Cooling Guide*, Second Edition, provides essential information to both the building owner and the building designer on the advantages, installation, and operation of district cooling systems.

Owner-specific concerns, such as selecting a design consultant, responsibilities for in-building equipment maintenance, and district cooling tariff structures, are presented in easily understandable, nontechnical language. Designer-specific concerns, such as connection types, components, metering, and maintenance, are covered in detail.

The guide also dedicates a chapter to issues related to existing buildings, such as design deficiencies and other constraints that could prevent achieving acceptable system performance, and also includes appendices with case studies on performance issues, such as low-$\Delta T$ syndrome.

*Owner's Guide for Buildings Served by District Cooling* is a useful resource for building owners and designers interested in implementing district cooling in their buildings. Use alongside the *District Cooling Guide*, Second Edition.



ASHRAE
1791 Tullie Circle
Atlanta, GA 30329-2305
404-636-8400 (worldwide)
www.ashrae.org

ISBN 978-1-947192-26-3 (paperback)
ISBN 978-1-947192-27-0 (PDF)
9 781947 192263
Product code: 90565 5/19