

CHAPTER 12. DISTRICT HEATING AND COOLING

DISTRICT heating and cooling (DHC) or district energy (DE) distributes thermal energy from a central source to residential, commercial, and/or industrial consumers for use in space heating, cooling, water heating, and/or process heating. The energy is distributed by steam or hot- or chilled-water lines. Thus, thermal energy comes from a distribution medium rather than being generated on site at each facility.

According to Pierce (1994), district heating was proposed in 1613 in London; however, the first known examples were hot-water systems installed in St. Petersburg, Russia, in the 1840s. In the United States, the first district heating system was installed at the Naval Academy in Annapolis, Maryland, in 1853. Birdsall Holly is often credited with inventing district heating in Lockport, New York, in 1877, but he was simply the first to make it a commercial enterprise, ultimately expanding it to other cities. Most major U.S. cities soon were served by steam-based district heating systems; the majority of these systems still survive today, the largest in New York City. Although steam systems were also built in a number of major European cities, hot-water-based district heating saw significant growth in Europe, in part because of the reconstruction after World War II and the construction or expansion of many U.S. military bases, which were an ideal application for district heating. Both steam and high-temperature hot-water systems were built on bases. In Europe, especially Scandinavia, hot-water-based district heating systems adopted low temperatures for supply; that trend continues today, with systems using supply temperatures of 158°F or lower.

Early attempts at district cooling date back to the 1880s (Pierce 1994). By the 1930s, commercial systems were being built. Although recent development in district cooling had been confined mostly to the United States, there has been increased international activity, notably in the Middle East and in Europe. The International District Energy Association (IDEA) represents both heating and cooling utilities and reported that approximately 86% of the conditioned building space added in the last six years by its member systems was added outside of the United States, the vast majority of that growth was district cooling development in the Middle East (IDEA 2008a).

Whether the system is a public utility or user owned, such as a multibuilding campus, it has economic and environmental benefits depending somewhat on the particular application. Political feasibility must be considered, particularly if a municipality or governmental body is considering a DHC installation. Historically, successful DHC systems have had the political backing and support of the community. IDEA (2008a) has many applicable suggestions in developing and designing district cooling systems.

Applicability

District heating and cooling systems are best used in markets where (1) the thermal load density is high and (2) the annual load factor or operating hours are 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 for district heating systems (normally lower for district cooling applications). The annual load factor is important because the total system is capital intensive. These factors make district heating and cooling systems most attractive in serving (1) industrial complexes, (2) densely populated urban areas, and (3) high-density building clusters with high thermal loads such as mixed-use developments, airports, college and university campuses, and large office developments. Low-density residential areas have usually not been attractive markets for district heating, although there have been many successful applications in Scandinavia using low-temperature hot water (Zinko et al. 2008). District heating is best suited to areas with a high building and population density in relatively cold climates; however, using absorption chillers can also provide summer load for systems operating with high-enough water supply temperatures or steam pressures. District cooling applies in most areas that have appreciable concentrations of cooling loads, usually associated with taller buildings and mixed use developments. For information on business models and business development for district cooling enterprises, see IDEA (2008a).

Components

District heating and cooling systems consist of three primary components: the central plant, the distribution network, and the consumer systems (Figure 1). The central source or production plant may be any type of boiler; a refuse incinerator; a geothermal heat source; solar energy; heat pump with heat sourced from sewage, seawater, groundwater, ground coupling, surface water, or waste heat; or thermal energy developed as a by-product of electrical generation. The last approach, called combined heat and power (CHP), has a high energy utilization efficiency; see Chapter 7 for information on CHP.

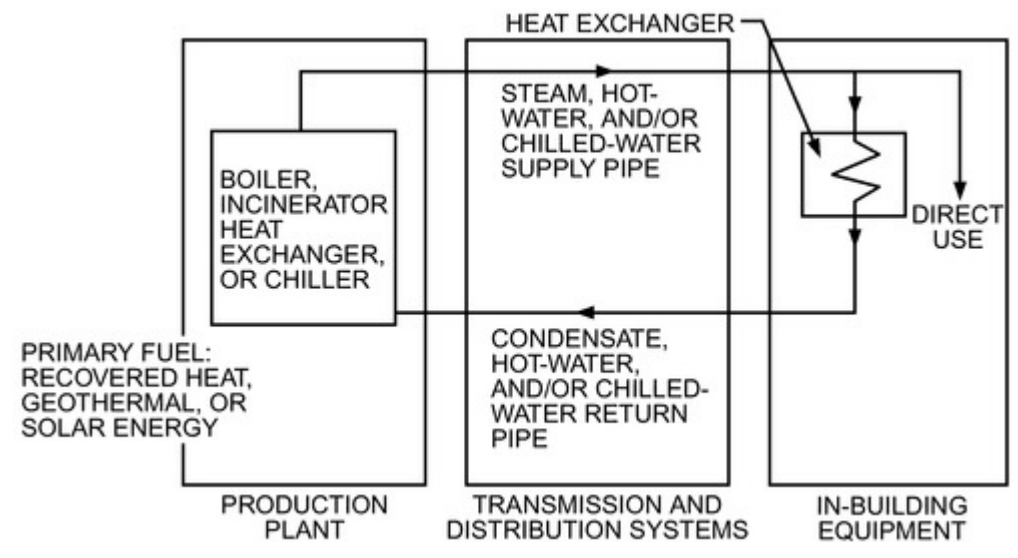


Figure 1. Major Components of District Heating System

Chilled water can be produced by

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

The second component is the **distribution or piping network** that conveys the energy. The piping is often the most expensive portion of a district heating or cooling system. The piping usually consists of a combination of preinsulated and field-insulated pipe in both concrete tunnel and direct burial applications. These networks require substantial permitting and coordinating with nonusers of the system for right-of-way if not on the owner's property. Because the initial cost is high in distribution systems, it is important to optimize its use.

The third component is the **consumer system**, which includes connections to the in-building equipment. When steam is supplied, it may be (1) used directly for heating; (2) directed through a pressure-reducing station for use in low-pressure (0 to 15 psig) steam space heating, service water heating, humidification, and absorption cooling; or (3) passed through a steam-to-water heat exchanger. When hot or chilled water is supplied, it may be used directly by the building HVAC systems or indirectly where isolated by a heat exchanger (see the section on Consumer Interconnections).

Environmental Benefits

Generating thermal energy from a central plant is normally more efficient than using in-building equipment (i.e., a decentralized approach) and thus reduces environmental impacts. The greater efficiencies arise because of 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. District energy plants also take advantage of diversity of demand across all users in the system, and may also implement some technologies (e.g., thermal storage) more readily than decentralized systems.

Emissions from central plants are easier to control than those from individual plants and, in aggregate, are lower because of higher-quality equipment, seasonal efficiencies, level of maintenance, load diversity, and lower system heat loss. Although today's political climate may not perceive new coal plants as desirable, existing central plants that burn coal can economically remove noxious sulfur emissions, where removal with individual combustors would not be cost effective. Furthermore, solid-fuel boilers also can burn other waste products, including biomass fuels and refuse. Similarly, the thermal energy and gaseous waste from municipal wastes can provide an environmentally sound system. Cogeneration of heat and electric power allows for combined efficiencies of energy use that greatly reduce emissions, decrease energy used, and allow for fuel flexibility. In addition, refrigerants 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 use of any more hazardous compounds such as ammonia for cooling systems.

1. SYSTEM MASTER PLANNING

Planning a "green field" district energy system or mapping out growth scenarios for an existing system takes a great deal of effort in gathering information regarding not only the customer base, but also plant sites, distribution routing, fuel sources and sensitivities, levels of redundancy, etc. This planning phase is typically referred to as **master planning**. Master plans take a long-term view of system development and growth, investigating many parameters and potential road blocks to development and growth. For a more complete explanation of the master planning process, see Chapter 2 of ASHRAE's *District Cooling Guide* and *District Heating Guide* (Pietheplace et al. 2013a, 2013b).

At a minimum, a 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 is usually because of at least one 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 master plan may be likened to a pyramid (Figure 2) (Bahrfleth 2004). The success of the plan depends on the foundation: a strong, accurate database that includes discovery and verification. All other aspects of a master plan stand on this basis and get their credibility from it. With the database in place, alternatives are identified and the preliminary estimates of cost (screening grade) for each option are used to work with the owner to select the most promising choices. These selections 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.

- Typical technical issues addressed in master plans include the following:
- Code and regulator implications
 - Phases of development and construction
 - Other uses of thermal energy (i.e., absorption chiller for steam system summer use)
 - Planned growth of system and potential of building use change
 - Siting of central plant (noise issues, survivability during severe weather event, flooding, aesthetics)
 - Topography
 - Fuel availability, days of on-site storage, fuel handling, and cost
 - Equipment selection (chillers, boilers, cooling towers, etc.)
 - Distribution system routing and material selection
 - Heating and cooling medium options (system temperatures and pressures)
 - Type of consumer interconnection (direct or indirect)
 - Metering technology and invoicing scheme

It is recommended that an experienced planner and designer be consulted in the early planning phase if those undertaking the master plan and design do not have wide experience with district cooling systems (DCS).

1.1 ECONOMIC CONSIDERATIONS

Consumer Economics

A district heating and cooling system offers many economic benefits. Even though the basic costs are still borne by the district energy provider (central plant owner/operator), the customer also benefits from a large, centralized system's economies of scale.

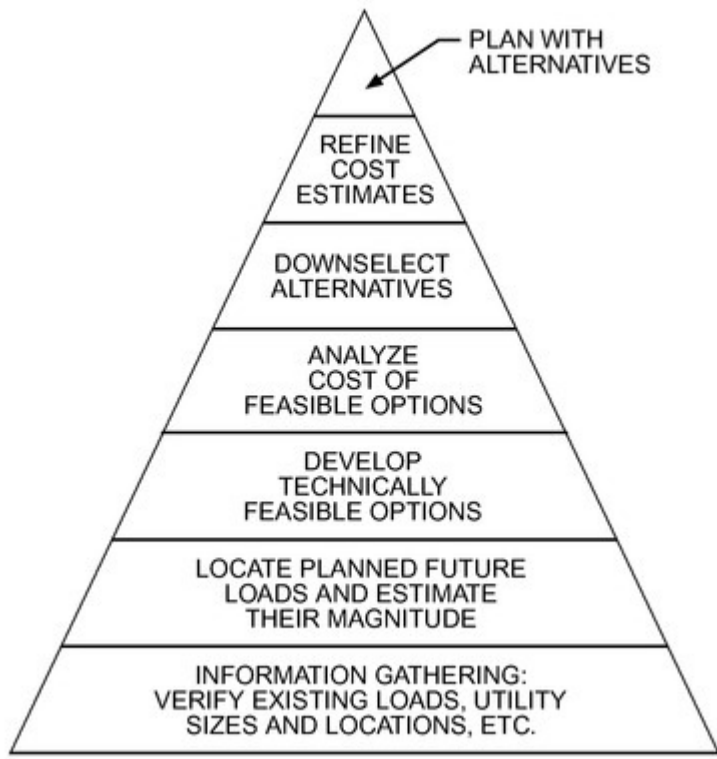


Figure 2. Master Planning Pyramid (bahrfleth 2004)

Operating Personnel. One of the primary advantages to 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 are in operation. Some older systems require trained operating personnel to be in the boiler/mechanical room at all times. When thermal energy is brought into the building as a utility, depending on the sophistication of the building HVAC controls, there may be opportunity to reduce or eliminate operating personnel.

Insurance. Both property and liability insurance costs are significantly reduced with the elimination of a boiler in the mechanical room, because risk of a fire or accident is reduced.

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

Equipment Maintenance. With less mechanical equipment, there is proportionately less equipment maintenance, resulting in less expense and a reduced maintenance staff.

Higher Thermal Efficiency. A larger central plant can achieve higher thermal and emission efficiencies than the sum of several smaller plants. Partial-load performance of central plants may be more efficient than that of many isolated, small systems because the larger plant can operate one or more capacity modules as the combined load requires and can modulate output. Furthermore, central plants take advantage of the fact that buildings that have different functions and their loads do not peak at the same time, but are diversified. With this system diversity, less equipment must be energized to serve the peak system load. Also, central plants generally have efficient base-load units and less costly peaking equipment for use in extreme loads or emergencies. Thornton et al. (2008) found that, although actual operating data on in-building cooling plants are scarce, the limited data available indicate that in-building systems operate at an average efficiency of 1.2 kW/ton (2.9 COP). Espeland (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 district plant might be estimated to be approximately 40% greater than that of an in-building chiller plant. However, IDEA (2008b) suggested much greater efficiency improvements relative to air-cooled, in-building cooling systems: approximately 1.65 kW/ton (2.1 COP) for air-cooled, in-building systems versus approximately 0.70 kW/ton (3.0 COP) for electrically driven district cooling with thermal storage, an efficiency increase of nearly 140% for district cooling. When strict regulations must be met, additional pollution control equipment is also more economical for larger plants than smaller plants. Cogeneration of heat and electric power results in much higher overall efficiencies than are possible from separate heat and power plants.

Lower Utility Rates. Buildings that connect to district energy systems, specifically for cooling, have flatter electrical load profiles once the chilled-water-generation equipment is removed from the metered equipment. Flatter electrical profiles are usually conducive to lower electrical rates, because the peaks are removed. Also, the fact that the building has no cooling towers drastically reduces the water and sewer utility bills (i.e., no makeup water and blowdown requirements). Similar benefits exist for gas usage or steam blowdown and makeup needs.

Reliability. DE systems are typically highly reliable and rarely have outages. Mature systems typically have reliability values greater than 99.98%.

Higher Level of Control and Monitoring. Typically, the level and quality of control and monitoring systems in DE systems are superior to commercial-grade HVAC systems in the customer's building, because the consumer's energy is tracked and invoices are prepared using the metered data.

Producer Economics

Available Fuels. Smaller heating plants are usually designed for one type of fuel, which is generally gas or oil. Larger facilities can often be designed for more than one fuel (e.g., gas, biofuels, biomass, coal, oil), and may be combined with power generation (see Chapter 2 for information on combined heat and power systems) for improved system efficiency. Often, larger central DHC plants can operate on less expensive fuels such as coal or municipal refuse.

Energy Source Economics. If an existing facility is the energy source, the available temperature and pressure of the thermal fluid is predetermined. If exhaust steam from an existing electrical generating turbine is used to provide thermal energy, the conditions of the bypass determine the maximum operating pressure and temperature of the DHC system. A trade-off analysis must be conducted to determine what percentage of the energy will be diverted for thermal generation and what percentage will be used for electrical generation. Based on the marginal value of energy, it is critical to determine the operating conditions in the economic analysis.

If a new central plant is being considered, a decision of whether to cogenerate electrical and thermal energy or to generate thermal energy only must be made. An example of cogeneration is a diesel or natural gas engine-driven generator with heat recovery equipment. The engine drives a generator to produce electricity, and heat is recovered from the exhaust, cooling, and lubrication systems. Recovered heat is used for steam or hot-water heating or even as a heat source for a steam-driven turbine or hot-water absorption chiller. Other systems may use one of several available steam turbine designs for cogeneration. These turbine systems combine the thermal and electrical output to obtain the maximum amount of available energy. Chapter 7 has further information on cogeneration.

Temperature and pressure can dramatically affect the economic feasibility of a DHC system design. If the temperature and/or pressure level chosen is too low, a potential customer base might be eliminated. On the other hand, if there is no demand for absorption chillers or high-temperature industrial processes, a low-temperature system usually provides the lowest delivered energy cost.

The availability and location of fuel sources must also be considered in optimizing the economic design of a DHC system. For example, a natural gas boiler might not be feasible where abundant sources of natural gas are not available.

Initial Capital Investment. The initial capital investment for a DHC system is usually the major economic driving force in determining whether there is acceptable payback for implementation. Normally, the initial capital investment includes concept planning and design phases as well as the construction costs of the three major system components: (1) thermal energy production plant, (2) distribution system, and (3) consumer interconnections [also known as energy transfer stations (ETs)].

Concept Planning. In concept planning phase, many areas are generally reviewed, and the technical feasibility of a DHC system is considered. This includes master planning and estimating system thermal loads and load growth potential, prospective plant site locations, piping routing, and interconnection or conversion requirements in the customer's building. The overall system concept design usually requires the services of an experienced power plant or DHC engineering firm.

Financial Feasibility. The overall capital and operating costs of the DHC system determine the energy rates charged to the prospective customers. These rates must be competitive with the customer's alternative HVAC system life-cycle costs, so a detailed analysis of this system tailored to the nature of the district energy provider is required to determine the system's financial feasibility. For example, a municipal or governmental body must consider availability of bond financing; if the entity is a private/for-profit organization, then the appropriate discount rates must be considered for the financing. Review **alternative energy choices** and **fuel flexibility**, because potential consumers are often asked to sign long-term contracts to justify a DHC system. Fuel flexibility offers the DE provider a method to keep generating costs low by being able to react to spikes in fuel costs. The financial analysis must take into account the equipment's life-cycle costs, including initial construction, operating, maintenance, and replacement, over a system life of at least 50 years.

Final Design. This phase is extremely intensive and may take several years to decades to complete the vision of the district energy provider. All technical assumptions for the design parameters must be confirmed (e.g., temperatures, pressures, flows, loads) while preparing the construction documents for the plant, distribution system, and customer interconnections. The importance of the consumer building's HVAC system water temperatures or steam pressures should not be overlooked, because they essentially drive the selection and sizing of the central plant equipment and distribution materials. Multiple building surveys may be required to determine the piping's exact entry point and routing to the appropriate mechanical or pump room.

System Construction and Costs. The accuracy of construction cost estimates for the central plant, distribution system, and ETs design depends on the quality and detail of the upfront planning and final design efforts. Although the construction cost usually accounts for most of the initial capital investment, neglect or over/underestimation in any of the other three planning phases could mean the difference between economic success and failure of the DHC system. As with any construction project, field changes usually increase the final cost and delay start-up; therefore, identifying all of the design issues and schedule in advance is extremely important. Even a small delay in start-up can adversely affect both economics and consumer confidence. It is also extremely important that the contractors have experience commensurate with the project difficulty.

Although the plant is a major component of the construction costs, the distribution system also accounts for a significant portion of this initial investment. Distribution design depends on the heat transfer medium chosen, its operating temperature and pressure, and the routing. This system component also has the most risk associated with its installation because it can encounter many underground obstructions. Often, utilities are not well documented in streets, and piping alignment conflicts are not identified until excavation begins. Pot-holing or exploratory excavations can save project time and costs when used appropriately. Failure to consider these key variables results in higher-than-planned installation costs. An analysis is recommended to determine the distribution system's insulation requirements to determine the required insulating properties and thickness. The section on Economical Thickness for Pipe Insulation discusses determining insulation values.

DHC project costs vary greatly and depend on local construction environment and site conditions such as

- Labor rates
- Construction environment (i.e., slow or busy period)
- Distance to ship equipment
- Permits and fees (e.g., franchise fees)
- Local authorities (e.g., traffic control, times of construction in city streets)
- Soil conditions (e.g., clay, bedrock)
- Quality of equipment and controls (e.g., commercial or industrial)
- Availability of materials
- Size of distribution piping system
- Type of insulation or cathodic protection for buried and aboveground piping system
- Type of distribution system installation (e.g., direct buried, tunnel)
- Distribution system depth of burial and restoration of existing conditions (e.g., city streets, green areas)
- Distribution systems below-grade conflict resolutions
- Economies of scale

Sample construction cost unit pricing is as follows. Typically, the lower unit costs are for larger systems. The designer is cautioned that costs can vary widely because of size, complexity, location, and economic conditions:

- Cooling plant (building, chillers, cooling towers, pumps, piping, controls) = \$1800 to \$3500 per ton
- Boiler plant (building, boilers, stacks, pumps, piping, controls) = \$1500 to \$2300 per boiler horsepower
- Distribution systems (includes excavation, backfill, surface restoration, piping, etc.):
 - Direct-buried chilled-water systems = \$500 to \$1250 per foot of trench
 - Direct-buried preinsulated heating piping (steam/condensate and hot water) = \$750 to \$1500 per foot of trench
 - Inaccessible tunnels = \$700 to \$1500 per foot of trench
 - Walkable tunnels = \$3500 to \$15,000 per foot of trench

Lead time needed to obtain equipment generally determines the time required to build a DHC system. In some cases (e.g., large, field-erected chillers; large motors; electrical transformers and switchgear; steam turbines), 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 interference with existing utilities. A distribution system in a new industrial park is simpler and requires less time to install than one in an established business district because of existing utility conflicts and the difficulty of working in active city streets.

Consumer Interconnection or Energy Transfer Stations. These costs are sometimes borne by the consumer either as a first cost or financed and added to their monthly bill. High interconnection costs may favor an in-building plant instead of a DHC system. For example, if an existing building is equipped for steam service, interconnection and conversion to a hot-water DHC system may be cost prohibitive, even though the cost of energy is lower. Similarly high connection costs are encountered if the point of interconnection is in a penthouse mechanical room of a high-rise building.

District Energy Economic Comparison

This section covers economic topics in more detail and helps consultants and building owners compare the costs of self-generated (in-building) central heating and cooling plants to those of a district energy contract or offer. A proper analysis requires more input than just energy and construction costs. The evaluation is extremely site specific and, though not complex, it has many input parameters, some of which are economically quantifiable and others that are qualitative and cannot be assigned a monetary value, although they do affect value.

For quantifiable parameters, the most common method of evaluation is a net present-value or life-cycle cost 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 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 energy contract is for a minimum 20-year period and the contract includes charges, including some the evaluator may not expect or know. The duration of the contract enables the district energy provider to recoup its costs for expenses incurred in connecting the building or reserving the capacity at the district energy plant for the building's load.

Ideally, a complete economic comparison between self-generated and district energy would include an 8760 h thermal energy load data from an existing building or the results from a computer energy model looking at annual energy usage. An 8760 h analysis is especially preferred for cooling applications using electrically driven equipment, because the reduction in electrical demand charges by connecting to district cooling often results in tremendous savings to the building owner. These savings are best captured by using 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 uses real-time pricing (RTP). Table 1 summarizes the key parameters in a detailed economic comparison calculation.

Table 1. Summary of Economic Analysis Factors

Factor	Comments
Capital Costs	
Construction costs of building plant versus energy transfer station equipment	Includes materials and labor for chillers, boilers, piping, pumps, heat exchangers, valving, instrumentation, controls, cost of electric service, cost of additional structures because of equipment weight on roof of building, etc.
Value of increased mechanical and electrical space that would house plant equipment	Includes value of penthouse, basement, roof, and vertical chases for flues, condenser water piping, etc.
Value of any equipment screening	Municipalities often require screening for any equipment mounted on grade or on the roof.
Cost of financing	Amount of project financed at loan interest rate for duration of loan.
Construction permits and fees	Typically a percentage of construction.
Life of major equipment overhauls and replacement costs	Could include replacement or overhauls of chillers, cooling towers, boilers, etc., over life of analysis/contract duration. If district energy contract is for 20 years and a piece of equipment must be replaced or overhauled (e.g., cooling tower replaced after 15 years, chiller/condenser water tube replacement) this cost must be accounted for.
Contract versus installed capacity	District energy capacity will most likely be less than planned in-building installed capacity, as dictated by consultant. Reasons include oversizing and diversity. Typically, estimated peak loads can be reduced to 70%.
Cost of redundant equipment for emergency or standby capacity	Similar to contract versus installed capacity, $N + 1$ redundancy requirements would be accommodated and added to first cost.
Energy and Utility Costs	
Electric rate	From use of each option from energy model or other estimate.
Natural gas rate	From use of each option from energy model or other estimate.
Water and sewer charges for steam, chilled-, and condenser-water systems	From use of each option from energy model or other estimate. Water is increasingly becoming an important resource, so makeup water and equipment blowdown/sewer discharge amount are estimated. It is not uncommon for this utility to have a different escalation rate.

Factor	Comments
Operations and Maintenance Costs	
Labor and benefits of operations staff assigned to central plant activities	Includes any staff assigned to maintain and operate the central plant, including supervisors, and overtime during 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 (e.g., gears, oil, tubes) for normal maintenance procedures.
Cost of chemical treatment for steam, chilled, condenser, and hot-water systems	Includes scale and corrosion inhibitors, biocides, oxygen scavengers, etc.; 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 energy demand rate and sizing of plant equipment (chillers, boilers, pumps, electrical service, water service, etc.).
Annual heating and cooling usage	Used to apply utilities' energy consumption rate to equipment meeting thermal loads.
Annual water and sewer usage	Quantify makeup water usage and blowdown discharge pertinent to 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.

Because 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 quantitative variables, but also qualitative variables and benefits that have intrinsic value to the proposition. For example, some qualitative variables that add value to a building connecting to district energy system could include

- Some of the mechanical and electrical space can be repurposed or rented out for other than storage (e.g., office space, roof garden)
- No plumes from cooling towers or boiler stacks
- Increased thermal energy source reliability
- More stable energy costs
- Other than a possible demand charge, a customer is only billed for energy used (metered)
- Fewer greenhouse gas emissions and smaller carbon footprint
- Not having any equipment idling and using energy in hot standby
- Freeing maintenance staff to perform other duties other than central plant operations

Of course, the building owner must determine whether any or all of these parameters pertain or are valuable in a particular building. See [Chapter 37 in the 2015 ASHRAE Handbook—HVAC Applications](#) for more detailed explanation of preparing a life-cycle cost calculation.

Example 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 but water/sewer will be escalated at 3.5%, with water/sewer at 10% per year (based on historical data from municipalities).

For alternative 1, the district cooling provider charges are \$285 capacity charge (\$/ton/year applied to peak load) and \$0.13 consumption charge (\$/ton·h) for a total annual energy cost of \$1,498,320. [Table 2](#) summarizes the major cost components (obtained from a local contractor) for each alternative. 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, 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; 90% of the cost will be financed at 5.5% interest rate. The owner has instructed the engineer to increase the chiller sizes (three at 900 tons) to accommodate for any future growth, add a little redundancy, and compensate for equipment aging. The cost estimate reflects this. An escrow account for major chiller plant overhaul (\$900/ton) was established and is expensed annually. Annual cost for preventative maintenance (\$6,000/ton for electrical chillers) was obtained from the local chiller vendor, there is one operator assigned to the plant with an annual salary of \$99,000 (includes benefits burden of 40%), and water and sewer charges at \$4/1000 gal for water and \$4/1000 gal for blowdown to sewer and chemical treatment (\$0.002\$/ton·h). The cost of insurance per year is based on 0.75% of the total construction costs.

[Table 2](#) summarizes the annual utility use for each alternative, and [Table 3](#) estimates annual maintenance and utility costs and commercial tower 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 \$0.10/kWh. An energy analysis 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 4](#) compares the two alternatives. For the values provided, alternative 1 has a 25-year life-cycle cost of \$51,525,050 and alternative 2 has a 25-year life cycle cost of \$53,034,510. Although these two values are very similar, if LCC is the only basis for the decision, alternative 1 is preferable because it has the lower life-cycle cost by \$1,509,460, and saves close to \$9,000,000 in initial costs.

[Figure 3](#) summarizes the major components of the analysis graphically. Note how much the water and sewer costs comprise of the total costs, and that the energy costs only makes up one-third of the life-cycle costs.

Table 2. Annual Utility Consumption Summary for Central Plant Alternatives		
Utility	Alternative 1	Alternative 2
Electrical, kWh	0	4,389,950
Water, 1000 gal	0	16,181
Sewer, 1000 gal	0	3,773

Cost Category	Alternative 1	Alternative 2
Annual Maintenance and Utilities		
Chiller operation and maintenance	\$0	\$ 32,460
Chiller replacement escrow	\$0	\$ 43,200
Operator salaries	\$0	\$ 99,000
Estimated water and sewer	\$0	\$ 78,816
Estimated chemicals	\$0	\$ 15,660
Insurance	\$0	\$ 67,358
Total	\$0	\$336,494
Electric Utility Cost	\$(182,316)	\$438,995
Probable Costs for Commercial Tower		
Electrical equipment	18,000	\$520,000
Chilled-water cost 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, equipment rooms)	\$5,000	\$120,000
General construction (walls, floors, equip pads, cooling tower support, etc.)	\$15,000	\$1,146,000
Controls (building management system, instrumentation, start-up, 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

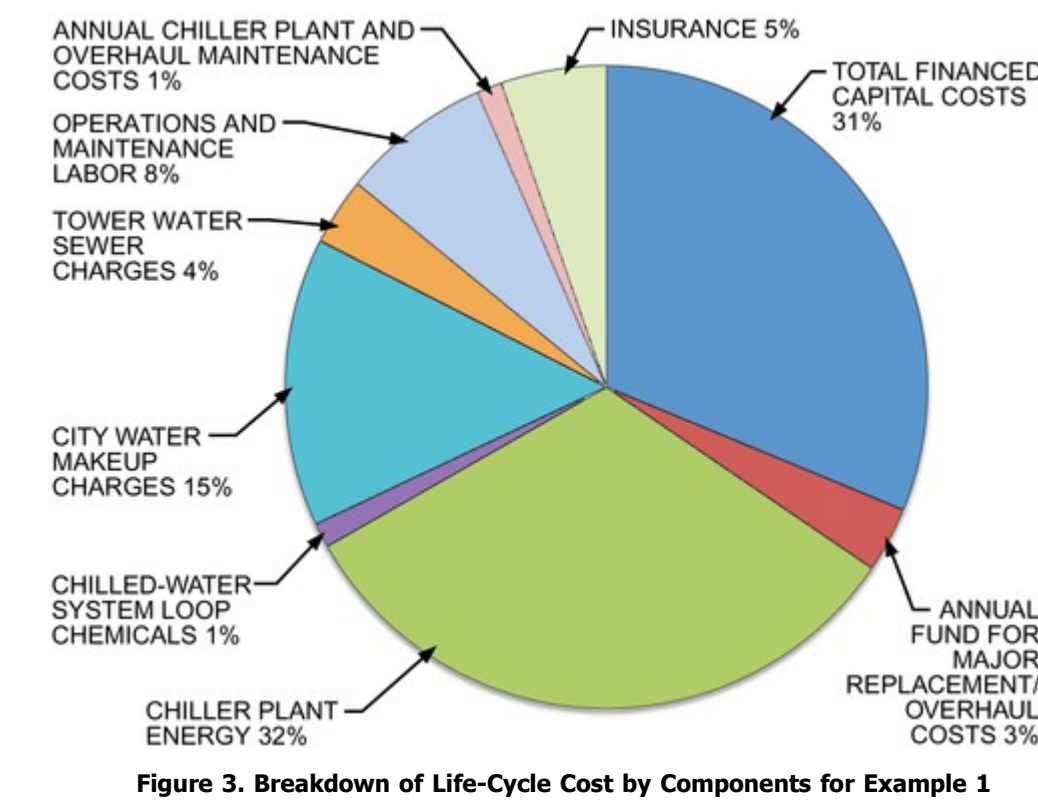


Table 4. LCC Example of District Cooling Evaluation																						
Alternative 1: Purchase Chilled Water from District Cooling Provider																						
	Year																					
	0	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	...	23	24	25		
Financing ETS first cost	\$0	\$21,582	\$21,582	\$21,582	\$21,582	\$21,582	\$21,582	\$21,582	\$21,582	\$21,582	\$21,582	\$21,582	\$21,582	\$21,582	\$21,582	\$21,582	—	\$21,582	\$21,582	\$21,582		
Chilled-water cost	\$0	\$1,498,320	\$1,550,761	\$1,605,038	\$1,661,214	\$1,719,357	\$1,779,534	\$1,841,818	\$1,906,281	\$1,973,001	\$2,042,056	\$2,113,528	\$2,187,502	\$2,264,064	\$2,343,307	\$2,425,322	—	\$3,193,686	\$3,305,465	\$3,421,157		
Electricity Savings	\$0	(\$189,316)	(\$195,943)	(\$202,801)	(\$209,899)	(\$217,245)	(\$224,849)	(\$232,718)	(\$240,863)	(\$248,019)	(\$256,050)	(\$267,396)	(\$276,396)	(\$286,070)	(\$296,083)	(\$306,446)	—	(\$403,530)	(\$417,654)	(\$432,272)		
Net annual cash flow	\$0	\$1,330,586	\$1,376,400	\$1,423,819	\$1,472,897	\$1,523,694	\$1,576,267	\$1,630,682	\$1,687,000	\$1,745,289	\$1,805,619	\$1,868,060	\$1,932,688	\$1,999,576	\$2,068,806	\$2,140,458	—	\$2,811,738	\$2,909,393	\$3,010,467		
25-year Net Present Value \$51,525,050																						
Alternative 2: Design and Install On-Site 2700 ton																						
	Year																					
	0	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	...	23	24	25		
Initial cost	\$898,100	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—		
Financing plant first cost	\$602,575	\$602,575	\$602,575	\$602,575	\$602,575	\$602,575	\$602,575	\$602,575	\$602,575	\$602,575	\$602,575	\$602,575	\$602,575	\$602,575	\$602,575	\$602,575	—	\$602,575	\$602,575	\$602,575		
Equip. replacement fund	\$0	\$44,712	\$46,277	\$47,897	\$49,573	\$51,308	\$53,104	\$54,962	\$56,886	\$58,877	\$60,938	\$63,071	\$65,278	\$67,563	\$69,928	\$72,375	—	\$95,304	\$96,640	\$102,092		
Electricity cost	\$0	\$438,995	\$454,359	\$470,262	\$486,721	\$503,756	\$521,388	\$539,637	\$558,524	\$578,072	\$598,305	\$619,245	\$640,919	\$663,351	\$686,568	\$710,598	—	\$935,722	\$958,473	\$1,002,369		
Chemical cost	\$0	\$16,775	\$17,363	\$17,970	\$18,599	\$19,250	\$19,924	\$20,621	\$21,343	\$22,090	\$22,863	\$23,663	\$24,492	\$25,349	\$26,236	\$27,154	—	\$35,757	\$37,008	\$38,304		
Water cost	\$0	\$78,318	\$86,150	\$94,765	\$104,241	\$114,665	\$126,132	\$138,745	\$152,620	\$167,882	\$184,670	\$203,137	\$223,450	\$245,795	\$270,375	\$297,412	—	\$637,530	\$701,283	\$771,411		
Sewer cost	\$0	\$18,259	\$20,085	\$22,094	\$24,303	\$26,733	\$29,407	\$32,347	\$35,582	\$39,140	\$43,054	\$47,360	\$52,096	\$57,305	\$63,036	\$69,339	—	\$148,635	\$163,498	\$179,848		
Labor cost	\$0	\$102,473	\$106,000	\$109,772	\$113,614	\$117,590	\$121,706	\$125,966	\$130,375	\$134,938	\$139,661	\$144,549	\$149,608	\$154,844	\$160,264	\$165,873	—	\$218,423	\$226,068	\$233,980		
Equipment operation and maintenance cost	\$0	\$17,354	\$17,961	\$18,590	\$19,241	\$19,914	\$20,611	\$21,332	\$22,079	\$22,852	\$23,652	\$24,479	\$25,336	\$26,223	\$27,141	\$28,091	—	\$36,990	\$38,285	\$39,625		
Insurance cost	\$0	\$72,155	\$74,680	\$77,294	\$80,000	\$82,800	\$85,698	\$88,697	\$91,801	\$95,014	\$98,340	\$101,782	\$105,344	\$109,031	\$112,847	\$116,797	—	\$153,799	\$159,182	\$164,754		
Net annual cash flow	\$1,500,675	\$1,391,616	\$1,425,510	\$1,461,219	\$1,498,867	\$1,538,591	\$1,580,545	\$1,624,882	\$1,671,785	\$1,721,440	\$1,774,058	\$1,829,861	\$1,889,098	\$1,952,036	\$2,018,970	\$2,090,214	—	\$2,864,735	\$2,995,012	\$3,134,958		
25-year Net Present Value \$53,034,510																						

2. CENTRAL PLANT

The central plant may include equipment to provide heat only, cooling only, both heat and cooling, or any of these three options in conjunction with electric power generation. In addition to the central plant, small "satellite" plants are sometimes used when a customer's building is in an area where distribution piping is not yet installed.

2.1 HEATING AND COOLING PRODUCTION

In plants serving hospitals, industrial customers, or cogeneration sites (i.e., those also generating electricity), steam is the usual choice for production in the plant and, often, for distribution to customers. For systems serving largely commercial buildings, hot water is usually an attractive medium. From the standpoint of distribution, hot water can accommodate a greater geographical area than steam because of the ease with which booster pump stations can be installed. The common attributes and relative merits of hot water and steam as heat-conveying media are described as follows. More information is also available in [Chapter 11](#) of this volume, and in [Chapter 3](#) of the *District Heating Guide* (ASHRAE 2013b).

Heat Capacity. Steam relies primarily on the latent heat capacity of water rather than on sensible heat. The net heat content for saturated steam at 100 psig (338°F) is approximately 1040 Btu/lb. Hot water cooled from 350 to 250°F has a net heat effect of 103 Btu/lb, or only about 10% as much as that of steam. Thus, a hot-water system must circulate about 10 times more mass than a steam system of similar heat capacity.

Pipe Sizes. Despite the fact that less steam is required for a given heat load, and flow velocities are greater, steam usually requires a larger pipe size for the supply line because of its lower density (Kamre and Phetteplace 1978). This is compensated for by a much smaller condensate return pipe. Therefore, piping costs for steam and condensate are often comparable with those for hot-water supply and return, especially given the fact that trap station vaults are not required.

Return System. Condensate return systems require more maintenance than hot-water return systems function as a closed loop with very low makeup water requirements. For condensate return systems, corrosion of piping and other components is a problem, particularly where feedwater is high in bicarbonates. Nonmetallic piping is essentially impervious to corrosion, but material temperature limitations mean it should only be used with caution and in carefully selected installations for condensate return. Suitable applications include low-temperature pumped returns, where it is possible to isolate the nonmetallic piping from live steam. Otherwise, avoid using nonmetallic piping in steam systems because its material temperature limits can easily be exceeded by steam trap failures and exposure to high-temperature condensate or live steam.

Similar concerns are associated with condensate drainage systems (steam traps, condensate pumps, and receiver tanks) for steam supply lines. Carefully consider condensate collection and return when designing a steam system. Although similar problems with water treatment occur in hot-water systems, they present less of a concern because makeup rates are much lower.

Pressure and Temperature Requirements. Flowing steam and hot water both incur pressure losses. Hot-water systems may use intermediate booster pumps to increase the pressure at points between the plant and the customer. Because of the higher density of water, pressure variations caused by elevation differences in a hot-water system are much greater than for steam systems. This can adversely affect the economics of a hot-water system by requiring use of a higher pressure class of piping and/or booster pumps or even heat exchangers used as pressure interceptors.

Regardless of the medium used, the temperature and pressure used for heating should be no higher than needed to satisfy consumer requirements; this cannot be overemphasized. Higher temperatures and pressures require additional engineering and planning to avoid higher heat losses and higher piping stress forces. Safety and comfort levels for operators and maintenance personnel also benefit from lower temperature and pressure. Higher temperatures may require higher pressure ratings for piping and fittings and may preclude the use of less expensive materials such as polyethylene foam insulation and nonmetallic conduits.

Hot-water systems are divided into three temperature classes:

- High-temperature hot-water (HTHW) systems supply temperatures over 350°F.
- Medium-temperature hot-water (MTHW) systems supply temperatures in the range of 250 to 350°F.
- Low-temperature hot-water (LTHW) systems supply temperatures of 250°F or lower.

The temperature drop at the consumer end should be as high as possible, preferably 40°F or greater. A large temperature drop allows the fluid flow rate through the system, pumping power, return temperatures, return line heat loss, and condensing temperatures in cogeneration power plants to be reduced. A large customer temperature drop can often be achieved by cascading loads operating at different temperatures. Typical Scandinavian LTHW systems have temperature drops of 70°F or greater. [Chapter 15](#) provides more information on medium- and high-temperature water-heating systems.

In many instances, existing equipment and processes require the use of steam, which precludes use of hot water or requires smaller high-pressure boilers at the buildings. See the section on Consumer Interconnections for further information.

Steam and Hot Water Generation

With few exceptions, boilers are constructed to meet requirements in the sections on constructing heating and power boilers of the ASME *Boiler and Pressure Vessel Code* (2013). Low-pressure boilers are constructed for maximum working pressures of 15 psig steam and up to 160 psig hot water. Low-pressure hot-water boilers are limited to 250°F operating temperature. High-pressure boilers are designed to operate above 15 psig steam or above 160 psig and/or 250°F for water boilers.

In the United States, boilers are rated by boiler horsepower, with 1 hp being equal to 33,475 Btu/h or the evaporation of 34.5 lb of water per hour at standard atmospheric pressure 14.7 psia and 212°F. Every steam or water boiler is rated for a maximum working pressure according to its applicable boiler code. When installed, it also must be equipped at a minimum with operation and safety controls and pressure/temperature-relief devices mandated by such codes.

Fire-tube and water-tube boilers are available for gas/oil firing (see [Chapter 32](#) for details). If coal is used, either package-type coal-fired boilers in small sizes (less than 20,000 to 25,000 lb/h) or field-erected boilers in larger sizes are available. Coal-firing underfed stokers are available up to a 30,000 to 35,000 lb/h capacity; traveling grate and spreader stokers are available up to 160,000 lb/h capacity in single-boiler installations. Fluidized-bed boilers can be installed for capacities over 300,000 lb/h. Larger coal-fired boilers are typically multiple installations of the three types of stokers or larger, pulverized fired or fluidized-bed boilers. Generally, the complexity of fluidized-bed or pulverized firing does not lend itself to small central heating plant operation. Coal-fired boilers typically can be partially cofired with other solid fuels such as wood biomass, refuse, or tire-derived fuels (TDF), or retrofitted to combust a higher percentage of nonfossil fuels.

Chilled-Water Generation

As with smaller in-building chilled-water plants, district plants have options for which chiller type, refrigerant, and prime drivers to use. Chiller types in district cooling systems vary by location, depending on parameters such as water availability, power availability, maximum power demand that can be offered by the utility, steam availability, gas or fuel oil availability, distribution temperature required, plant location with respect to development, and applied environment impact regulations (including pollution and noise control).

The chillers may be classified according to

- Refrigerant and compressor technology (centrifugal, screw, absorption, etc.)
- Heat rejection source (air cooled or water cooled)
- Driving energy source (electric motor, steam turbine, hot water or steam absorption, internal combustion engine, etc.)
- Supply temperature required
- Size: capacity in tons

Water-cooled heat rejection can be air cooled or water cooled. **Air-cooled chillers** are typically packaged, with controls, compressors, evaporator, and air-cooled condenser all on the same skid. Their capacity may be as high as 450 tons, and they have been implemented in several central plants by installing multiple units in chiller farms, either on grade or on building rooftops.

Chiller-cooled chillers reject heat carried by the water circuit through either 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. Because the wet-bulb temperature is lower than the coincident dry-bulb temperature, a cooling tower with a water-cooled chiller is more efficient than an air-cooled one at removing chiller heat because of the lower condensing temperature: more energy can be rejected by evaporating one pound of water than by raising the temperature of an equivalent amount of air. Because water-cooled chiller plants are typically considerably more efficient than either air-cooled or water-cooled radiator plants, they require less heat rejection area. [Tables 5](#) and [6](#) summarize the available chiller technologies, efficiencies with first cost and maintenance costs.

The chilled-water supply temperature for a conventional system ranges from 40 to 44°F. Chilled-water return temperatures are set as high as practical to increase the temperature difference Δt between supply and return, reducing the quantity of chilled water circulated in the system. Traditional systems use a 12°F Δt , resulting in a flow rate of 2 gpm/ton of refrigeration. Because of the cost of the distribution system piping and pumping energy, modern chilled-water systems operate at lower supply water temperatures and/or higher return water temperatures to allow a larger Δt to be achieved, thereby reducing chilled-water flow per ton of capacity. For systems involving stratified chilled-water storage, a practical lower limit is 39°F because of water density considerations; however, chemical additives can suppress this temperature below 28°F. For ice storage systems, supply water temperatures as low as 34°F have been used.

Multiple air-conditioning loads connected to a central chilled-water system provide economic advantages and energy conservation opportunities over a decentralized system approach. In addition, central plants afford the opportunity to use refrigerants such as ammonia that may be impractical for use in individual buildings. The size of air-conditioning loads served, diversity among the loads, and their distance from the chilling plant are principal factors in determining the feasibility of large central plants. The distribution system pipe capacity is directly proportional to the operating temperature difference between the supply and return lines, and it benefits additionally from increased diversity in the connected loads.

For extremely large district systems, several plants may be required to meet the loads economically and provide redundancy. In some areas, plants over 20,000 tons are common for systems exceeding a total connected load of 50,000 tons. Another reason for multiple plants is that single plants over 30,000 tons can require large piping headers (over 48 in. in diameter) in the plant, as well as large distribution headers in streets already congested with other utilities, making piping layout problematic. Multiple plants use smaller distribution piping and thus increase system constructability, redundancy, and reliability.

Table 5. Chiller Technology				
Compression Chillers			Absorption Chillers	
Parameter	Reciprocating	Screw	Centrifugal	Two-stage
Primary energy	Electric motor	Electric motor	Hot water, 149 to 176°F	Steam or fire, temperature > 338°F
Fluids	R-134a, HCFC, NH ₃	R-134a, HCFC, NH ₃	H ₂ O with LiBr, NH ₃ with H ₂ O	H ₂ O with LiBr, NH ₃ with H ₂ O
COP	4 to 6	4 to 6	>7.0	1.2

Chiller Type	Typical Efficiency	Size Range, ton	Equipment Cost, \$/ton	Estimated Annual Maintenance Costs ²
Electric centrifugal (standard single compressor)	0.61 to 0.7 kW/ton (COP 4.7 to 5.4)	500 to >1500	200 to 275	\$3700
Electric centrifugal (standard dual compressor)	0.61 to 0.7 kW/ton (COP 4.7 to 5.4)	1500 to >4000	250 to 350	\$4400 to \$5000
			400 to 450	\$3700 to \$4200
			650 to 800	\$4800 to \$5500
Electric centrifugal (single compressor industrial, field erected)	0.61 to 0.7 kW/ton (COP 4.7 to 5.4)	2500 to >5500	450 to 650	\$4400 + engine maintenance
Engine-driven centrifugal	(COP 1.5 to 1.9)	100 to >3000	—	\$3500+ depending on size
Steam-driven centrifugal	—	100 to >4000	—	—
Direct-fired (double effect) absorption chiller	(COP 0.85 to 1.20)	<100 to >3250	400 to 2000	\$4800 to \$5500
Hot-water absorption chiller (single effect)	(COP 0.55 to 0.70)	<60 to >3250	450 to 1000	\$4800 to \$5500
Steam absorption chiller (single effect)	(COP 0.60 to 0.75)	<60 to >3250	450 to 800	\$4800 to \$5500

² Maintenance costs courtesy of Johnson Controls, Inc./York International. 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. Approximate pricing for those additional items follows:

- Cleaning evaporator or condenser tubes as required, use \$1200.
- Eddy current testing for under 500 ton chiller, use \$1700; for larger chillers, \$2500.
- Complete oil replacement: contact vendor's service department.

For all absorbers, a bromide test is conducted twice per year. Costs do not include chemicals.

An economic evaluation of piping and pumping costs versus chiller power requirements can establish the most suitable supply and return water temperatures. When sizing piping and calculating pumping cost, the heat load on the chiller generated by the frictional heating of the flowing fluid should be considered because most of the pumping power adds to the system heat load. For high chiller efficiency, it is often more efficient to use isolated auxiliary equipment for special process requirements and to allow the central plant supply water temperature to float up at times of lower load. As with heating plants, optimum chilled-water control may require a combination of temperature modulation and flow modulation. However, the designer must investigate the effects of higher chilled-water supply temperatures on chilled-water secondary system distribution flows and air-side system performance (humidity control) before applying this to individual central water plants.

Thermal Storage

Both hot- and chilled-water thermal storage can be implemented for district systems. In North America, the current economic situation primarily results in chilled-water storage applications. Depending on plant design and loading, thermal storage can reduce chiller equipment requirements and lower operating costs. By shifting part of the chilling load, chillers can be sized closer to the average load than the peak load. Shifting some or all of the refrigeration load to off peak reduces the on-peak electrical demand load while using the same (or slightly larger) chiller machine capacity. Because many utilities offer lower rates (and perhaps some rebates) during off-peak periods, operating costs for electrically driven chillers can be substantially reduced.

Both ice and chilled-water storage have been applied to district-sized chiller plants. In general, the largest systems (>20,000 ton-hour capacity) use chilled-water (CHW) storage and small- to moderate-sized systems use ice storage. Storage capacities in the 10,000 to 30,000 ton-hour range are now common and systems have been installed up to 125,000 ton-hour for district cooling systems. When CHW storage is feasible, be careful not to reduce the chilled-water temperature below 39.2°F, to allow proper temperature stratification in the thermal energy storage (TES) tank. A TES tank charging temperature lower than 39°F without low-temperature additives will result in mixing in the tank, and possibly disturbance of the system supply temperature and the storage concept.

For these reasons, most chilled-water plants are designed for a 40°F supply temperature and a 56°F or greater return temperature. Some plants may be designed for lower temperatures by using multiple cascaded heat exchangers in series, such as high-rise towers; however, note that each 1°F reduction in supply temperature increases chiller-specific energy consumption (i.e., kW/ton) by approximately 2%. Some plants are designed for a higher return temperature (60°F or higher) to increase the ΔT , but this requires a great deal of additional coordination with the design of the building's HVAC system to ensure the system operates per design intent.

In Europe, several cooling systems use naturally occurring underground aquifers (caverns) for storage of chilled water. Selection of the storage configuration (chilled-water steel tank above grade, chilled-water concrete tank below grade, ice direct, ice indirect) is often influenced by space limitations. Depending on the system design temperatures, chilled-water storage requires four to six times the volume of ice storage for the same capacity. For chilled-water storage, the footprint of steel tanks (depending on height) can be less than concrete tanks for the same volume (Andrzejewski 1995). Furthermore, the cost of above-grade tanks is usually less than below-grade tanks.

[Chapter 5](#), has additional information on thermal storage; for thermal storage specifically in district cooling and heating, also see [Phetteplace et al. \(2013a, 2013b\)](#), respectively.

Auxiliaries

Numerous pieces of auxiliary support equipment related to the boiler and chiller operations are not unique to the production plant of a DHC system and are found in similar installations. Some components of a DHC system deserve special consideration because of their critical nature and potential effect on operations.

Although instrumentation can be either electronic or pneumatic, electronic instrumentation systems offer the flexibility of combining control systems with data acquisition systems. This combination brings improved efficiency, better energy management, and reduced operating staff for the central heating and/or cooling plant. For systems involving multiple fuels and/or thermal storage, computer-based controls are indispensable for accurate decisions about boiler and chiller operation.

Boiler feedwater treatment has a direct bearing on equipment life. Condensate receivers, filters, polishers, and chemical feed equipment must be accessible for proper management, maintenance, and operation. Depending on the temperature, pressure, and quality of the heating medium, water treatment may require softeners, alkalizers, and/or demineralizers for systems operating at high temperatures and pressures.

Equipment and layout of a central heating and cooling plant should reflect what is required for proper plant operation and maintenance. The plant should have an adequate service area for equipment and a sufficient number of electrical power outlets and floor drains. Equipment should be placed on housekeeping pads to protect the bases from spills or leaks. [Figure 4](#) presents a typical layout for a large hot-water/chilled-water plant. Notice that the layout provides space for future expansion as well as storage of spare parts. The control room is typically close to the operating equipment for ease of visual inspection. Other functions that should be considered include adequate space for maintenance, chemical treatment laboratory and chemical storage, and operator conveniences such as locker rooms and lunch rooms. Designers must follow the requirements of ASHRAE *Standard 15* for ventilation as well as laying out the equipment room, and should coordinate with the architect regarding tight-sealing doors, minimizing penetrations, etc.

Expansion Tanks and Water Makeup. The expansion tank is usually located in the central plant building. To control pressure, either air or nitrogen is introduced to the air space in the expansion tank. To function properly, the expansion tank must be the single point of the system where no pressure change occurs. Multiple, air-filled tanks may cause erratic and possibly harmful movement of air through the piping and should be avoided. Although diaphragm expansion tanks eliminate air movement, the possibility of hydraulic surge should be considered. On large chilled-water systems, a makeup water pump generally is used to makeup water loss. The pump is typically controlled from level switches on the expansion tank or from a desired pump suction pressure.

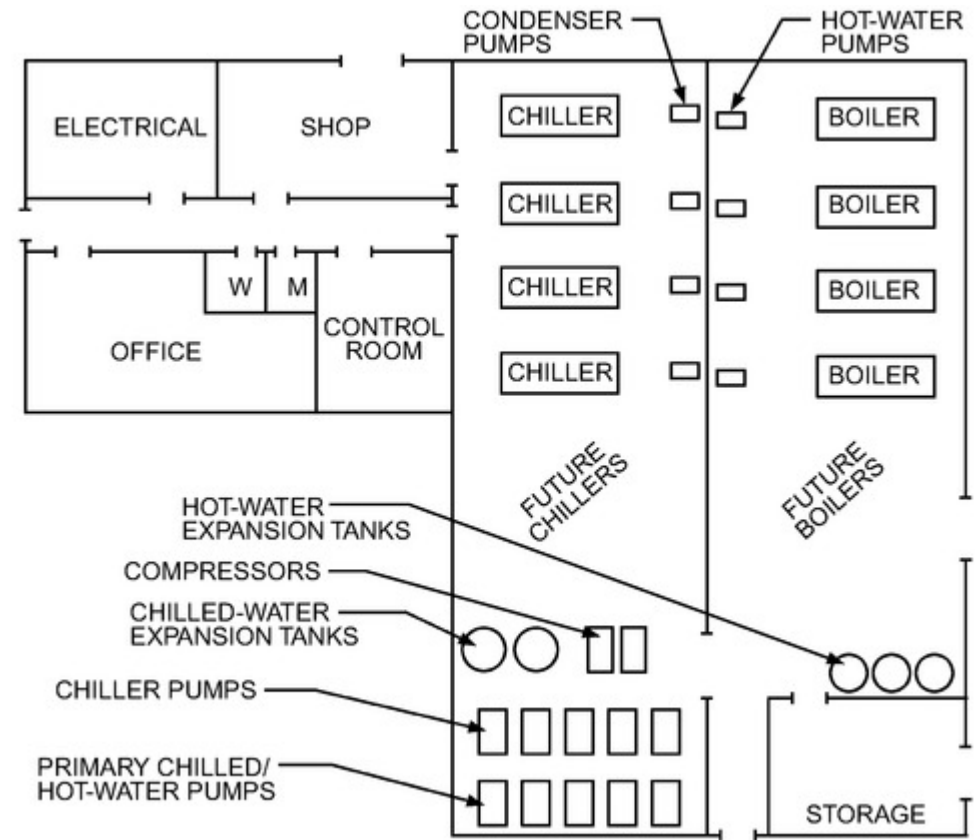


Figure 4. Layout for Hot-Water/Chilled-Water Plant

A conventional water meter on the makeup line can show water loss in a closed system. This meter also provides necessary data for water treatment. The fill valve should be controlled to open or close and not modulate to a very low flow, so that the water meter can detect all makeup. For systems with thermal energy storage (TES) tanks, consider providing a rapid-fill connection from either the domestic water system or an adjacent fire hydrant connection. Use caution if connecting to the fire line, because this water would be untreated and raw, and would require chemical treatment; hence, the chemical treatment system should also be sized for a rapid-fill event.

Air Venting. Air in water systems can cause major issues for district systems, because it acts as insulator for heat transfer and increases corrosion in system components (e.g., chillers, pipes, heat exchangers). Trapped air causes air pockets in the CHW system that can accumulate at fittings and prohibit flow, causing the system to become **air bound**. Sources of air in a closed system include

- Makeup water containing normal amounts of dissolved air
- Air trapped in the system after initial filling
- Diffusion
- Air ingress caused by negative pressure

Several devices are used to eliminate air from CHW systems. Air exists within the system in three forms: (1) stagnant bubbles, (2) entrained in the flow, and (3) dissolved. Large, stagnant bubbles can be removed by manual or automatic venting if enough pressure is available. If bubble size exceeds the air vent passage, a capillary problem occurs and prevents bubble removal. In this case, use either a surface tension breaker or a large-bore vent (0.5 in.). Centrifugal separators are ideally suited for continuous gas venting in district plants.

For proper separator function, velocity through the separator should not exceed 6 fpm and should allow for water turbulence. Some separators contain a zone to collect dirt and blow it down. Dissolved air (invisible) is difficult to remove through separators or air vents, but can be removed using a vacuum-pressure degasser, in which a fraction of the circulated water is put into a vacuum, allowing separation and removal of dissolved gas.

Emission Control. In all cases, the cleanest fuel should be used to minimize emissions and environmental impacts. Emission treatment equipment, including electrostatic precipitators, baghouses, and scrubbers, is required to meet emission standards for coal-fired or solid-waste-fired operations. Proper control is critical to equipment operation, and it should be designed and located for easy access by maintenance personnel.

A baghouse gas filter provides good service if gas flow and temperature are properly maintained. Because baghouses are designed for continuous online use, they are less suited for cyclic operation. Cyclical heating and cooling of the fabric significantly reduces the useful life of the bags through acidic condensation. Using an economizer to preheat boiler feedwater and help control flue gas temperature may enhance baghouse operation. Contaminants generated by plant operation and maintenance, such as washdown of floors and equipment, may need to be contained.

Local codes and regulations may also require low-NO_x burners on gas- or oil-fired boilers or engine generators. [Chapter 28 of the 2013 ASHRAE Handbook—Fundamentals](#) and [Chapter 30](#) of this volume have information on air pollution and its control.

2.2 CHILLED-WATER DISTRIBUTION DESIGN CONSIDERATIONS

Water distribution systems are designed for either constant flow (variable return temperature) or variable flow (constant return temperature). The design decision between constant- or variable-volume flow affects the (1) selection and arrangement of the chiller(s), (2) design of the distribution system, and (3) design of the customer connection to the distribution system. Unless very unusual circumstances exist, most systems large enough to be considered in the district category are likely to benefit from variable-flow design. See [Chapter 11](#) for additional information on pumping.

Constant Flow

In the past, constant chilled-water flow was applied only to smaller systems where simplicity of design and operation were important and where distribution pumping costs were low, before variable- and adjustable-speed drives were available and affordable. (However, for new systems, designers should refer to ASHRAE *Standard* 90.1, which requires variable-flow pumping.) Chillers were also arranged in series to handle higher system design temperature differentials. Flow rate through a full-load distribution system depended on the type of constant-flow system used. A common technique connected the building and its terminal units across the distribution system. The central plant pump circulated chilled water through air-side terminal units controlled by three-way valves (constant-volume direct primary pumping). Balancing problems could occur in this design when many separate flow circuits were interconnected ([Figure 5](#)).

Constant-flow distribution was also applied to in-building (secondary or tertiary) circuits with separate pumps. This arrangement isolates the flow balance problem between buildings. In this case, flow through the distribution system could be significantly lower than the sum of the flows needed by the terminal if the in-building system supply temperature were higher than the distribution system supply temperature ([Figure 3](#)). The water temperature rise in the distribution system was determined by the connected in-building systems and their controls.

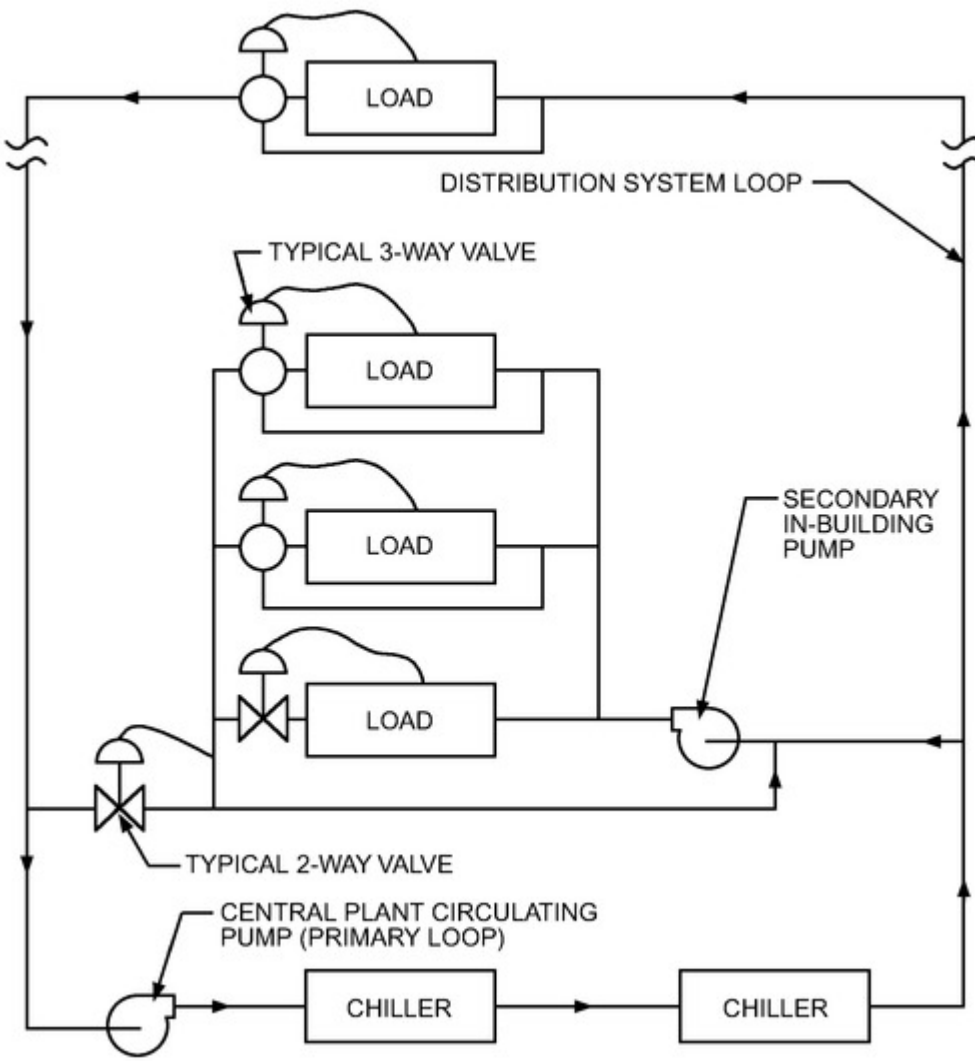


Figure 5. Constant-Flow Primary Distribution with Secondary Pumping

In constant-flow design, chillers arranged in parallel have decreased entering water temperatures at part load; thus, several machines might have needed to run simultaneously, each at a reduced load, to produce the required chilled-water flow. In this case, chillers in series were better because constant flow could be maintained though the chilled-water plant at all times, with only the chillers required for producing chilled water energized. Legacy constant-flow systems should be analyzed thoroughly when considering multiple chillers in a parallel arrangement, because the auxiliary electric loads of condenser water pumps, tower fans, and central plant circulating pumps are a significant part of the total energy input. Contemporary control systems mitigate the reason for constant flow through the chillers, and variable flow should be used for larger systems.

Variable Flow

Variable-flow design can significantly reduce energy use and expand the capacity of the distribution system piping by using diversity. To maintain a high temperature differential at part load, the distribution system flow rate must track the load imposed on the central plant. Multiple parallel pumps or, more commonly, variable-speed pumps can reduce flow and pressure, and lower pumping energy at part load. Terminal device controls should be selected to ensure that variable flow objectives are met. Correctly sized terminal unit flow-throttling (two-way) valves, and especially pressure-independent control valves (PICV), provide the continuous high return temperature needed to correlate the system load change to a system flow change.

Systems in each building are usually two-pipe, with individual in-building pumping. In some cases, the pressure of the distribution system may cause flow through the in-building system without in-building pumping. Distribution system pumps can provide total building system pumping if (1) the distribution system pressure drops are minimal, and (2) the distribution system is relatively short-coupled (3000 ft or less). To implement this pumping method, the total flow must be pumped at a pressure sufficient to meet the requirements of the building with the largest pressure differential requirement. Consequently, all buildings on the system should have their pressure differentials monitored and transmitted to the central plant, where pump speeds are adjusted to provide adequate pressure to the building with the lowest margin of pressure differential (i.e., hydraulically most remote or the critical node).

If the designer has control over the design of each in-building system, this pumping method can be achieved in a reasonable manner. In retrofits where existing buildings under different ownership are connected to a new central plant, coordination is difficult and individual building pumps are more practical.

When buildings have separate circulating pumps, use hydraulic isolating piping and pumping design to ensure that two-way control valves are subjected only to the differential pressure established by the in-building pump. [Figure 5](#) shows a connection using in-building pumping with hydraulic isolation from the primary loop.

When in-building pumps are used, all series interconnections between the distribution system pump and the in-building pumps should be removed. Without adequate instrumentation and controls, a series connection can cause the distribution system return to operate at a higher pressure than the distribution system supply and disrupt flow to adjacent buildings. Series operation often occurs during improper use of three-way mixing valves in the distribution-to-building connection.

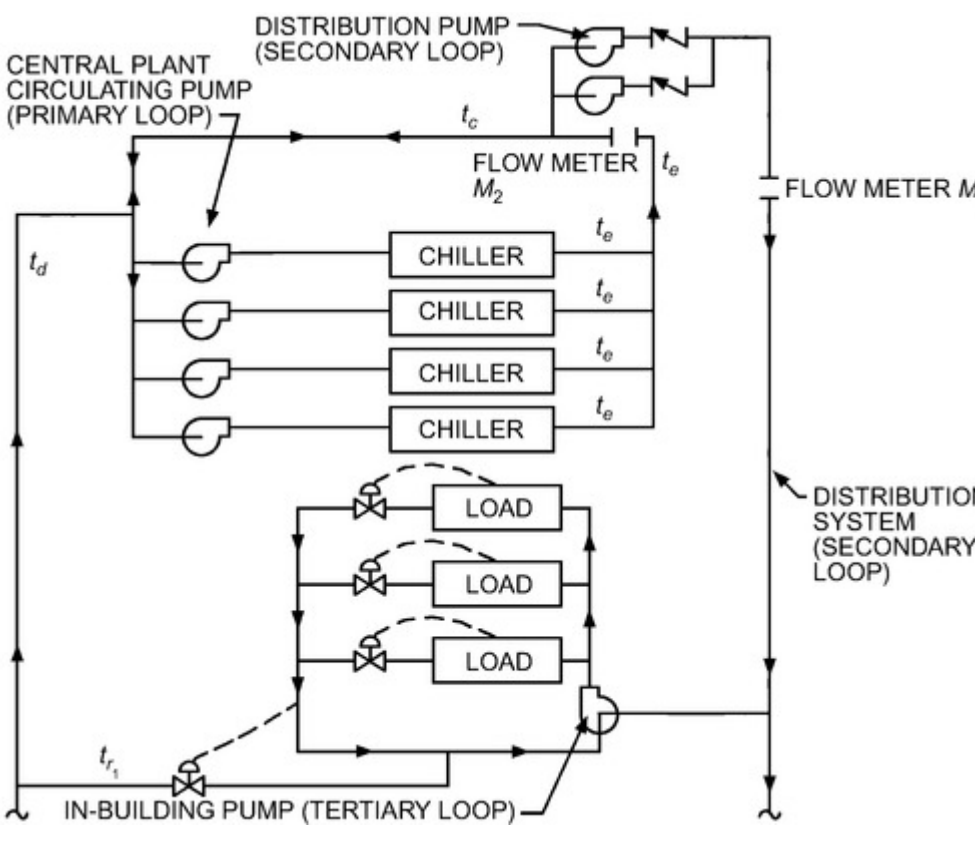


Figure 6. Variable-Flow Primary/Secondary Systems

In very large systems, **distributed pumping** may be used. Under this approach, the distribution pumps in the central plant are eliminated and relocated to the buildings, so all the electrical energy is borne by the pumps in the user buildings. Where the distribution network piping constitutes a significant pressure loss (systems covering a large area), this design allows the distributed pumps in the buildings to be sized for just the pressure loss imposed at that particular location. Ottmer and Rishel (1993) found that this approach reduces total chilled-water pump power by 20 to 25% in very large systems. It is best applied in new construction where the central plant and distributed building systems can be planned fully and coordinated initially. Note that this system is not the best approach for a system that expects dynamic growth, such as a college and university campus, because adding a large load anywhere in the system adversely affects the pressure drop of each distribution pump. Furthermore, it is a common design approach to oversize distribution piping in distributed pumping systems to keep the distribution pressure drop low, to reduce the pressure drop and size of the building distribution pumps.

Usually, a positive pressure must be maintained at the highest point of the system at all times. This height determines the static pressure at which the expansion tank operates. Excessively tall buildings that do not isolate the in-building systems from the distribution system can impose unacceptable static pressure on the distribution system. To prevent excessive operating pressure in distribution systems, heat exchangers have been used to isolate the in-building system from the distribution system to act as pressure interceptors. To ensure reasonable temperature differentials between supply and return temperatures, flow must be controlled on the distribution system side of the heat exchanger.

In high-rise buildings, all piping, valves, coils, and other equipment may be required to withstand high pressure. Where system static pressure exceeds safe or economical operating pressure, either the heat exchanger method or pressure-sustaining valves in the return line with check valves in the supply line may be used to minimize pressure. However, the pressure-sustaining/check valve arrangement may overpressurize the entire distribution system if a malfunction of either valve occurs.

Chilled-Water System Design Guidelines

Guidelines for plant design and operation include the following:

- Variable-speed pumping saves energy and should be considered for distribution system pumping.
- Design chilled-water systems to optimize the temperature differential of greater than 14 to 16°F. A 16 to 24°F maximum temperature differential with 14°F minimum temperature differential can be achieved with this design.
- Avoid using constant-flow chilled-water systems.
- Investigate using chillers arranged in series for larger temperature differentials or increased chiller efficiency.
- Larger central chilled-water plants with high distribution system pressure drops can benefit from variable flow in the distribution system. This can be achieved with variable primary, primary/secondary, or primary/secondary/tertiary pumping systems.
- Size the distribution system for a low overall total pressure loss. Short-coupled distribution systems (3000 ft or less) can be used for a total pressure loss of 20 to 40 ft of water. With this maximum differential between any points in the system, size the distribution pumps to provide the necessary pressure to circulate chilled water through the in-building systems, eliminating the need for in-building pumping systems. This decreases the complexity of operating central chilled-water systems. Newer controls on chillers enable all-variable-flow systems. Check with the manufacturer about minimum flows on chiller evaporators to achieve stable operation over all load ranges.
- All two-way valves must have proper close-off ratings and a design pressure drop of at least 20% of the maximum design pressure drop for controllability. Commercial quality automatic temperature control valves generally have low shutoff ratings; but industrial valves can achieve higher ratings. Three-way control valves should be avoided except to accommodate minimum pump flow and shutdown. See [Chapter 43](#) for more information on control valves.
- Although chillers can easily produce colder water, the lower practical limit for chilled-water supply temperatures is 39°F. Temperatures below that should be carefully analyzed for energy usage, although systems with thermal energy storage may operate advantageously at lower temperatures.

3. DISTRIBUTION SYSTEM

3.1 HYDRAULIC CONSIDERATIONS

Objectives of Hydraulic Design

Usually, the distribution of a thermal utility such as hot water encompasses many of the aspects of domestic hot-water distribution, many dissimilarities also exist; thus the design should not be approached in the same manner. Thermal utilities must supply sufficient energy at the appropriate temperature and pressure to meet consumer needs. Within the constraints imposed by the consumer's end use and equipment, the required thermal energy can be delivered with various combinations of temperature and pressure. Computer-aided design methods are available for thermal piping networks (Bloomquist et al. 1999; COWI/consult 1985; Rasmussen and Lund 1987; Reisman 1985). Using these methods allows rapid evaluation of many alternative designs.

General steam system design can be found in [Chapter 11](#), as well as in IDHA (1983) and Phetteplace et al. (2013b). For water systems, see [Chapter 13](#), IDHA (2008b), IDHA (1983), and Phetteplace et al. (2013a, 2013b).

Water Hammer

The term water hammer is used to describe several phenomena that occur in fluid flow. Although these phenomena differ in nature, they all result in stresses in the piping that are higher than normally encountered. Water hammer can have a disastrous effect on a thermal utility by bursting pipes and fittings and threatening life and property.

In steam systems (IDHA 1983), water hammer is caused primarily by condensate collecting in the bottom of the steam piping. Steam flowing at velocities 10 times greater than normal water flow picks up a slug of condensate and accelerates it to a high velocity. The slug of condensate subsequently collides with the pipe wall at a point where flow changes direction. To prevent this type of water hammer, condensate must be prevented from collecting in steam pipes by using proper steam pipe pitch and adequate condensate collection and return facilities.

Water hammer also occurs in steam systems because of rapid condensation of steam during system warm-up. Rapid condensation decreases the specific volume and pressure of steam, which precipitates pressure shock waves. This form of water hammer is prevented by controlled warm-up of the piping. Valves should be opened slowly and in stages during warm-up. Large steam valves should be provided with smaller bypass valves to slow the warm-up. Water hammer in hot- and chilled-water distribution systems is caused by sudden changes in flow velocity, which causes pressure shock waves. The two primary causes are pump failure (i.e., power failure) and sudden valve closures. A simplified method to determine maximum resultant pressure may be found in [Chapter 22 of the 2013 ASHRAE Handbook—Fundamentals](#). More elaborate methods of analysis can be found in Fox (1977), Stephenson (1981), and Streeter and Wylie (1979). Preventive measures include operational procedures and special piping fixtures such as surge columns.

Pressure Losses

Frictional pressure losses occur at the interface between the inner wall of a pipe and a flowing fluid due to shear stresses. In steam systems, these pressure losses are compensated for with increased steam pressure at the point of steam generation. In water systems, pumps are used to increase pressure at either the plant or intermediate points in the distribution system. Calculation of pressure loss is discussed in [Chapters 3 and 22 of the 2013 ASHRAE Handbook—Fundamentals](#). Hydraulic calculations reveal that a great deal of the system pressure drop is caused by pipe fittings and offsets; therefore, it is prudent to lay out distribution systems with as few offsets as possible. Some designers prefer two 45° elbows in lieu of a single 90° elbow, assuming that the pressure drop is lower. As pressure drop calculations show, this is not so.

Pipe Sizing

Ideally, the appropriate pipe size should be determined from an economic study of the life-cycle cost for construction and operation. In practice, however, this study is seldom performed because of the effort involved. Instead, criteria that have evolved from practice are frequently used for design. These criteria normally take the form of constraints on the maximum flow velocity or pressure drop. [Chapter 22 of the 2013 ASHRAE Handbook—Fundamentals](#) provides velocity and pressure drop constraints. Noise generated by excessive flow velocities is usually not a concern for thermal utility distribution systems outside of buildings. For steam systems, maximum flow velocities of 200 to 250 fpm are recommended (IDHA 1983). For water systems, Europeans use the criterion that pressure losses should be limited to 0.44 psi per 100 ft of pipe (Behm 1988). Other studies indicate that higher levels of pressure loss may be acceptable (Stewart and Dona 1987) and warranted from an economic standpoint (Behm 1986; Koskela 1980; Phetteplace 1989). For chilled-water systems (excluding condenser water piping) in buildings, the requirements of ASHRAE *Standard* 90.1-2013 must be followed for water velocity and piping pressure drops.

When establishing design flows for thermal distribution systems, consider the diversity of consumer demands (i.e., the various consumers' maximum demands do not occur at the same time). Thus, the heat supply and main distribution piping may be sized for a maximum load that is somewhat less than the sum of the individual consumers' maximum demands. For steam systems, Geiringer (1963) suggests diversity factors of 0.80 for space heating and 0.65 for domestic hot-water heating and process loads. Geiringer also suggests that these factors may be reduced by approximately 10% for high-temperature water systems. Werner (1984) conducted a study of the heat load on six operating low-temperature hot-water systems in Sweden and found diversity factors ranging from 0.57 to 0.79, with the average being 0.685. Use caution when applying these diversity factors, because the size and number of loads in a system affect the values.

Network Calculations

Calculating flow rates and pressures in a piping network with branches, loops, pumps, and heat exchangers can be difficult without the aid of a computer. Methods have been developed primarily for domestic water distribution systems (Jeppson 1977; Stephenson 1981). These may apply to thermal distribution systems with appropriate modifications. Computer-aided design methods usually incorporate methods for hydraulic analysis as well as for calculating heat losses and delivered water temperature at each consumer. Calculations are usually carried out in an iterative fashion, starting with constant supply and return temperatures throughout the network. After initial estimates of the design flow rates and heat losses are determined, refined estimates of the actual supply temperature at each consumer are computed. Flow rates at each consumer are then adjusted to ensure that the load is met with the reduced supply temperature, and the calculations are repeated.

Most calculations are performed for a static or steady-state condition and have several input assumptions (e.g., temperatures, pressures, flow rates) for a specific moment in time. Some computer modeling software packages are dynamic and take real-time inputs of flow and pressure from field instrumentation to efficiently control and optimize the speed of the distribution pumps.

Condensate Drainage and Return in Steam Systems

Condensate forms in operating steam lines as a result of heat loss. When a steam system's operating temperature is increased, condensate also forms as steam warms the piping. At system start-up, these loads usually exceed any operating heat loss loads; thus, special provisions should be made.

To drain the condensate, steam piping should slope toward a collection point called a **drip station**. Drip stations are located in access areas or buildings where they are accessible for maintenance. Steam piping should slope toward the drip station at least 1 in. in 40 ft. If possible, the steam pipe should slope in the same direction as steam flow; otherwise, increase the pipe size to at least one size greater than would normally be used. This reduces the flow velocity of the steam and provides better condensate drainage against the steam flow. Drip stations should be spaced no further than 500 ft apart in the absence of other requirements.

Drip stations consist of a short piece of pipe (called a **drip leg**) positioned vertically on the bottom of the steam pipe, as well as a steam trap and appurtenant piping. The drip leg should be the same diameter as the steam pipe. The length of the drip leg should provide a volume equal to 50% of the condensate load from system start-up for steam pipes of 4 in. diameter and larger and 25% of the start-up condensate load for smaller steam pipes (IDHA 1983). Steam traps should be sized to meet the normal load from operational heat losses only. Start-up loads should be accommodated by manual operation of the bypass valve.

Steam traps are used to separate the condensate and noncondensable gases from the steam. For drip stations on steam distribution piping, use inverted bucket or bimetallic thermostatic traps. Some steam traps have integral strainers; others require separate strainers. Ensure that drip leg capacity is adequate when thermostatic traps are used because they always accumulate some condensate.

If it is to be returned, condensate leaving the steam trap flows into the condensate return system. If steam pressure is sufficiently high, it may be used to force the condensate through the condensate return system. With low-pressure steam or on systems where a large pressure exists between drip stations and the ultimate destination of the condensate, condensate receivers and pumps must be provided. High-pressure condensate from drip traps may be sparged into low-pressure condensate mains with a sparge tube (essentially, the high-pressure condensate flashes in the sparge tube).

Schedule 80 steel piping is recommended for condensate lines because of the extra allowance for corrosion it provides. Steam traps have the potential of failing in an open position, thus nonmetallic piping must be protected from live steam where its temperature/pressure would exceed the limitations of the piping. Nonmetallic piping should not be located so close to steam pipes that heat losses from the steam pipes could overheat it. Additional information on condensate removal may be found in [Chapter 11](#). Information on sizing condensate return piping may be found in [Chapter 22 of the 2013 ASHRAE Handbook—Fundamentals](#).

3.2 THERMAL CONSIDERATIONS

Thermal Design Conditions

- Three thermal design conditions must be met to ensure satisfactory system performance:
- 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, make appropriate allowances in the cost analysis.
 - Maximum heat transfer rate determines the load on the central plant due to the distribution system. It also determines the temperature drop (or increase, in the case of chilled-water distribution), which determines the delivered temperature to the consumer. For this calculation, each component's thermal conductivity 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 carrier medium and the soil or air. The burial depth is normally at its lowest value for this calculation. During operation, none of the thermal capabilities of the materials (or any other materials in the area influenced thermally by the system) must exceed design conditions. To satisfy this objective, each component and the surrounding environment must be examined to determine whether thermal damage is possible. A numerical heat transfer analysis may be necessary in some cases.
 - The conditions of these analyses must be chosen to represent the worst-case scenario from the perspective of the component being examined. For example, in assessing the suitability of a coating material for a metallic conduit, the thermal insulation is assumed to be saturated, the soil moisture is at its lowest probable level, and the burial depth is maximum. These conditions, combined with the highest anticipated pipe and soil temperatures, give the highest conduit surface temperature to which the coating could be exposed.
- Heat transfer in buried systems is influenced by the thermal conductivity of the soil and by the depth of burial, particularly when the insulation has low thermal resistance. Soil thermal conductivity changes significantly with moisture content; for example, Bortoff (1951) indicated that soil thermal conductivity ranges from 0.083 Btu/h · ft · °F during dry soil conditions to 1.25 Btu/h · ft · °F during wet soil conditions. For details on calculating thermal effects on district energy distribution piping, see Phetteplace et al. (2013a, 2013b).

Table 7. Comparison of Commonly Used Insulations in Underground Piping Systems				
	Calcium Silicate Type I/II ASTM C533	Urethane Foam	Cellular Glass ASTM C552	Mineral Fiber/Preformed Glass Fiber Type 1 ASTM C547
Thermal conductivity ^a (Values in parentheses are maximum permissible by ASTM standard listed), Btu/h · ft · °F				
Mean temp. = 100°F	0.028	0.013	0.033 (0.030)	0.022 (0.021)
200°F	0.031 (0.038/0.045)	0.014	0.039 (0.037)	0.025 (0.026)
300°F	0.034 (0.042/0.048)		0.046 (0.045)	0.028 (0.033)
400°F	0.038 (0.046/0.051)		0.053 (0.054)	(0.043)
Density (max.), lb/ft ³	15/22		6.7 to 9.2	8 to 11
Maximum temperature, °F	1200	250	800	850
Compressive strength (min), ^b psi	100 at 5% deformation		60	N/A
Dimensional stability, linear shrinkage at maximum use temperature	2%		N/A	2%
Flame spread	0		5	25
Smoke index	0		0	50
Water absorption	As-shipped moisture content, 20% max. (by weight)		0.5	Water vapor sorption, 5% max. (by weight)

^a Thermal conductivity values in this table are from previous editions of this chapter and have been retained as they were used in examples. Thermal conductivity of insulation may vary with temperature, temperature gradient, moisture content, density, thickness, and shape. ASTM maximum values given are comparative for establishing quality control compliance, and are suggested for preliminary calculations where actual values are not available. They may not represent installed performance of insulation under actual conditions that may differ substantially from test conditions. The manufacturer should be able to supply appropriate design values.

^b Compressive strength for cellular glass shown is for flat material, capped as per ASTM Standard C240.

Thermal Properties of Pipe Insulation and Soil

Uncertainty in heat transfer calculations for thermal distribution systems results from uncertainty in the thermal properties of materials involved. Generally, the designer must rely on manufacturers' 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.

Insulation. Insulation provides the primary thermal resistance against heat loss or gain in thermal distribution systems. Thermal properties and other characteristics of insulations normally used in thermal distribution systems are listed in [Table 7](#). Material properties such as thermal conductivity, density, compressive strength, moisture absorption, dimensional stability, and combustibility are typically reported in ASTM standards 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 Standards C177, C518, or C1114. For piping containing hot media, thermal conductivity for insulation material fabricated or molded for use on piping is measured using ASTM Standard C335.

Chyu et al. (1997a, 1997b, 1998a, 1998b) studied the effect of moisture on the thermal conductivity of insulating materials commonly used in underground district energy systems (ASHRAE research project RP-721). The results are summarized in [Table 8](#). The insulated pipe was immersed in water maintained at 46 to 100°F to simulate possible conduit water temperatures during a failure. The fluid temperature in the insulated pipe ranged from 35 to 450°F. All insulation materials were tested unfaced and/or unf jacketed to simulate installation in a conduit.

Table 8. Effect of Moisture on Underground Piping System Insulations				
Characteristics	Polyurethane ^a	Cellular Glass	Mineral Wool ^b	Fibrous Glass
Heating Test				
Pipe temp. 35 to 260°F Water bath 46 to 100°F	Pipe temp. 35 to 420°F Water bath 46 to 100°F		Pipe temp. 35 to 450°F Water bath 46 to 100°F	Pipe temp. 35 to 450°F Water bath 46 to 100°F
Length of submersion time to reach steady-state <i>k</i> -value	70 days	5	10 days	2 h
Effective <i>k</i> -value increase from dry conditions after steady state achieved in submersion	14 to 19 times at steady state. Estimated water content of insulation 70% (by volume).	Avg. 10 times, process unsteady. ⁵ Insulation showed evidence of moisture zone on inner diameter.	Up to 50 times at steady state. Insulation completely saturated	52 to 185 times. Insulation completely saturated at steady state.
Primary heat transfer mechanism	Conduction	5	Conduction and convection	Conduction and convection
Length of time for specimen to return to dry steady-state <i>k</i> -value after submersion	Pipe at 260°F, after 16 days moisture content 10% (by volume) remaining	Pipe at 420°F, 8 h	Pipe at 450°F, 9 days	Pipe at 380°F, 6 days
Cooling Test				
Pipe temp. 37°F Water bath at 52°F	Pipe temp. 36°F Water bath 46°F to 58°F		Pipe temp. 35°F to 45°F Water bath 55°F	Insulation 35°F to 450°F Water bath 46°F to 100°F
Length of submersion time to reach steady-state conditions for <i>k</i> -value	16 days	Data recorded at 4 days constant at 12 days	6 days	1/2 h
Effective <i>k</i> -value increase from dry conditions after steady state achieved	2 to 4 times. Water absorption minimal, ceased after 7 days.	None. No water penetration.	14 times. Insulation completely saturated at steady state.	20 times. Insulation completely saturated at steady state.
Primary heat transfer mechanism	Conduction	Conduction	Conduction and convection	Conduction and convection
Length of time for specimen to return to dry steady-state <i>k</i> -value after submersion	Pipe at 37°F, data curve extrapolated to 10+ days	Pipe at 33°F, no change	Pipe at 35°F, data curve extrapolated to 25 days	Pipe at 35°F, 15 days

Source: Chyu et al. (1997a, 1997b; 1998a, 1998b).

^a Polyurethane material tested had a density of 2.9 lb/ft³.

^b Mineral wool tested was a preformed molded basalt designed for pipe systems operating up to 1200°F. It was specially formulated to withstand the Federal Agency Committee (FAC) 96 h boiling water test. See Phetteplace (2013b) for the FAC test protocol.

⁵ Cracks formed on heating for all samples of cellular glass insulation tested. Flooded heat loss mechanism involved dynamic two-phase flow of water through cracks; period of dynamic process was about 20 min. Cracks had negligible effect on thermal conductivity of dry cellular glass insulation before and after submersion. No cracks formed during cooling test.

Painting chilled-water piping before insulating is recommended in areas of high humidity. Insulations used today for chilled water include polyurethane, polysiocyanurate, phenolics, cellular plastics, and fiberglass. No insulation system is totally water- and vaportight; thus, water from the atmosphere can enter the insulation system in small amounts, which can cause corrosion of the pipe. Chloride ions present in most atmospheric water exacerbate corrosion. The best way to minimize corrosion is to make the insulation system highly water resistant by using a closed-cell insulation material coupled with a high-performance vapor retarder, and painting the pipe exterior with a strong rust-preventative coating (two-part epoxy) before insulating. This is good engineering practice and most insulation manufacturers suggest this, but it may not be in their literature. In addition, a good vapor retarder is required on the exterior of the insulation.

When assessing the potential for pipe corrosion under insulation, the pipe operating temperature must be considered. Pipes operating permanently or predominantly at temperatures below 50°F or above 250°F are not prone to corrosion. Pipes operating between these temperatures, particularly if they are subject to temperature cycling in use or from frequent shutdowns, are most susceptible to corrosion.

Soil. If a soil analysis is available or can be done, the thermal conductivity of the soil can be estimated from published data (e.g., Farouki (1981); Lunardini (1981)). The thermal conductivity factors in [Table 9](#) may be used as an estimate when detailed information on the soil is not known. Because dry soil is rare in most areas, only assume a low moisture content for system material design, or where it can be validated for calculation of heat losses in the normal operational condition. Values of 0.8 to 1 Btu/h · ft · °F are commonly used where soil moisture content is unknown. Because moisture migrates toward a chilled pipe, use a thermal conductivity value of 1.25 Btu/h · ft · °F for chilled-water systems in the absence of any site-specific soil data. 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.

3.3 METHODS OF HEAT TRANSFER ANALYSIS

Because heat transfer in piping is not related to the load factor, it can be a large part of the total load. The most important factors affecting heat transfer are the difference between ambient and fluid temperatures and the thermal insulation. For direct-buried piping, the ambient temperature is the earth or soil temperature. For example, the extremes might be a 6 in., insulated, 400°F water line in 40°F earth with 100 to 200 Btu/h · ft loss; and a 6 in., uninsulated, 55°F chilled-water return in 60°F earth with 10 Btu/h · ft gain. The former requires analysis to determine the required insulation and its effect on the total heating system; the latter suggests analysis and insulation needs might be minimal. Other factors that affect heat transfer are (1) depth of burial, related to the earth temperature and soil thermal resistance; (2) soil thermal conductivity, related to soil moisture content and density; and (3) distance between adjacent pipes.

Table 9. Soil Thermal Conductivities			
Soil Moisture Content (by mass)	Thermal Conductivity, Btu/h · ft · °F		
	Sand	Silt	Clay
Low, <4%	0.17	0.08	0.08
Medium, 4 to 20%	1.08	0.75	0.58
High, >20%	1.25	1.25	1.25

Computing transient heat gains or losses in underground piping systems requires using numerical methods that approximate any physical problem and include factors such as the effect of temperature on thermal properties. For most designs, numerical analyses may not be warranted; one exception is where the potential exists to thermally damage components of the distribution system or something adjacent to it. Also, complex geometries may require numerical analysis. Albert and Phetteplace (1983), Minkowicz et al. (1988), and Rao (1982) have further information on the theory of methods of numerical analysis, and many commercial software packages are available for conducting numerical analyses.

Steady-state calculations are appropriate for determining the annual heat loss/gain from a buried system if average annual earth temperatures are used. Steady-state calculations may also be appropriate for worst-case analyses of thermal effects on materials. Steady-state calculations for a one-pipe system may be done without a computer, but it becomes increasingly difficult for a two-, three-, or four-pipe system.

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. *Note:* For consistency and simplicity, all thermal conductivities are given in Btu/h · ft · °F with all dimensions in feet (not the more traditional Btu · in/h · ft² · °F, which is widely used in building envelope design).

Calculation of Undisturbed Soil Temperatures

Before any heat loss/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 Conditions. For example, if the purpose of the calculation is to determine whether a material will exceed its temperature limit, the maximum expected undisturbed ground temperature is used. 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 and suggestions on appropriate circumstances to use them are as follows:

- Use the average annual air or groundwater temperature to approximate the average annual soil temperature. This estimate is appropriate when the objective of the calculation is to yield average heat loss over the yearly weather cycle. Mean annual air temperatures may be obtained from various sources of climatic data (e.g., [Chapter 14 of the 2013 ASHRAE Handbook—Fundamentals](#)).
- 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 an appropriate conservative assumption when checking the temperatures to determine if the temperature limits of any of materials proposed for use will be exceeded. Maximum and minimum expected air temperatures may be calculated from [Equations \(1\) and \(2\)](#) and setting the depth to zero.
- For systems that are buried more deeply, maximum/minimum undisturbed soil temperatures may be estimated as a function of depth, soil thermal properties, and prevailing climate. This estimate is appropriate when checking temperatures in a system to determine whether the temperature limits of any of the materials proposed for use will be exceeded. The following equations may be used to estimate the minimum and maximum expected undisturbed soil temperatures:

$$\text{Maximum temperature} = t_{s,z} = t_{ms} + A_s e^{-z\sqrt{\pi/\alpha\tau}} \tag{1}$$

$$\text{Minimum temperature} = t_{s,z} = t_{ms} - A_s e^{-z\sqrt{\pi/\alpha\tau}} \tag{2}$$

where

- $t_{s,z}$ = temperature, °F
- z = depth, ft
- τ = annual period, 365 days
- α = thermal diffusivity of the ground, ft²/day
- t_{ms} = mean annual surface temperature, °F
- A_s = surface temperature amplitude, °F

Values for the climatic constants t_{ms} and A_s may be found at t62.ashraets.org/pdf/ASHRAE_Climatic_Data.pdf for all worldwide weather stations included in the CD accompanying the 2009 *ASHRAE Handbook—Fundamentals*.

Thermal diffusivity for soil may be calculated as follows:

$$\alpha = \frac{24k_s}{\rho_s[c_s + c_w(w/100)]} = \frac{24k_s}{\rho_s[c_s + (w/100)]} \tag{3}$$

where

- ρ_s = soil density, lb/ft³
- c_s = dry soil specific heat, Btu/lb · °F
- c_w = specific heat of water = 1.0 Btu/lb · °F
- w = moisture content of soil, % (dry basis)
- k_s = soil thermal conductivity, Btu/h · ft · °F

Because the specific heat of dry soil is nearly constant for all types of soil, c_s may be taken as 0.175 Btu/lb · °F.

- For buried systems, 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 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 time of year are desired. Substituting 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). (*Note:* The argument for the sine function is in radians.)

$$t_{s,z} = t_{ms} - A_s e^{-z\sqrt{\pi/\alpha\tau}} \sin \left[\frac{2\pi(\theta - \theta_{lag})}{\tau} - z\sqrt{\frac{\pi}{\alpha\tau}} \right] \tag{4}$$

where

- θ = Julian date, days
- θ_{lag} = phase lag of soil surface temperature, days

Use [Equation \(3\)](#) to calculate soil thermal diffusivity. Values for the climatic constants t_{ms} , A_s , and θ_{lag} may be found at t62.ashraets.org/pdf/ASHRAE_Climatic_Data.pdf for all worldwide weather stations included in the CD accompanying the 2009 *ASHRAE Handbook—Fundamentals*. Phetteplace et al. (2013b) contains equations that may be used to calculate the climatic constants for any weather data set, real or contrived.

[Equation \(4\)](#) does not account for latent heat effects from freezing, thawing, or evaporation. However, for soil adjacent to a buried heat distribution system, the equation provides a good estimate, because heat losses from the system tend to prevent the adjacent ground from freezing. For buried chilled-water systems, freezing may be a consideration; therefore, systems that are not used or drained during the winter months should be buried below the seasonal frost depth. For simplicity, 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, use the following method to compensate for the convective thermal resistance to heat transfer at the ground surface.

Convective Heat Transfer at Ground Surface

Heat transfer between the ground surface and the ambient air occurs by convection. In addition, heat transfer with the soil occurs due to 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. However, in some circumstances, approximations that included impacts beyond an average convective heat transfer coefficient were found. For example, McCabe et al. (1995) observed significant temperature variations caused by the type of surface over district heating and cooling systems. Phetteplace et al. (2013a) contains methods to approximate the impacts of surface type.

Normally, only convection is considered, and as a first approximation, 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:

$$\delta = k_s/h \tag{5}$$

where

- δ = effective thickness of fictitious soil layer, ft
- k_s = thermal conductivity of soil, Btu/h · ft · °F
- h = convective heat transfer coefficient at ground surface, Btu/h · ft² · °F

The effective thickness calculated with [Equation \(5\)](#) is simply added to the actual burial depth of the pipes in calculating the soil thermal resistance using [Equations \(6\), \(7\), \(20\), \(21\), and \(27\)](#).

Uninsulated Buried Pipe

For this case ([Figure 2](#)), an estimate for soil thermal resistance may be used. This estimate is sufficiently accurate (within 1%) for the ratios of burial depth to pipe radius indicated next to [Equations \(6\) and \(7\)](#). Both the actual resistance and the approximate resistance are presented, along with the depth/radius criteria for each.

$$R_s = \frac{\ln \{ (d/r_o) + [(d/r_o)^2 - 1]^{1/2} \}}{2\pi k_s} \quad \text{for } d/r_o > 2 \tag{6}$$

$$R_s = \frac{\ln(2d/r_o)}{2\pi k_s} \quad \text{for } d/r_o > 4 \tag{7}$$

where

- R_s = thermal resistance of soil, h · ft · °F/Btu
- k_s = thermal conductivity of soil, Btu/h · ft · °F
- d = burial depth to centerline of pipe, ft
- r_o = outer radius of pipe or conduit, ft

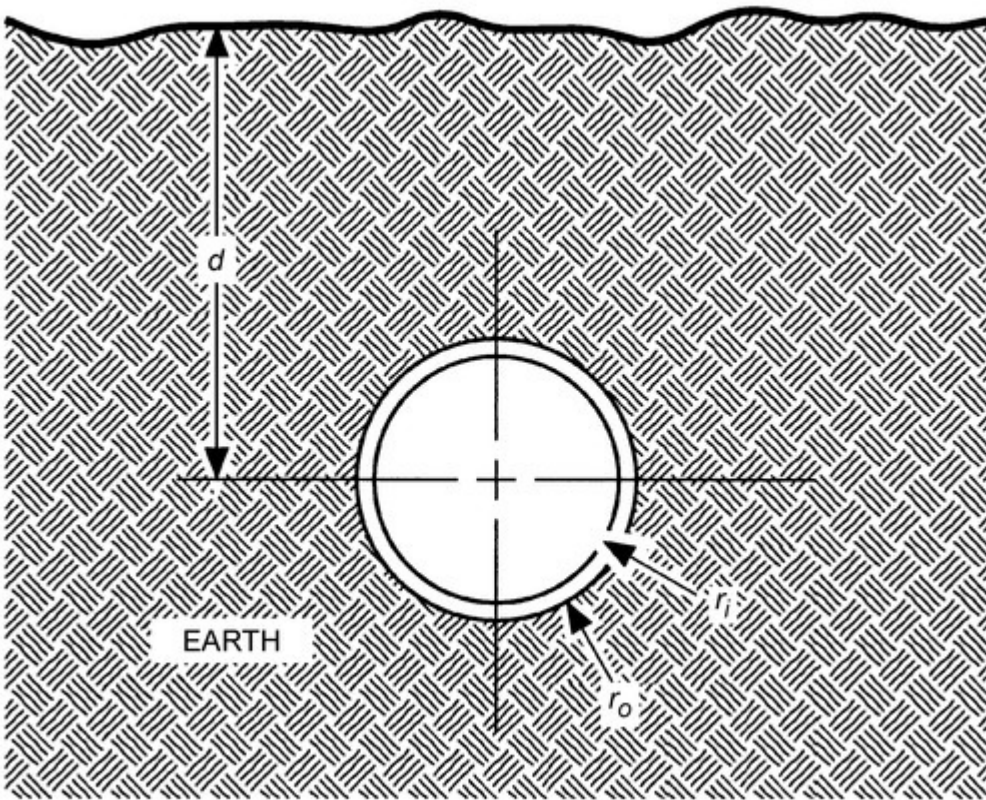


Figure 7. Uninsulated Buried Pipe

The pipe's thermal resistance is included if it is significant compared to the soil resistance. The thermal resistance of a pipe or any concentric circular region is given by

$$R_p = \frac{\ln(r_o/r_i)}{2\pi k_p}$$

(8)

where

- R_p = thermal resistance of pipe wall, h · ft · °F/Btu
- k_p = thermal conductivity of pipe, Btu/h · ft · °F
- r_i = inner radius of pipe, ft

Example 2. Consider an uninsulated, 3 in. Schedule 40 PVC chilled-water supply line carrying 45°F water. Assume the pipe is buried 3 ft deep in soil with a thermal conductivity of 1 Btu/h · ft · °F, and no other pipes or thermal anomalies are in close proximity. Assume the average annual soil temperature is 60°F.

- r_i = 1.54 in. = 0.128 ft
- r_o = 1.75 in. = 0.146 ft
- d = 3 ft
- k_o = 1 Btu/h · ft · °F
- k_o = 0.10 Btu/h · ft · °F

Solution: Calculate the thermal resistance of the pipe using [Equation \(8\)](#):

$$R_p = 0.21 \text{ h} \cdot \text{ft} \cdot ^\circ\text{F}/\text{Btu}$$

Calculate the thermal resistance of the soil using [Equation \(7\)](#). [Note: $d/r_o = 21$ is greater than 4; thus [Equation \(7\)](#) may be used in lieu of [Equation \(6\)](#).]

$$R_s = 0.59 \text{ h} \cdot \text{ft} \cdot ^\circ\text{F}/\text{Btu}$$

Calculate the rate of heat transfer by dividing the overall temperature difference by the total thermal resistance:

$$q = \frac{t_f - t_s}{R_t} = \frac{(45 - 60)}{0.80 \text{ h} \cdot \text{ft} \cdot ^\circ\text{F}/\text{Btu}} = -19 \text{ Btu/h} \cdot \text{ft}$$

where

- R_t = total thermal resistance, (i.e., $R_s + R_p$ in this case of pure series heat flow), h · ft · °F/Btu
- t = fluid temperature, °F
- t_s = average annual soil temperature, °F
- q = heat loss or gain per unit length of system, Btu/h · ft

The negative result indicates a heat gain rather than a loss. Note that the thermal resistance of the fluid/pipe interface has been neglected, which is a reasonable assumption because 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 to 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.

Insulated Buried Pipe
[Equation \(8\)](#) 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, use actual rather than nominal thickness to obtain the most accurate results.

Example 3. Consider the effect of adding 1 in. of urethane foam insulation and a 1/8 in. thick PVC jacket to the chilled-water line in Example 2. Calculate the thermal resistance of the insulation layer from [Equation \(8\)](#) as follows:

$$R_i = \frac{\ln(0.229/0.146)}{2\pi \times 0.0125} = 5.75 \text{ h} \cdot \text{ft} \cdot ^\circ\text{F}/\text{Btu}$$

For the PVC jacket material, use [Equation \(8\)](#) again:

$$R_j = \frac{\ln(0.240/0.229)}{2\pi \times 0.10} = 0.07 \text{ h} \cdot \text{ft} \cdot ^\circ\text{F}/\text{Btu}$$

The thermal resistance of the soil as calculated by [Equation \(7\)](#) decreases slightly to $R_s = 0.51 \text{ h} \cdot \text{ft} \cdot ^\circ\text{F}/\text{Btu}$ because of the increase in the outer radius of the piping system. The total thermal resistance is now

$$R_t = R_p + R_i + R_j + R_s = 0.21 + 5.75 + 0.07 + 0.51 \\ = 6.54 \text{ h} \cdot \text{ft} \cdot ^\circ\text{F}/\text{Btu}$$

Heat gain by the chilled-water pipe is reduced to about 2 Btu/h · ft. 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 if insulation is present.

Buried Pipe in Conduit with Air Space

Systems with air spaces ([Figure 8](#)) may be treated by adding an appropriate resistance for the air space. For simplicity, assume a heat transfer coefficient of 3 Btu/h · ft² · °F (based on the outer surface area of the insulation), which applies in most cases. The resistance caused by this heat transfer coefficient is then

$$R_a = 1/(3 \times 2\pi \times r_{oi}) = 0.053/r_{oi}$$

(9)

where

- r_{oi} = outer radius of insulation, ft
- R_a = resistance of air space, h · ft · °F/Btu

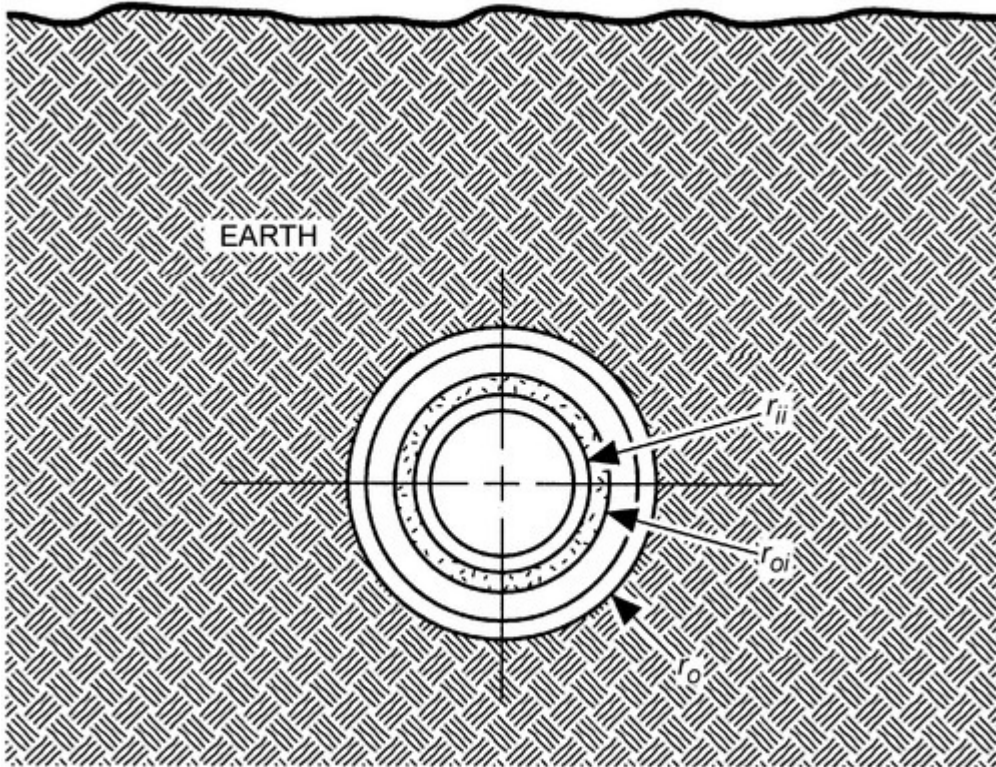


Figure 8. Insulated Buried Pipe with Air Space

A more precise value for the resistance of an air space can be developed with empirical relations available for convection in enclosures such as those given by Grober et al. (1961). Consider the effect of radiation in the annulus when high temperatures are expected in the air space. For the treatment of radiation, refer to Siegel and Howell (1981).

Example 4. Consider a 6 in. nominal diameter (6.625 in. outer diameter) high-temperature water line operating at 375°F. Assume the pipe is insulated with 2.5 in. of mineral wool with a thermal conductivity $k = 0.026$ Btu/h · ft · °F at 200°F and $k = 0.030$ Btu/h · ft · °F at 300°F.

The pipe is encased in a steel conduit with a concentric air gap of 1 in. The steel conduit is 0.125 in. thick and has a corrosion resistant coating approximately 0.125 in. thick. The pipe is buried 4 ft deep to pipe centerline in soil with an average annual temperature of 60°F. The soil thermal conductivity is assumed to be 1 Btu/h · ft · °F. Thermal resistances of the pipe, conduit, and conduit coating will be neglected.

Solution: Calculate the thermal resistance of the pipe insulation. To do so, assume a mean insulation temperature of 250°F to establish its thermal conductivity, which is equivalent to assuming the insulation outer surface temperature is 125°F. Interpolating the data listed previously, insulation thermal conductivity $k_i = 0.028$ Btu/h · ft · °F. Then calculate insulation thermal resistance using [Equation \(8\)](#):

$$R_i = \frac{\ln(0.484/0.276)}{2\pi \times 0.028} = 3.19 \text{ h} \cdot \text{ft} \cdot ^\circ\text{F}/\text{Btu}$$

Calculate the thermal resistance of the air space using [Equation \(9\)](#):

$$R_a = 0.053/0.484 = 0.11 \text{ h} \cdot \text{ft} \cdot ^\circ\text{F}/\text{Btu}$$

Calculate the thermal resistance of the soil from [Equation \(7\)](#):

$$R_s = \frac{\ln(8.0/0.589)}{2\pi \times 1} = 0.42 \text{ h} \cdot \text{ft} \cdot ^\circ\text{F}/\text{Btu}$$

The total thermal resistance is

$$R_t = R_i + R_a + R_s = 3.19 + 0.11 + 0.42 = 3.72 \text{ h} \cdot \text{ft} \cdot ^\circ\text{F}/\text{Btu}$$

The first estimate of the heat loss is then

$$q = (375 - 60)/3.72 = 84.7 \text{ Btu/h} \cdot \text{ft}$$

Now repeat the calculation with an improved estimate of the mean insulation temperature obtained, and calculate the outer surface temperature as

$$t_{io} = t_{po} - (qR_i) = 375 - (84.7 \times 3.19) = 105^\circ\text{F}$$

where t_{io} = outer surface temperature of pipe, °F.

The new estimate of the mean insulation temperature is 240°F, which is close to the original estimate. Thus, the insulation thermal conductivity changes only slightly, and the resulting thermal resistance is $R_i = 3.24 \text{ h} \cdot \text{ft} \cdot ^\circ\text{F}/\text{Btu}$. Other thermal resistances in the system remain unchanged, and the heat loss becomes $q = 83.8$ Btu/h · ft. The insulation surface temperature is now approximately $t_{io} = 104^\circ\text{F}$, and no further calculations are needed.

Buried Pipe with Composite Insulation

Many systems use more than one insulating material. The motivation for doing so is usually to use an insulation that has desirable thermal properties or lower cost, but it is unable to withstand the service temperature of the carrier pipe (e.g., polyurethane foam). Another insulation with acceptable service temperature limits (e.g., calcium silicate or mineral wool) is normally placed adjacent to the carrier pipe in sufficient thickness to reduce the temperature at its outer surface to below the limit of the insulation that is desirable to use (e.g., urethane foam). Calculation of the heat loss and/or temperature in a composite system is straightforward using the equations presented previously. The following two examples show calculation of both heat losses and temperatures of interest in such systems.

Example 5. A high-temperature water line operating at 400°F is installed in southern Texas. It consists of a 6 in. nominal diameter (6.625 in. outer diameter) carrier pipe insulated with 1.5 in. of calcium silicate. In addition, 1 in. polyurethane foam insulation is placed around the calcium silicate insulation. The polyurethane insulation is encased in a 0.5 in. thick fiberglass-reinforced plastic (FRP) jacket. The piping system is buried 10 ft deep to the pipe centerline. Neglect the thermal resistances of the pipe and FRP jacket.

Calcium silicate thermal conductivity

0.038 Btu/h · ft · °F at 200°F

0.042 Btu/h · ft · °F at 300°F

0.046 Btu/h · ft · °F at 400°F

Polyurethane foam thermal conductivity

0.013 Btu/h · ft · °F at 100°F

0.014 Btu/h · ft · °F at 200°F

0.015 Btu/h · ft · °F at 275°F

Assumed soil properties

Thermal conductivity = 0.2 Btu/h · ft · °F

Density (dry soil) = 105 lb/ft³

Moisture content = 5% (dry basis)

What is the maximum operating temperature of the polyurethane foam insulation?

Solution: Because the maximum operating temperature of the materials is sought, the maximum expected, rather than the average annual, soil temperature must be found. Also, the lowest anticipated soil thermal conductivity and deepest burial depth are assumed. These conditions produce the maximum internal temperatures of the components.

To solve the problem, first calculate the maximum soil temperature expected at the installation depth using [Equation \(1\)](#). Assume the following values for climatic constants for this region as $t_{aw} = 71.8^\circ\text{F}$ and $A_s = 15.0^\circ\text{F}$ (see [tc62.ashraetcs.org/pdf/ASHRAE_Climatic_Data.pdf](#) for climatic data constants). Estimate soil thermal diffusivity using [Equation \(3\)](#).

$$\alpha = \frac{24 \times 0.2}{105[0.175 + 1.0(5/100)]} = 0.20 \text{ ft}^2/\text{day}$$

Then use [Equation \(1\)](#) to calculate the maximum soil temperature at the installation depth as follows:

$$t_{z,z} = 71.8 + 15.0 \exp \left[-10 \sqrt{\frac{\pi}{0.20 \times 365}} \right] = 73.7^\circ\text{F}$$

Now calculate the first estimates of the thermal resistances of the pipe insulations. For calcium silicate, assume a mean temperature of the insulation of 300°F to establish its thermal conductivity. From the data listed previously, calcium silicate's thermal conductivity $k = 0.042$ Btu/h · ft · °F at this temperature. For the polyurethane foam, assume the insulation's mean temperature is 200°F. From the data listed previously, the polyurethane foam's thermal conductivity $k_i = 0.014$ Btu/h · ft · °F at this temperature. Now calculate the insulation thermal resistances using [Equation \(8\)](#):

$$\text{Calcium silicate } R_{i,1} = \frac{\ln(0.401/0.276)}{2\pi(0.042)} = 1.42 \text{ h} \cdot \text{ft} \cdot ^\circ\text{F}/\text{Btu}$$

$$\text{Polyurethane foam } R_{i,2} = \frac{\ln(0.484/0.401)}{2\pi(0.014)} = 2.14 \text{ h} \cdot \text{ft} \cdot ^\circ\text{F}/\text{Btu}$$

Calculate the thermal resistance of the soil from [Equation \(7\)](#):

$$R_s = \frac{\ln(2 \times 10/0.526)}{2\pi(0.2)} = 2.89 \text{ h} \cdot \text{ft} \cdot ^\circ\text{F}/\text{Btu}$$

The total thermal resistance is

$$R_t = 1.42 + 2.14 + 2.89 = 6.45 \text{ h} \cdot \text{ft} \cdot ^\circ\text{F}/\text{Btu}$$

The first estimate of the heat loss is then

$$q = (400 - 73.7)/6.45 = 50.6 \text{ Btu/h} \cdot \text{ft}$$

Next, calculate the estimated insulation surface temperature with this first estimate of the heat loss. Find the temperature at the interface between the calcium silicate and polyurethane foam insulations.

$$t_{io,1} = t_{po} - qR_{i,1} = 400 - (50.6 \times 1.42) = 328^\circ\text{F}$$

where t_{io} is the outer surface temperature of the pipe and $t_{io,1}$ is the outer surface temperature of first insulation (calcium silicate).

$$t_{io,2} = t_{po} - q(R_{i,1} + R_{i,2}) = 400 - 50.6(1.42 + 2.14) = 220^\circ\text{F}$$

where $t_{io,2}$ is the outer surface temperature of second insulation (polyurethane foam).

The new estimate of the mean insulation temperature is $(400 + 328)/2 = 364^\circ\text{F}$ for the calcium silicate and $(328 + 220)/2 = 274^\circ\text{F}$ for the polyurethane foam. Thus, the insulation thermal conductivity for the calcium silicate is interpolated to be 0.045 Btu/h · ft · °F, and the resulting thermal resistance is $R_{i,1} = 1.32 \text{ h} \cdot \text{ft} \cdot ^\circ\text{F}/\text{Btu}$.

For the polyurethane foam insulation, the thermal conductivity is interpolated to be 0.015 Btu/h · ft · °F, and the resulting thermal resistance is $R_{i,2} = 2.00 \text{ h} \cdot \text{ft} \cdot ^\circ\text{F}/\text{Btu}$. The soil thermal resistance remains unchanged, and the heat loss is recalculated as $q = 52.5$ Btu/h · ft.

The calcium silicate insulation outer surface temperature is now approximately $t_{io,1} = 331^\circ\text{F}$, and the outer surface temperature of the polyurethane foam is calculated to be $t_{io,2} = 226^\circ\text{F}$. Because these temperatures are within a few degrees of those calculated previously, no further calculations are needed.

The maximum temperature of the polyurethane insulation of 331°F occurs at its inner surface (i.e., the interface with the calcium silicate insulation). This temperature clearly exceeds the maximum accepted 30-year service temperature of polyurethane foam of 250°F (CEN *Standard* EN 253). Thus, the amount of calcium silicate insulation needs to be increased significantly to achieve a maximum temperature for the polyurethane foam insulation within its long-term service temperature limit. Under the conditions of this example, it would take about 5 in. of calcium silicate insulation to reduce the insulation interface temperature to less than 250°F.

Example 6 examines the effects of soil thermal conductivity and decreasing burial depth for a system with composite insulation and an air space.

Example 6. As in Example 5, assume a high-temperature water line operating at 400°F installed in southern Texas. Here it also consists of a 6 in. nominal diameter (6.625 in. outer diameter) insulated carrier pipe, but the first layer of insulation is initially proposed to be 1.5 in. of mineral wool. The carrier pipe and mineral wool insulation are contained inside a 1 in. air space between the insulation exterior and the conduit interior. On the exterior of the steel conduit is 1 in. polyurethane foam insulation. The polyurethane insulation is encased in a 0.25 in. thick high-density polyethylene (HDPE) jacket. The piping system is buried 10 ft deep to the pipe centerline. The initial assumption for the burial depth and soil thermal properties is the same as those used in Example 5, but the effect of those design parameters will be examined. Neglect the thermal resistances of the pipe, conduit, and HDPE jacket.

Mineral wool thermal conductivity

0.026 Btu/h · ft · °F at 200°F

0.033 Btu/h · ft · °F at 300°F

0.043 Btu/h · ft · °F at 400°F

Polyurethane foam thermal conductivity

0.013 Btu/h · ft · °F at 100°F

0.014 Btu/h · ft · °F at 200°F

0.015 Btu/h · ft · °F at 275°F

Assumed soil properties

Thermal conductivity = 0.2 Btu/h · ft · °F

Density (dry soil) = 105 lb/ft³

Moisture content = 5% (dry basis)

What is the maximum operating temperature of the polyurethane foam insulation and the HDPE jacket?

Solution: As in Example 5, the maximum operating temperature of the materials is sought; thus, the maximum expected soil temperature rather than the average annual soil temperature must be used, along with the lowest anticipated soil thermal conductivity and the deepest burial depth. The effect of these design parameters will be explored briefly once the proposed design has been evaluated. Calculation of the thermal resistances of the insulations, soil, and air space follow the methods outlined in Examples 4 and 5 and thus are not repeated here in detail. [Table 10](#) summarizes results from the calculation as well as the final results for the thermal resistances after three iterations.

As shown in [Table 10](#), for the design initially proposed with 1.5 in. of mineral wool insulation, the maximum temperature of the urethane foam (at its interface with the steel casing) is 312°F. This clearly exceeds the long-term temperature capabilities of urethane foam. Additionally, note that the temperature of the HDPE casing would be 223°F, which would likely be well in excess of its capabilities, as well. [Table 7](#) provides results for other thicknesses of the mineral wool insulation; as can be seen from these results, it would require 4 in. of mineral wool insulation to reduce the maximum temperature seen by the urethane foam to within its 250°F limit for long-term exposure. Even with this insulation thickness, the HDPE casing temperature would be at 186°F, which could be problematic depending on the HDPE formulation. As will be shown by the following calculations, the casing temperature could be reduced by adding thickness to the urethane insulation. However, this has the unfortunate consequence of increasing the maximum temperature of the urethane insulation, in the absence of any other changes to the design.

The burial depth and soil thermal conductivity in this example and Example 5 could be considered extreme; however, they are encountered in practice. In fact, deeper burials have not been entirely infrequent in practice and the results of these heat transfer calculations should only reinforce the cautions later in this chapter regarding deep burial. Low soil thermal conductivity may result not just from low rainfall and deep water tables, but also soil type and the impact of the constant heat loss from the buried heat distribution system. [Table 11](#) shows the effects of soil thermal conductivity and burial depth on the design of this example.

[Table 12](#) contains the results of several calculations with increased soil thermal conductivity and reduced burial depth. For each of these cases, the thickness of the mineral wool insulation has been adjusted in 0.5 in. increments to find the minimum mineral wool insulation thickness to reduce the maximum temperature of the polyurethane insulation to or below 250°F. The results show that, where soil conditions result in the soil thermal conductivity being increased to 0.80 Btu/h · ft · °F, the thickness of the mineral wool insulation may be reduced from 4 in. to 2 in. Where the soil conditions result in a thermal conductivity of 0.2 Btu/h · ft · °F, as was the case in the original calculation of the example, if the burial depth is decreased from 10 ft to 5 ft, the thickness of the mineral wool insulation may be reduced only by 0.5 in., to 3.5 in. For a case with both lower burial depth and high soil thermal conductivity, the thickness of the mineral wool insulation still must be 2 in.; thus, the decrease in burial depth from 10 ft to 5 ft results in slightly lower maximum temperature for the urethane foam but no net reduction in the mineral wool insulation thickness required.

The last column in [Table 11](#) shows how increasing the thickness of the polyurethane insulation affects the temperature of the HDPE jacket. Although adding 1 in. of urethane insulation to bring its thickness to 2 in. reduces the maximum temperature of the HDPE jacket from 186°F to 170°F, it increases the maximum temperature of the urethane foam insulation from 241°F to 263°F, above the accepted limit of 250°F.

Note that all of the calculations in Examples 5 and 6 are based on the assumption of a single, insulated buried pipe. Where supply and return lines are buried side by side and both are operating above the surrounding soil temperature, heat losses will be reduced but the temperature of system components will be elevated. The effect can be considered as somewhat analogous to burying the pipelines in a region with higher soil temperatures, which of course also reduces their heat loss. The formulation for calculating heat losses for adjacent supply and return lines presented later in this chapter does not allow for direct calculation of system temperatures. As an approximation, it is suggested that the individual heat losses for the supply pipes, where the limiting condition will occur, be first calculated using [Equations \(18\)](#) to [\(25\)](#) (shown in Example 8). To approximate the system temperatures, use the methods outlined in this example as if the pipeline were buried by itself, but adjust the soil temperature upward to achieve the same rate of heat loss as determined by [Equations \(18\)](#) to [\(25\)](#) for the supply pipe in the supply and return configuration. This somewhat underestimates the impacts of the adjacent return pipeline, so be conservative when using this method.

Table 10. Effect of Interior Insulation Thickness on Exterior Insulation Temperature (Example 6)										
Insulation Thickness, in.				Thermal Resistances, h · ft · °F/Btu					Heat Loss, Btu/h · ft	Foam Maximum Temp., °F
Mineral Wool	Urethane Foam	Air Space, in.	Burial Depth, ft	Mineral Wool	Air Space	Urethane Foam	Soil			
1.5	1	1	10	1.52	0.132	1.66	2.79	53.4	312	223
2.5	1	1	10	2.39	0.109	1.47	2.69	49.1	278	206
3	1	1	10	2.78	0.101	1.39	2.64	47.2	264	198
3.5	1	1	10	3.16	0.0934	1.31	2.60	45.6	252	192
4	1	1	10	3.51	0.0870	1.25	2.55	44.1	241	186

Table 11. Effects of Soil Thermal Conductivity and Burial Depth on Exterior Insulation Temperature (Example 6)										
Insulation Thickness, in.				Thermal Resistances, h · ft · °F/Btu					Heat Loss, Btu/h · °F	Foam Maximum Temp., °F
Soil Thermal Conductivity, Btu/h · ft · °F	Mineral Wool	Urethane Foam	Air Space, in.	Burial Depth, ft	Mineral Wool	Air Space	Urethane Foam	Soil		

Soil Thermal Conductivity, Btu/h · ft · °F	Insulation Thickness, in.		Air Space, in.	Burial Depth, ft	Thermal Resistances, h · ft · °F/Btu				Heat Loss, Btu/h · °F	Foam Maximum Temp., °F	HDPE Casing Temp., °F
	Mineral Wool	Urethane Foam			Mineral Wool	Air Space	Urethane Foam	Soil			
0.20	4.0	1	1	10	3.51	0.0870	1.25	2.55	44.1	241	186
0.80	2.0	1	1	10	2.08	0.120	1.65	0.685	72.0	241	123
0.20	3.5	1	1	5	3.20	0.0934	1.33	2.04	48.9	239	174
0.80	2.0	1	1	5	2.09	0.120	1.65	0.547	73.9	236	114
0.20	4.0	2	1	10	3.43	0.0870	2.37	2.48	39.0	263	170

Examples 5 and 6 show the need to fully consider burial conditions when assessing systems that have limitations on any of their component temperatures that are below the carrier fluid's operating temperature. Although the examples here are from systems with multiple insulations with temperature limitation on one of the insulations, this consideration also applies to any system, with or without composite insulation. Nonmetallic jacket materials as well as corrosion-resistant coatings are other examples of materials that often have temperature limitations well below carrier pipe temperatures encountered in practice. Deep burial of systems and/or burial of systems in low-thermal-conductivity soil requires that calculations such as those from Examples 5 and 6 be conducted where materials with temperature limitations below the carrier pipe temperature exist.

Two Pipes Buried in Common Conduit with Air Space

For this case (Figure 9), make the same assumption as made in the Buried Pipe in Conduit with Air Space section. For convenience, add some of the thermal resistances as follows:

$$R_1 = R_{p1} + R_{i1} + R_{a1}$$

(10)

$$R_2 = R_{p2} + R_{i2} + R_{a2}$$

(11)

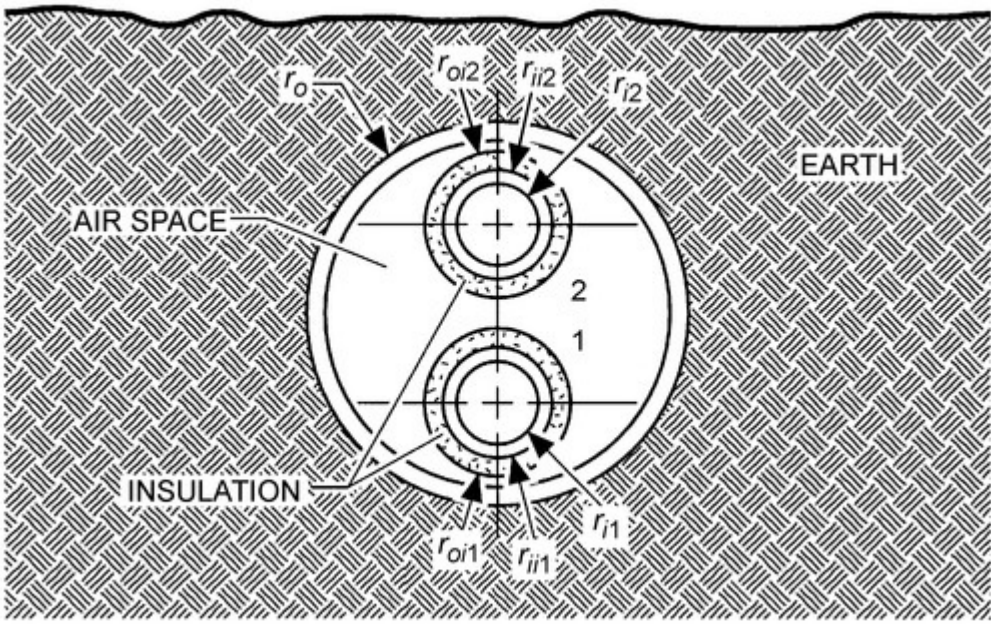


Figure 9. Two Pipes Buried in Common Conduit with Air Space

Subscripts 1 and 2 differentiate between the two pipes within the conduit. The combined heat loss is then given by

$$q = \frac{[(t_{f1} - t_s)/R_1] + [(t_{f2} - t_s)/R_2]}{1 + (R_{cs}/R_1) + (R_{cs}/R_2)}$$

(12)

where R_{cs} is the total thermal resistance of conduit shell and soil.
Once the combined heat flow is determined, calculate the bulk temperature in the air space:

$$t_a = t_s + qR_{cs}$$

(13)

$$t_{i1} = t_a + (t_{f1} - t_a)(R_{a1}/R_1)$$

(14)

$$t_{i2} = t_a + (t_{f2} - t_a)(R_{a2}/R_2)$$

(15)

The heat flows from each pipe are given by

$$q_1 = (t_{f1} - t_a)/R_1$$

(16)

$$q_2 = (t_{f2} - t_a)/R_2$$

(17)

When the insulation thermal conductivity is a function of its temperature, as is usually the case, an iterative calculation is required, as shown in the following example.
Example 7. A pair of 4 in. NPS medium-temperature hot-water supply and return lines run in a common 21 in. outside diameter conduit. Assume that the supply temperature is 325°F and the return temperature is 225°F. The supply pipe is insulated with 2.5 in. of mineral wool insulation, and the return pipe has 2 in. of mineral wool insulation. This insulation has the same thermal properties as those given in Example 4. The pipe is buried 4 ft to centerline in soil with a thermal conductivity of 1 Btu/h · ft · °F. Assume the thermal resistance of the pipe, the conduit, and the conduit coating are negligible. As a first estimate, assume the bulk air temperature within the conduit is 100°F. In addition, use this temperature as a first estimate of the insulation surface temperatures to obtain the mean insulation temperatures and subsequent insulation thermal conductivities.

Solution: By interpolation, estimate the insulation thermal conductivities from the data given in Example 4 at 0.0265 Btu/h · ft · °F for the supply pipe and 0.0245 Btu/h · ft · °F for the return pipe. Calculate the thermal resistances using Equations (10) and (11):

$$R_1 = R_{i1} + R_{a1} = \ln(0.396/0.188)/(2\pi \times 0.0265) + 0.053/0.396$$
$$= 4.48 + 0.13 = 4.61 \text{ h} \cdot \text{ft} \cdot ^\circ\text{F}/\text{Btu}$$

$$R_2 = R_{i2} + R_{a2} = \ln(0.354/0.188)/(2\pi \times 0.0245) + 0.053/0.354$$
$$= 4.11 + 0.15 = 4.26 \text{ h} \cdot \text{ft} \cdot ^\circ\text{F}/\text{Btu}$$

$$R_{cs} = R_s = \ln(8/0.875)/(2\pi \times 1) = 0.352 \text{ h} \cdot \text{ft} \cdot ^\circ\text{F}/\text{Btu}$$

Calculate first estimate of combined heat flow from Equation (12):

$$q = \frac{(325 - 60)/4.61 + (225 - 60)/4.26}{1 + (0.352/4.61) + (0.352/4.26)} = 83.0 \text{ Btu/h} \cdot \text{ft}$$

Estimate bulk air temperature in the conduit with Equation (13):

$$t_a = 60 + (83.0 \times 0.352) = 89.2^\circ\text{F}$$

Then revise estimates of the insulation surface temperatures with Equations (14) and (15):

$$t_{i1} = 89.2 + (325 - 89.2)(0.13/4.61) = 95.8^\circ\text{F}$$
$$t_{i2} = 89.2 + (225 - 89.2)(0.15/4.26) = 94.0^\circ\text{F}$$

These insulation surface temperatures are close enough to the original estimate of 100°F that further iterations are not warranted. If the individual supply and return heat losses are desired, calculate them using Equations (16) and (17).

Two Buried Pipes or Conduits

This case (Figure 10) may be formulated in terms of the thermal resistances used for a single buried pipe or conduit and some correction factors. The correction factors needed are

$$\theta_1 = (t_{p2} - t_s)(t_{p1} - t_s)$$

(18)

$$\theta_2 = 1/\theta_1 = (t_{p1} - t_s)/(t_{p2} - t_s)$$

(19)

$$P_1 = \frac{1}{2\pi k_s} \ln \left[\frac{(d_1 + d_2)^2 + a^2}{(d_1 - d_2)^2 + a^2} \right]^{0.5}$$

(20)

$$P_2 = \frac{1}{2\pi k_s} \ln \left[\frac{(d_2 + d_1)^2 + a^2}{(d_2 - d_1)^2 + a^2} \right]^{0.5}$$

(21)

where a = horizontal separation distance between centerline of two pipes, ft.

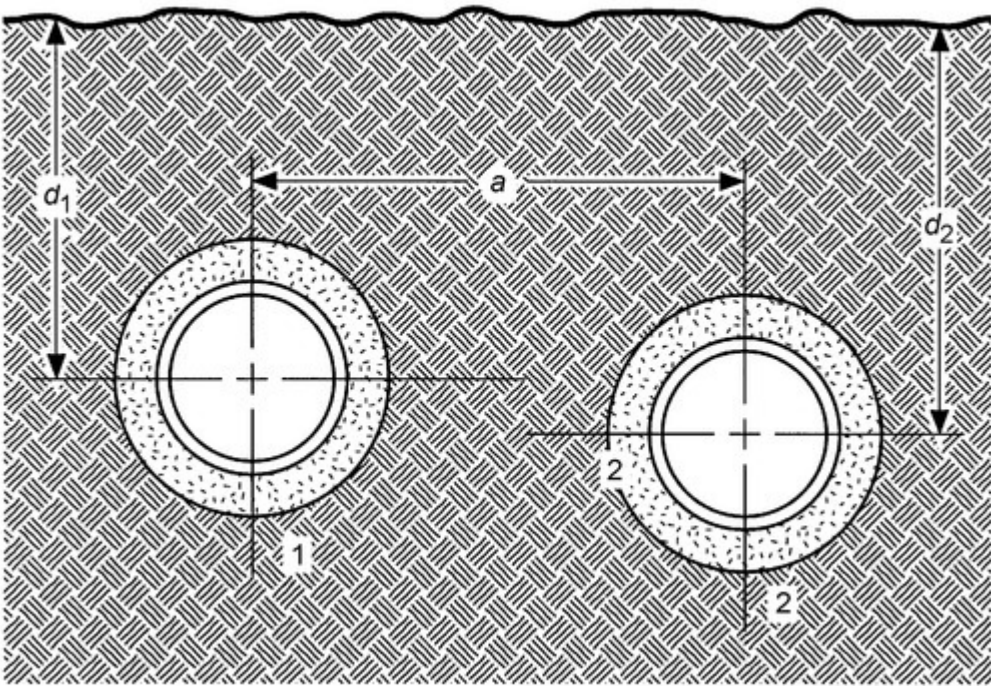


Figure 10. Two Buried Pipes or Conduits

The thermal resistance for each pipe or conduit is given by

$$R_{e1} = \frac{R_{i1} - (P_1^2/R_{i2})}{1 - (P_1\theta_1/R_{i2})}$$

(22)

$$R_{e2} = \frac{R_{i2} - (P_2^2/R_{i1})}{1 - (P_2\theta_2/R_{i1})}$$

(23)

where
 θ = temperature correction factor, dimensionless
 P = geometric/material correction factor, h · ft · °F/Btu
 R_e = effective thermal resistance of one pipe/conduit in two-pipe system, h · ft · °F/Btu
 R_i = total thermal resistance of one pipe/conduit if buried separately, h · ft · °F/Btu
Heat flow from each pipe is then calculated from

$$q_1 = (t_{p1} - t_s)/R_{e1}$$

(24)

$$q_2 = (t_{p2} - t_s)/R_{e2}$$

(25)

Example 8. Consider buried supply and return lines for a low-temperature hot-water system. The carrier pipes are 4 in. NPS (4.5 in. outer diameter) with 1.5 in. of urethane foam insulation. The insulation is protected by a 0.25 in thick PVC jacket. The thermal conductivity of the insulation is 0.013 Btu/h · ft · °F and is assumed constant with respect to temperature. The pipes are buried 4 ft deep to the centerline in soil with a thermal conductivity of 1 Btu/h · ft · °F and a mean annual temperature of 60°F. The horizontal distance between the pipe centerlines is 2 ft. The supply water is at 250°F, and the return water is at 150°F.

Solution: Neglect the thermal resistances of the carrier pipes and the PVC jacket. First, calculate the resistances from Equations (7) and (8) as if the pipes were independent of each other:

$$R_{s1} = R_{s2} = \frac{\ln(8.0/0.333)}{2\pi \times 1.0} = 0.51 \text{ h} \cdot \text{ft} \cdot ^\circ\text{F}/\text{Btu}$$

$$R_{i1} = R_{i2} = \frac{\ln(0.313/0.188)}{2\pi \times 0.013} = 6.25 \text{ h} \cdot \text{ft} \cdot ^\circ\text{F}/\text{Btu}$$

$$R_{t1} = R_{t2} = 0.51 + 6.25 = 6.76 \text{ h} \cdot \text{ft} \cdot ^\circ\text{F}/\text{Btu}$$

From Equations (20) and (21), the correction factors are

$$P_1 = P_2 = \frac{1}{2\pi \times 1} \ln \left[\frac{(4 + 4)^2 + 2^2}{(4 - 4)^2 + 2^2} \right]^{0.5} = 0.225 \text{ h} \cdot \text{ft} \cdot ^\circ\text{F}/\text{Btu}$$

$$\theta_1 = (150 - 60)(250 - 60) = 0.474$$

$$\theta_2 = 1/\theta_1 = 2.11$$

Calculate the effective total thermal resistances as

$$R_{e1} = \frac{6.76 - (0.225^2/6.76)}{1 - (0.225 \times 0.474/6.76)} = 6.87 \text{ h} \cdot \text{ft} \cdot ^\circ\text{F}/\text{Btu}$$

$$R_{e2} = \frac{6.76 - (0.225^2/6.76)}{1 - (0.225 \times 2.11/6.76)} = 7.32 \text{ h} \cdot \text{ft} \cdot ^\circ\text{F}/\text{Btu}$$

The heat flows are then

$$q_1 = (250 - 60)/6.87 = 27.7 \text{ Btu/h} \cdot \text{ft}$$

$$q_2 = (150 - 60)/7.32 = 12.3 \text{ Btu/h} \cdot \text{ft}$$

$$q_t = 27.7 + 12.3 = 40.0 \text{ Btu/h} \cdot \text{ft}$$

Note that when the resistances and geometry for the two pipes are identical, the total heat flow from the two pipes is the same if the temperature corrections are used or if they are set to unity. The individual losses vary somewhat, however. These equations may also be used with air space systems. When the thermal conductivity of the pipe insulation is a function of temperature, iterative calculations must be done.

Pipes in Buried Trenches or Tunnels

Buried rectangular trenches or tunnels (Figure 11) require several assumptions to obtain approximate solutions for the heat transfer. First, assume that air within the tunnel or trench is uniform in temperature and that the same is true for the inside surface of the trench/tunnel walls. Field measurements by Phetteplace et al. (1991) on operating shallow trenches indicate maximum spatial air temperature variations of about 10°F. Air temperature variations of this magnitude in a tunnel or trench do not cause significant errors for normal operating temperatures when using the following calculation methods.

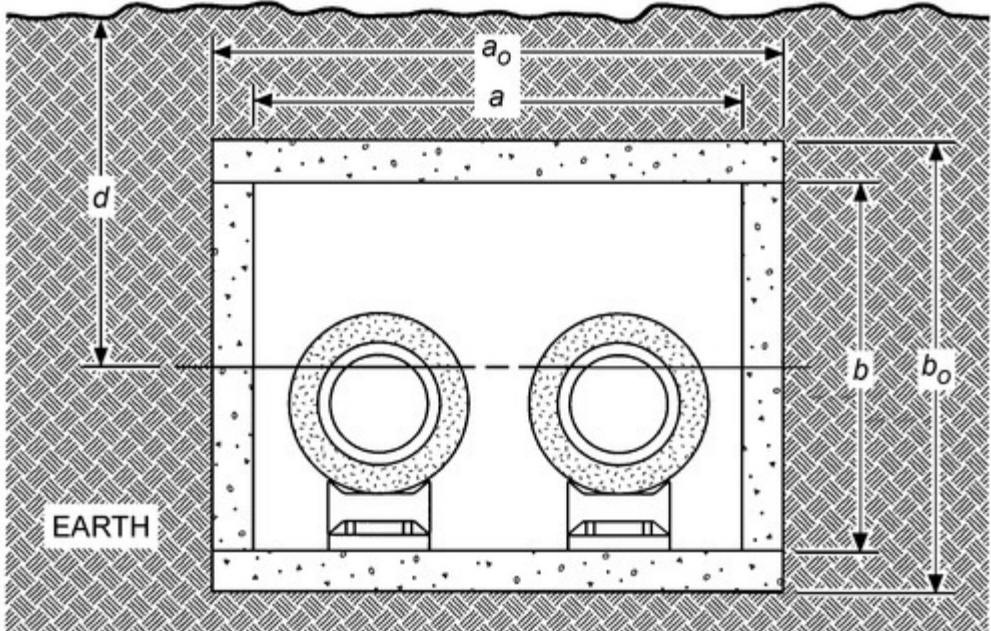


Figure 11. Pipes in Buried Trenches or Tunnels

Unless numerical methods are used, the resistance of a rectangular region such as the walls of a trench or tunnel must be approximated. One procedure is to assume linear heat flow through the trench or tunnel walls, which yields the following resistance for the walls (Phetteplace et al. 1981):

$$R_w = \frac{x_w}{2k_w(a + b)}$$

(26)

where
 R_w = thermal resistance of trench/tunnel walls, h · ft · °F/Btu

x_w = thickness of trench/tunnel walls, ft
 a = width of trench/tunnel inside, ft
 b = height of trench/tunnel inside, ft
 k_w = thermal conductivity of trench/tunnel wall material, Btu/h · ft · °F

As an alternative to Equation (26), the thickness of the trench/tunnel walls may be included in the soil burial depth. This approximation is only acceptable when the thermal conductivity of the trench/tunnel wall material is similar to that of the soil. The thermal resistance of the soil surrounding the buried trench/tunnel is calculated using the following equation (Rohsenow 1998):

$$R_{ts} = \frac{\ln[3.5d/(b_o^{0.75}a_o^{0.25})]}{k_s[(a_o/2b_o) + 5.7]} \qquad a_o > b_o$$

(27)

where
 R_{ts} = thermal resistance of soil surrounding trench/tunnel, h · ft · °F/Btu
 a_o = width of trench/tunnel outside, ft
 b_o = height of trench/tunnel outside, ft
 d = burial depth of trench to centerline, ft

Equations (26) and (27) can be combined with the equations already presented to calculate heat flow and temperature for trenches/tunnels. As with the conduits described in earlier examples, the heat transfer processes in the air space inside the trench/tunnel are too complex to warrant a complete treatment for design purposes. The thermal resistance of this air space may be approximated by several methods (e.g., by using Equation (9)). Thermal resistances at the pipe insulation/air interface can also be calculated from heat transfer coefficients as done in the section on Pipes in Air. If the thermal resistance of the air/trench wall interface is also included, use Equation (28):

$$R_{aw} = 1/[2h_i(a + b)]$$

(28)

where
 R_{aw} = thermal resistance of air/trench wall interface, h · ft · °F/Btu
 h_i = total heat transfer coefficient at air/trench wall interface, Btu/h · ft² · °F
The total heat loss from the trench/tunnel is calculated from the following relationship:

$$q = \frac{(t_{p1} - t_s)/R_1 + (t_{p2} - t_s)/R_2}{1 + (R_{ss}/R_1) + (R_{ss}/R_2)}$$

(29)

where
 R_1, R_2 = thermal resistances of two-pipe/insulation systems within trench/tunnel, h · ft · °F/Btu
 R_{ss} = total thermal resistance on soil side of air within trench/tunnel, h · ft · °F/Btu
Once the total heat loss has been found, the air temperature in the trench/tunnel may be found as

$$t_{ta} = t_s + qR_{ss}$$

(30)

where t_s is air temperature in the trench/tunnel.
The individual heat flows for the two pipes in the trench/tunnel are then

$$q_1 = (t_{p1} - t_{ta})/R_1$$

(31)

$$q_2 = (t_{p2} - t_{ta})/R_2$$

(32)

If the thermal conductivity of the pipe insulation is a function of temperature, assume an air temperature for the air space before starting calculations. Iterate the calculations if the air temperature calculated with Equation (30) differs significantly from the initial assumption.
Example 9. The walls of a buried trench are 6 in. thick, and the trench is 3 ft wide and 2 ft tall. The trench is constructed of concrete, with a thermal conductivity of $k_w = 1$ Btu/h · ft · °F. The soil surrounding the trench also has a thermal conductivity of $k_s = 1$ Btu/h · ft · °F. The centerline of the trench is 4 ft below grade, and the soil temperature is assumed be 60°F. The trench contains supply and return lines for a medium-temperature water system with the physical and operating parameters identical to those in Example 7.
Solution: Assuming the air temperature within the trench is 100°F, the thermal resistances for the pipe/insulation systems are identical to those in Example 7, or

$$R_1 = 4.61 \text{ h} \cdot \text{ft} \cdot ^\circ\text{F}/\text{Btu}$$
$$R_2 = 4.26 \text{ h} \cdot \text{ft} \cdot ^\circ\text{F}/\text{Btu}$$

The thermal resistance of the soil surrounding the trench is given by Equation (27):

$$R_{ts} = \frac{\ln[(3.5 \times 4)/(3^{0.75} \times 4^{0.25})]}{1[(4/6) + 5.7]} = 0.231 \text{ h} \cdot \text{ft} \cdot ^\circ\text{F}/\text{Btu}$$

The trench wall thermal resistance is calculated with Equation (26):

$$R_w = 0.5[2(3 + 2)] = 0.050 \text{ h} \cdot \text{ft} \cdot ^\circ\text{F}/\text{Btu}$$

If the thermal resistance of the air/trench wall is neglected, the total thermal resistance on the soil side of the air space is

$$R_{ss} = R_w + R_{ts} = 0.050 + 0.231 = 0.281 \text{ h} \cdot \text{ft} \cdot ^\circ\text{F}/\text{Btu}$$

Find a first estimate of the total heat loss using Equation (29):

$$q = \frac{(325 - 60)/4.61 + (225 - 60)/4.26}{1 + (0.281/4.61) + (0.281/4.26)} = 85.4 \text{ Btu/h} \cdot \text{ft}$$

The first estimate of the air temperature in the trench is given by Equation (30):

$$t_{ta} = 60 + (85.4 \times 0.281) = 84.0^\circ\text{F}$$

$$t_{t1} = 84.0 + [(325 - 84.0)(0.13/4.61)] = 90.8^\circ\text{F}$$
$$t_{t2} = 84.0 + [(225 - 84.0)(0.15/4.26)] = 89.0^\circ\text{F}$$

From these estimates, calculate the revised mean insulation temperatures to find the resultant resistance values. Repeat the calculation procedure until satisfactory agreement between successive estimates of the trench air temperature is obtained. Calculate the individual heat flows from the pipes with Equations (31) and (32).
If the thermal resistance of the trench walls is added to the soil thermal resistance, the thermal resistance on the soil side of the air space is

$$R_{ss} = \frac{\ln[14/(2^{0.75} \times 3^{0.25})]}{1[(3/4) + 5.7]} = 0.286 \text{ h} \cdot \text{ft} \cdot ^\circ\text{F}/\text{Btu}$$

The result is less than 2% higher than the resistance previously calculated by treating the trench walls and soil separately. In the event that the soil and trench wall material have significantly different thermal conductivities, this simpler calculation will not yield as favorable results and should not be used.

Pipes in Shallow Trenches

The cover of a shallow trench is exposed to the environment. Thermal calculations for such trenches require the following assumptions: (1) the interior air temperature is uniform as discussed in the section on Pipes in Buried Trenches or Tunnels, and (2) the soil and the trench wall material have the same thermal conductivity. This assumption yields reasonable results if the thermal conductivity of the trench material is used, because most of the heat flows directly through the cover. The thermal resistance of the trench walls and surrounding soil is usually a small portion of the total thermal resistance, and thus the heat losses are not usually highly dependent on this thermal resistance. Using these assumptions, Equations (22) and (29) to (32) may be used for shallow trench systems.

Example 10. Consider a shallow trench having the same physical parameters and operating conditions as the buried trench in Example 9, except that the top of the trench is at grade level. Calculate the thermal resistance of the shallow trench using Equation (27):

$$R_{ts} = R_{ss} = \frac{\ln[(3.5 \times 1.5)/(2^{0.75} \times 3^{0.25})]}{1[(3/4) + 5.7]} = 0.134 \text{ h} \cdot \text{ft} \cdot ^\circ\text{F}/\text{Btu}$$

Use thermal resistances for the pipe/insulation systems from Example 9, and use Equation (24) to calculate q :

$$q = \frac{(325 - 60)/4.61 + (225 - 60)/4.26}{1 + (0.134/4.61) + (0.134/4.26)} = 90.7 \text{ Btu/h} \cdot \text{ft}$$

From this, calculate the first estimate of the air temperature, using Equation (30):

$$t_{ta} = 60 + (90.7 \times 0.134) = 72.2^\circ\text{F}$$

Then refine estimates of the pipe insulation surface temperatures using Equations (14) and (15):

$$t_{t1} = 72.2 + [(325 - 72.2)(0.13/4.61)] = 79.3^\circ\text{F}$$
$$t_{t2} = 72.2 + [(225 - 72.2)(0.15/4.26)] = 77.6^\circ\text{F}$$

From these, calculate the revised mean insulation temperatures to find resultant resistance values. Repeat the calculation procedure until satisfactory agreement between successive estimates of the trench air temperatures are obtained. If the individual heat flows from the pipes are desired, calculate them using Equations (31) and (32).

Another method for calculating heat losses in a shallow trench assumes an interior air temperature and treats the pipes as pipes in air (see the section on Pipes in Air). Interior air temperatures in the range of 70 to 120°F have been observed in a temperate climate (Phetteplace et al. 1991). It may be necessary to drain the system or provide heat tracing in areas with significant subfreezing air temperatures in winter, especially for shallow trenches that contain only chilled-water lines (and thus have no source of heat) if the system is not in operation, or is in operation at very low flow rates.

Buried Pipes with Other Geometries

Other geometries not specifically addressed by the previous cases have been used for buried thermal utilities. In some instances, the equations presented previously may be used to approximate the system. For instance, the soil thermal resistance for a buried system with a half-round day tile on a concrete base could be approximated as a circular system using Equation (6) or (7). In this case, the outer radius r_o is taken as that of a cylinder with the same circumference as the outer perimeter of the day tile system. The remainder of the resistances and subsequent calculations would be similar to those for a buried trench/tunnel. The accuracy of such calculations varies inversely with the proportion of the total thermal resistance that the thermal resistance in question comprises. In most instances, the thermal resistance of the pipe insulation overshadows other resistances, and the errors induced by approximations in the other resistances are acceptable for design calculations with appropriate conservative assumptions applied.

Pipes in Air

Pipes surrounded by gases may transfer heat via conduction, convection, and/or thermal radiation. Heat transfer modes depend mainly on the surface temperatures and geometry of the system being considered. For air, conduction is usually dominated by the other modes and thus may be neglected. Thermal resistances for cylindrical systems can be found from heat transfer coefficients derived using the following equations. Generally, piping systems used for thermal distribution have sufficiently low surface temperatures to preclude any significant heat transfer by thermal radiation.

Example 11. Consider a 6 in. nominal (6.625 in. outside diameter), high-temperature hot-water pipe that operates in air at 375°F with 2.5 in. of mineral wool insulation. The surrounding air annually averages 60°F (520°R). The average annual wind speed is 4 mph. The insulation is covered with an aluminum jacket with an emittance $\epsilon = 0.26$. The thickness and thermal resistance of the jacket material are negligible. Because the heat transfer coefficient at the outer surface of the insulation is a function of the temperature there, initial estimates of this temperature must be made. This temperature estimate is also required to estimate mean insulation temperature.
Solution: Assuming that the insulation surface is at 100°F (560°R) as a first estimate, the mean insulation temperature is calculated as 238°F. Using the properties of mineral wool given in Example 4, the insulation thermal conductivity is $k_i = 0.0275$ Btu/h · ft · °F. Using Equation (8), the thermal resistance of the insulation is $R_i = 3.25$ h · ft · °F/Btu. The forced convective heat transfer coefficient at the surface of the insulation can be found using the following equation (ASTM Standard C680):

$$h_{cv} = 1.016 \left(\frac{1}{d} \right)^{0.2} \left(\frac{2}{t_{oi} + t_g} \right)^{0.181} (t_{ta} - t_g)^{0.266} (1 + 1.277 V')^{0.5}$$
$$= 1.016 \left(\frac{1}{11.625} \right)^{0.2} \left(\frac{2}{160} \right)^{0.181} (40)^{0.266} (6.11)^{0.5}$$
$$= 1.86 \text{ Btu/h} \cdot \text{ft}^2 \cdot ^\circ\text{F}$$

where
 d = outer diameter of surface, in.
 t_a = ambient air temperature, °F
 V = wind speed, mph
The radiative heat transfer coefficient must be added to this convective heat transfer coefficient. Determine the radiative heat transfer coefficient as follows (ASTM Standard C680):

$$h_{rad} = \epsilon \sigma \left(\frac{T_a^4 - T_s^4}{T_a - T_s} \right)$$
$$h_{rad} = 0.26 \times 1.713 \times 10^{-9} \frac{(560^4 - 520^4)}{560 - 520} = 0.28 \text{ Btu/h} \cdot \text{ft}^2 \cdot ^\circ\text{F}$$

Add the convective and radiative coefficients to obtain a total surface heat transfer coefficient h_o of 2.14 Btu/h · ft² · °F. The equivalent thermal resistance of this heat transfer coefficient is calculated from the following equation:

$$R_{surf} = \frac{1}{2\pi r_{oi} h_o} = \frac{1}{2\pi \times 0.484 \times 2.14} = 0.15 \text{ h} \cdot \text{ft} \cdot ^\circ\text{F}/\text{Btu}$$

With this, the total thermal resistance of the system becomes $R_t = 3.40$ h · ft · °F/Btu, and the first estimate of the heat loss is $q = 92.6$ Btu/h · ft.
An improved estimate of the insulation surface temperature is $t_w = 375 - (92.6 \times 3.25) = 74^\circ\text{F}$. From this, a new mean insulation temperature, insulation thermal resistance, and surface resistance can be calculated. The heat loss is then 90.0 Btu/h · ft, and the insulation surface temperature is calculated as 77°F. These results are close enough to the previous results that further iterations are not warranted.
Note that the contribution of thermal radiation to the heat transfer could have been omitted with negligible effect on the results. In fact, the entire surface resistance could have been neglected and the resulting heat loss would have increased by only about 4%.

In Example 11, the convective heat transfer was forced. In cases with no wind, where the convection is free rather than forced, the radiative heat transfer is more significant, as is the total thermal resistance of the surface. However, in instances where the piping is well insulated, the thermal resistance of the insulation dominates, and minor resistances can often be neglected with little resultant error. By neglecting resistances, a conservative result is obtained (i.e., the heat transfer is overpredicted).

Economical Thickness for Pipe Insulation

A life-cycle cost analysis may be run to determine the economical thickness of pipe insulation. Because the insulation thickness affects other parameters in some systems, each insulation thickness must be considered as a separate system. For example, a conduit system or one with a jacket around the insulation requires a larger conduit or jacket for greater insulation thicknesses. The cost of the extra conduit or jacket material may exceed that of the additional insulation and is therefore usually included in the analysis. It is usually not necessary to include excavation, installation, and backfill costs in the analysis.

A system's life-cycle cost is the sum of the initial capital cost and the present worth of the subsequent cost of heat lost or gained over the life of the system. The initial capital cost needs only to include those costs that are affected by insulation thickness. The following equation can be used to calculate the life-cycle cost:

$$LCC = CC + (q_i C_i \text{PWF})$$

(33)

where
 LCC = present worth of life-cycle costs associated with pipe insulation thickness, \$/ft
 CC = capital costs associated with pipe insulation thickness, \$/ft
 q = annual average rate of heat loss, Btu/h · ft
 t_u = utilization time for system each year, h
 C_i = cost of heat lost from system, \$/Btu
 PWF = present worth factor for future annual heat loss costs, dimensionless
The present worth factor is the reciprocal of the capital recovery factor, which is found from the following equation:

$$\text{CRF} = \frac{i(1 + i)^n}{(1 + i)^n - 1}$$

(34)

where
 CRF = cost recovery factor, dimensionless
 i = interest rate
 n = useful lifetime of system, years
If heat costs are expected to escalate, the present worth factor may be multiplied by an appropriate escalation factor and the result substituted in place of PWF in Equation (33).

Example 12. Consider a steel conduit system with an air space. The insulation is mineral wool with thermal conductivity as given in Examples 4 and 5. The carrier pipe is 4 in. NPS and operates at 350°F for the entire year (8760 h). The conduit is buried 4 ft to the centerline in soil with a thermal conductivity of 1 Btu/h · ft · °F and an annual average temperature of 60°F. Neglect the thermal resistance of the conduit and carrier pipe. The useful lifetime of the system is assumed to be 20 years and the interest rate is taken as 10%.
Solution: Find CRF from Equation (34):

$$\text{CRF} = \frac{0.10(1 + 0.10)^{20}}{(1 + 0.10)^{20} - 1} = 0.11746$$

The value C_o of heat lost from the system is assumed to be \$10 per million Btu. The following table summarizes the heat loss and cost data for several available insulation thicknesses.

Insulation thickness, in.	1.5	2.0	2.5	3.0	3.5	4.0
Insulation outer radius, ft	0.313	0.354	0.396	0.438	0.479	0.521
Insulation k_i , Btu/h · ft · °F	0.027	0.027	0.027	0.027	0.027	0.027
R_i , Eq. (8), h · ft · °F/Btu	3.00	3.73	4.39	4.99	5.52	6.00
R_i , Eq. (9), h · ft · °F/Btu	0.17	0.15	0.13	0.12	0.11	0.10
Conduit outer radius, ft	0.448	0.448	0.531	0.531	0.583	0.667
R_r , Eq. (7), h · ft · °F/Btu	0.46	0.46	0.43	0.43	0.42	0.40
R_r , h · ft · °F/Btu	3.63	4.34	4.95	5.54	6.04	6.50
q , heat loss rate, Btu/h · ft	79.9	66.8	58.6	52.3	48.0	44.6
Conduit system cost, \$/ft	23.00	24.50	28.25	30.00	33.00	40.00
ft	82.59	74.32	71.95	69.00	68.80	73.27

The table indicates that 3.5 in. of insulation yields the lowest life-cycle cost for the example. Because the results depend highly on the economic parameters used, they must be accurately determined.

3.4 EXPANSION PROVISIONS

All piping expands or contracts because of temperature changes, whether it contains chilled water, steam, or hot water. The piping's length increases or decreases with its temperature. Field conditions and the type of system govern the method used to absorb the movement. Turns where the pipe changes direction must be used to provide flexibility. When the distance between changes in direction becomes too large for the turns to compensate for movement, add expansion loops at appropriate locations (this may require additional right-of-way). If field conditions allow, use the piping's flexibility to allow expansion. Where space constraints do not allow expansion loops and/or changes in direction, mechanical methods, such as expansion joints or ball joints, must be used. However, because ball joints change the direction of a pipe, a third joint may be required to reduce the length between changes in direction.

Chapter 46 covers the design of the pipe bends, loops, and the use of expansion joints. However, the chapter uses conservative stress values. Computer-aided design programs that calculate stress from pressure, thermal expansion, and weight simultaneously allow the designer to meet the requirements of ASME Standard B31.1. When larger pipe diameters are required, a computer program should be used to determine whether the pipe will provide the required flexibility. For example, Table 11 in Chapter 46 indicates that a 12 in. standard weight pipe with 12 in. of movement requires a 15.5 ft wide by 31 ft high expansion loop. One program recommends a 13 ft wide by 26 ft high loop; if equal height and width is specified, the loop is 23.5 ft in each direction.

Although the inherent flexibility of the piping should be used to handle expansion as much as possible, expansion joints must be used where space is too small to allow a loop to be constructed to handle the required movement. For example, expansion joints are often used in walk-through tunnels and vaults because there is seldom space to construct pipe loops. Either pipe loops or expansion joints can be used for aboveground, concrete shallow trench, direct-buried, and poured-envelope systems. The manufacturer of the conduit or envelope material should design loops and offsets in conduit and poured-envelope systems because clearance and design features are critical to the performance of both the loop and the pipe.

All expansion joints require maintenance, and should therefore always be accessible for service. Joints in direct-buried and poured-envelope systems and trenches without removable covers should be located in access ports.
Cold springing is normally used when thermal expansion compensation is used. In DHC systems with natural flexibility, cold springing minimizes the clearance required for pipe movement only. The pipes are sprung 50% of the total amount of movement, toward the anchor. However, ASME Standard B31.1 does not allow cold springing in calculating the stresses in the piping. When expansion joints are used, they are installed in an extended position to achieve maximum movement. Contact the manufacturer of the expansion joint for the proper amount of extension.

In extremely hot climates, anchors may also be required, to compensate for pipe contraction when pipes are installed in high ambient temperatures and then filled with cold water. This can affect buried tees in the piping, especially at branch service line runouts to buildings. Crushable insulation may be used in the trench as part of the backfill, to compensate for the contractions. Anchors should be sized using computer-aided design software.

Pipe Supports, Guides, and Anchors

For premanufactured conduit and poured-envelope systems, the system manufacturer usually designs the pipe supports, guides, and anchors in consultation with the expansion joint manufacturer, if such devices are used. For example, the main anchor force of an in-line axial expansion joint is the sum of the pressure thrust (system pressure times the cross-sectional area of the expansion joint and the joint friction or spring force) and the pipe friction forces. Consult the manufacturer of the expansion device when determining anchor forces. Anchor forces are normally less when expansion is absorbed through the system instead of with expansion joints.

Pipe guides used with expansion joints should be spaced according to the manufacturer's recommendations. They must allow longitudinal or axial motion and restrict motion perpendicular to the axis of the pipe. Guides with graphite or low-friction fluorocarbon slide surfaces are often desirable for long pipe runs (Figure 12). In addition, these surface finishes do not corrode or increase sliding resistance in aboveground installations. Select guides to handle twice the expected movement, so they may be installed in a neutral position without the need for cold-springing the pipe. A computerized stress analysis program can help the designer calculate stresses and moments in the piping system to adequately size any anchors and anchor blocks to ensure compliance with ASME Standard B31.1.

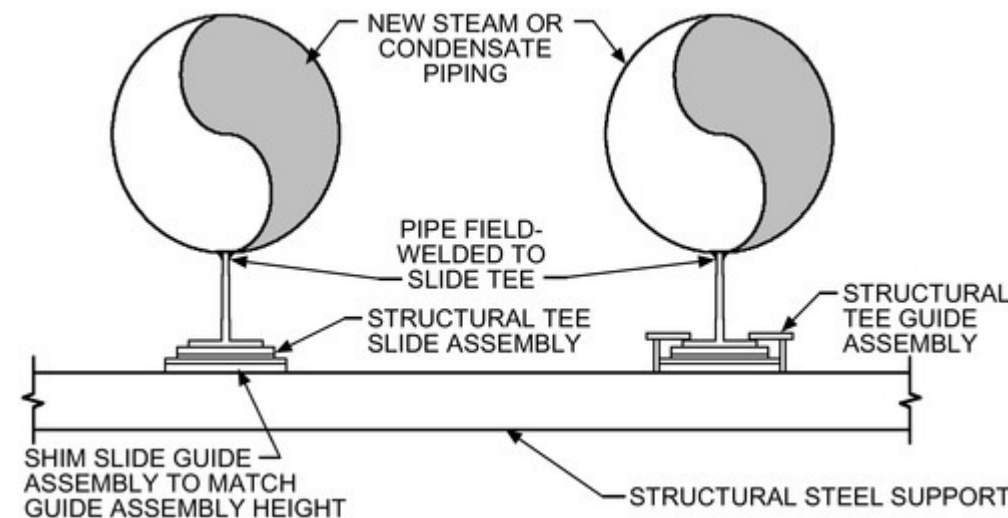


Figure 12. Slide and Guide Detail

3.5 DISTRIBUTION SYSTEM CONSTRUCTION

The combination of aesthetics, first cost, safety, and life-cycle cost naturally divide distribution systems into two distinct categories: aboveground and underground. The materials needed to ensure long life and low heat loss further classify DHC systems into low-, medium-, and high-temperature systems. The temperature range for medium-temperature systems is usually too high for materials used in low-temperature systems; however, the same materials used in high-temperature systems are typically used for medium-temperature systems. Because low-temperature systems have a lower temperature differential between the working fluid temperatures and the environment, heat loss is inherently less. In addition, the options for efficient insulation materials and inexpensive pipe materials that resist corrosion are much greater for low-temperature systems.

The aboveground system has the lowest first cost and the lowest life-cycle cost because it can be maintained easily 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 and the risk of vehicle damage to the aboveground system remove it from contention for many projects. Although the aboveground system is sometimes partially factory prefabricated, more typically it is entirely field fabricated of components such as pipes, insulation, pipe supports, and insulation jackets or protective enclosures that are commercially available. Other common systems that are completely field fabricated include walk-through tunnels (see Figure 14), concrete surface trenches (see Figure 15), deep-burial small tunnels (see Figure 16), and underground systems that use poured insulation (see Figure 17) or rigid closed-cell insulation (e.g., cellular glass) (see Figure 18) to form an envelope around the carrier pipes.

Field-assembled systems must be designed in detail, and all materials must be specified by the project design engineer. Evaluation of the project site conditions indicates which type of system should be considered for the site. For instance, the shallow trench system is best where utilities that are buried deeper than the trench bottom need to be avoided and where the covers can serve as sidewalks. Direct-buried conduit, with a thicker steel casing coated in either epoxy or HDPE or wrapped in fiberglass-reinforced plastic (FRP), may be the only system that can be buried where the conduit is in direct contact with groundwater. The conduit system is used where aesthetics is important. It is often used for short distances between buildings and the main distribution system, and where the owner is willing to accept higher life-cycle costs.

Direct-buried conduit, concrete surface trench, and other underground systems must be routed to avoid existing utilities, which requires a detailed site survey and considerable design effort. Exterior surface temperature of the conduit is an important issue that must be addressed when selecting the appropriate construction materials and components that compose the conduit; temperature may be calculated using the methods previously outlined. In the absence of a detailed soil temperature distribution study, direct-buried heating systems should be spaced more than 15 ft from other utilities constructed of plastic pipes because the temperature of the soil during dry conditions may be high enough to reduce the strength of plastic pipe to an unacceptable level. The strength of PVC at 140°F, for example, is only 20% of its room-temperature value. Rigid, extruded polystyrene insulation board may be used to insulate adjacent utilities from the impact of a buried heat distribution pipe; however, the temperature limit of the extruded polystyrene insulation must not be exceeded. Numerical analysis of the thermal problem may be required to ensure that the desired effect is achieved. Waterproofing the concrete trench or tunnel depends on the groundwater level. There are many methods of waterproofing belowground concrete structures, including membranes and concrete additives. Carefully address these issues so that the concrete structure outlives the piping system.

Tunnels that provide walk-through or crawl-through access can be buried in nearly any location without causing future problems because utilities are typically placed in the tunnel. Regardless of the type of construction, it is usually cost effective to route distribution piping through the basements of buildings, but only after addressing liability issues. In laying out the main supply and return piping, consider redundancy of supply and return. 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.

Access ports for underground systems should be 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
- Elevation changes in the distribution system that are needed to maintain the required constant slope
- Major branches with isolation valves
- Steam traps and condensate drainage points on steam lines
- Mechanical expansion devices

To facilitate leak location and repair and to limit damage caused by leaks, access points generally should be spaced no farther than 500 ft apart. Pay special attention to the safety of personnel who come in contact with distribution 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-temperature or high-pressure piping can significantly affect the access design, which must be addressed by the designer. Gravity venting of tunnels is good practice, and access ports should have lighting and convenience outlets to aid in inspection and maintenance of anchors, expansion joints, and piping.

Piping Materials and Standards

Supply Pipes for Steam and Hot Water. Adequate temperature and pressure ratings for the intended service should be specified for any piping. All piping, fittings, and accessories should be in accordance with ASME *Standard* B31.1 or with local requirements if more stringent. For steam and hot water, all joints should be welded and pipe should conform to either ASTM *Standard* A53 seamless or ERW, Grade B; or ASTM *Standard* A106 seamless, Grade B. Do not use ASTM *Standard* A53, Type F, because of its lower allowable stress and because of how its seams are manufactured. Mechanical joints of any type are not recommended for steam or hot water. Pipe wall thickness is determined by the maximum operating temperature and pressure. In the United States, most piping for steam and hot water is Schedule 40 for 10 in. NPS and below and standard weight for 12 in. NPS and above.

Many European low-temperature water systems have piping with a wall thickness similar to Schedule 10. The reduced piping material in these systems means they are not only less expensive, but they also develop reduced expansion forces and thus require simpler methods of expansion compensation, a point to consider when choosing between a high- and low-temperature system. Welding pipes with thinner walls requires extra care and may require additional inspection. Also, extra care is required to avoid internal and external corrosion because the thinner wall provides a much lower corrosion allowance.

Condensate Return Pipes. Condensate pipes require special consideration because condensate is much more corrosive than steam. This corrosiveness is caused by the oxygen that the condensate accumulates. The usual method used to compensate for condensate corrosiveness is to select steel pipe that is thicker than the steam pipe. For highly corrosive condensate, stainless steel and/or other corrosion-resistant materials should be considered. Materials that are corrosion resistant in air may not be corrosion resistant when exposed to condensate; therefore, a material with good experience handling condensate should be selected. FRP or glass-reinforced plastic (GRP) pipe used for condensate return has not performed well and should be avoided. Failed steam traps, pipe resin solubility, deterioration at elevated temperatures, and thermal expansion are thought to be the causes of the premature failures seen.

Chilled-Water Distribution. For chilled-water systems, a variety of pipe materials, such as steel, ductile iron, reinforced concrete, HDPE, PVC, and FRP/GRP, have been used successfully. If ductile iron or steel is used, the designer must resolve the internal and external corrosion issue, which may be significant unless cement-lined ductile iron is used. Soil temperature is usually highest when chilled-water loads are highest; therefore, it is usually life-cycle cost-effective to insulate the chilled-water pipe. Phetteplace et al. (2013a) has more information on thermal design of chilled-water distribution systems, including the effects of surface type (e.g., pavement, grass). Life-cycle cost analysis often favors a factory-prefabricated product insulated with sprayed or injected foam with a waterproof casing or a field-fabricated system insulated with rigid closed-cell insulation. If a nonmetallic product is selected, be careful to maintain an adequate distance from any high-temperature underground system that may be near. Damage to the chilled-water system would most likely occur from elevated soil temperatures when the chilled-water circulation stopped. Proper pressure relief devices should be designed into all systems to compensate for the increase in system pressure caused by increases in fluid temperatures.

Uninsulated chilled-water piping placement should take into account the proximity of any heating piping, especially pipes that are poorly insulated and old. Consider increasing the distance between these piping systems, especially for nonmetallic piping. The heating distribution system should be at least 15 ft from a chilled-water system containing plastic unless a detailed study of the soil temperature distribution indicates otherwise. Rigid extruded polystyrene insulation board may be used to insulate adjacent chilled-water lines from the effects of a buried heat distribution pipe; however, care must be taken not to exceed the temperature limits of the extruded polystyrene insulation; numerical analysis of the thermal problem may be required. Finally, at valve locations or when transitioning from ductile or plastic piping to steel at the buildings, flanged connections are usually best, but should be located inside the building or a small access port and not directly buried. Experience shows that buried flanges, or flanged connections such as to valves, are prone to leakage, and should be avoided. Proper gasket selection and bolt torque are also critical to a leakfree system.

Table 12 summarizes some of the important aspects of the various piping materials, and Figure 13 shows approximate relative costs of the most popular materials. The major advantages and disadvantages of each of these materials, as well as applicable standards when used for the carrier pipe, are as follows:

- **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, ASTM A106A/106M.
- **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. and smaller).
 - Standards: ASTM B88.
- **Ductile Iron**
 - Advantages: Reasonable strength and flexibility, available in sizes from 4 to about 64 in., familiar material to many workforces. Faster installation.
 - Disadvantages: Heavy, susceptible to corrosion and requires corrosion protection, can only be joined by mechanical joints, some mechanical joint designs require thrust blocks at all changes in direction. Fittings are expensive. Allowable leakage per standards.
 - Standards: AWWA C151.
- **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 are required, difficult to add branch line piping to. Lower pressure and velocity limits. Allowable leakage per standards.
 - Standards: AWWA C300, AWWA C301, AWWA C302, AWWA C303.
- **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, ASTM D2996 e1.
- **PVC**
 - Advantages: Light weight, low cost, available in sizes up to 48 in.
 - 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, AWWA C905, ASTM D1785, ASTM D2241.
- **PE and HDPE**
 - Advantages: Light weight, very flexible, can be fusion welded for high-integrity joints, available in sizes up to 63 in. Leak free and fully restrained (no anchor blocks).
 - Disadvantages: Low strength compared to steel and FRP results in significant wall thickness and thus cost in larger diameters.
 - Increased wall thickness also reduces inside diameter, which results 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 fluctuates with petroleum pricing.
 - Standards: AWWA C901, AWWA C906.

Aboveground Systems

An aboveground system consists of a distribution pipe, insulation that surrounds the pipe, and a protective jacket that surrounds the insulation. In outdoor applications, the protective jacket should be metal with a 3 mil thick polyurethane moisture barrier heat-laminated to the inside surface. Note that this moisture barrier is used to reduce the likelihood of corrosion on the interior surface of the jacket, is not the same as a vapor retarder, and does not replace a vapor retarder on cold systems. When the distribution system carries chilled water or other cold media, a vapor retarder is required for all types of insulation except cellular glass. In an ASHRAE test by Chyu et al. (1998a), cellular glass absorbed essentially zero water in a chilled-water application. In heating applications, the vapor retarder is not needed nor recommended; however, a reasonably watertight jacket is required to keep stormwater out of the insulation. The jacket material can be aluminum, stainless steel, galvanized steel, other metals, plastic sheet, a multilayered fabric and organic cement composite, or a combination of these. In outdoor applications, only metal or ultraviolet-resistant fabric and organic cement composites should be used.

Table 12. Relative Merits of Piping Materials Commonly Used for District Cooling Distribution Systems

Piping System	Carrier Pipe Joint Integrity	Joint Inspection	Insulated Joints Possible ²	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 <i>D</i> = 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 <i>D</i> = Low	Low	
Large <i>D</i> = Medium	Excellent	Small <i>D</i> = Low						
Large <i>D</i> = Medium/high								

² Insulated joints are not recommended for piping systems that have allowable leakage rates for joints.

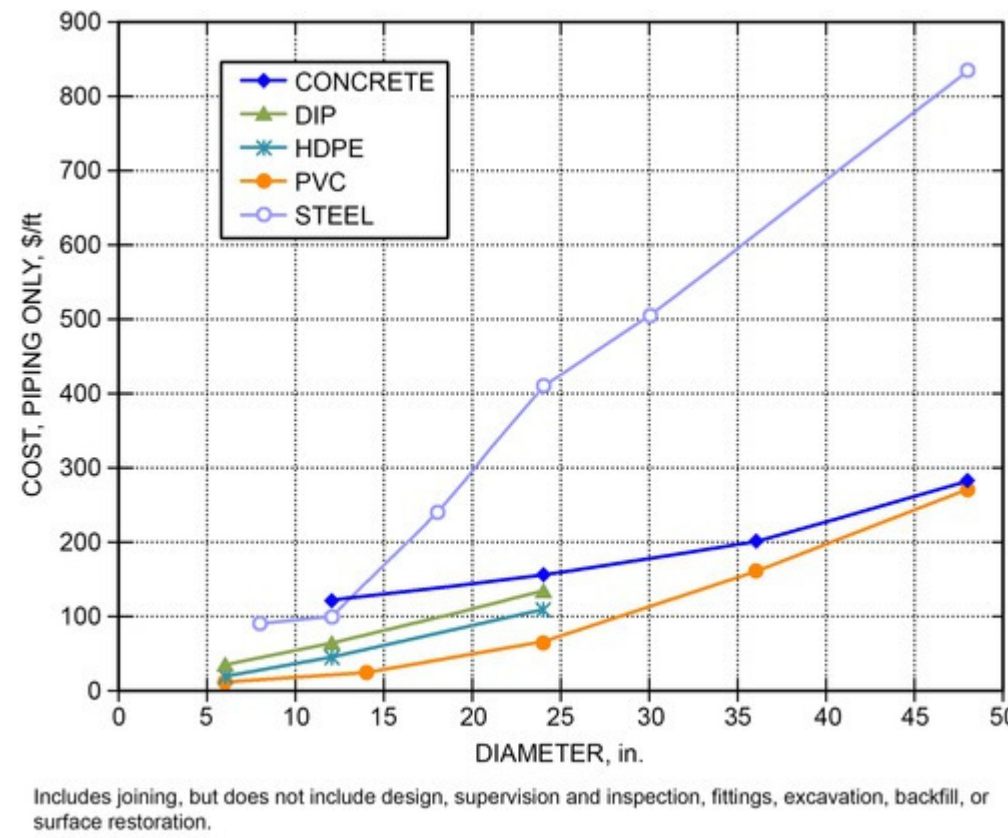


Figure 13. Relative Costs for Piping Alone, Uninsulated Cost data are from RSMeans-CostWorks® (www.rsmeansonline.com) for third-quarter 2012 water utilities.

Structural columns and supports are typically made of wood, steel, or concrete. A crossbar is often placed across the top of the column when more than one distribution pipe is supported from one column. Sidewalk and road crossings require an elaborate support structure to elevate the distribution piping above traffic. Pipe expansion and contraction is taken up in loops, elbows, and bends. Manufactured expansion joints may be used, but they are usually not recommended because of a shorter life or a higher frequency of required maintenance than the remainder of the system. Supports that attach the distribution pipes to the support columns are commercially available as described in MSS *Standards* SP-58 and SP-69. The distribution pipes should have welded joints.

An aboveground system has the lowest first cost and is the easiest to inspect and maintain; therefore, it has the lowest life-cycle cost. It is the standard against which all other systems are compared. Its major drawbacks are its poor aesthetics, its safety hazard if struck by vehicles and equipment, and its susceptibility to freezing in cold climates if circulation is stopped or if heat is not added to the working medium. These drawbacks often remove this system from contention as a viable alternative.

Underground Systems

An underground system solves the problems of aesthetics and exposure to vehicles of the aboveground system; however, burying a system causes other problems with materials, design, construction, and maintenance that have historically been difficult to solve. An underground heat distribution system is not a typical utility like gas, domestic water, and sanitary systems, and requires an order of magnitude more design effort and construction inspection accuracy when compared to other utility distribution projects. The thermal effects and difficulty of keeping the insulation dry make it much more difficult to design and construct when compared to systems operating near the ambient temperature. Underground systems cost almost 10 times as much to build, and require much more to operate and maintain. Heat distribution systems must be designed for zero leakage and must account for thermal expansion, degradation of material as a function of temperature, high-pressure and transient shock waves, heat loss restrictions, and accelerated corrosion. In the past, resolving one problem in underground systems often created a new, more serious problem that was not recognized until premature failure occurred. Segan and Chen (1984) describe the types of premature failures that may occur if this guidance is ignored.

Common types of underground systems are the walk-through tunnel, concrete surface trench, deep-buried small tunnel, poured insulation envelope, rigid closed-cell insulation, and conduit system.

Walk-Through Tunnel. This system (Figure 14) consists of a field-erected tunnel large enough for a person to walk through after the distribution pipes are in place. It is essentially an aboveground system enclosed with a tunnel. The tunnel is buried deep enough to cover the top with earth, and is large enough for routine maintenance and inspection to be done easily without excavation. The preferred construction material for the tunnel walls and top cover is reinforced concrete. Masonry units and metal preformed sections have been used to construct the tunnel and top with less success, because of groundwater leakage and metal corrosion. The distribution pipes are supported from the tunnel wall or floor with pipe supports that are commonly used on aboveground systems or in buildings. Some groundwater will penetrate the top and walls of the tunnel; therefore, a water drainage system must be provided. Usually, electric lights and electric service outlets are provided for ease of inspection and maintenance. This system has the highest first cost of all underground systems; however, it can have the lowest life-cycle cost because of its ease of maintenance, the ability to correct construction errors easily, and an extremely long life. If steam or HTHW piping is located in the tunnel, ambient temperatures may become extreme; such tunnels are typically either forced ventilated or gravity ventilated. Additional insulation may be required for chilled-water piping that shares the tunnel with steam or HTHW piping, because of the higher ambient temperatures. Selected pipe insulation material should be coupled with a high-performance vapor retarder and possibly a protective jacket.

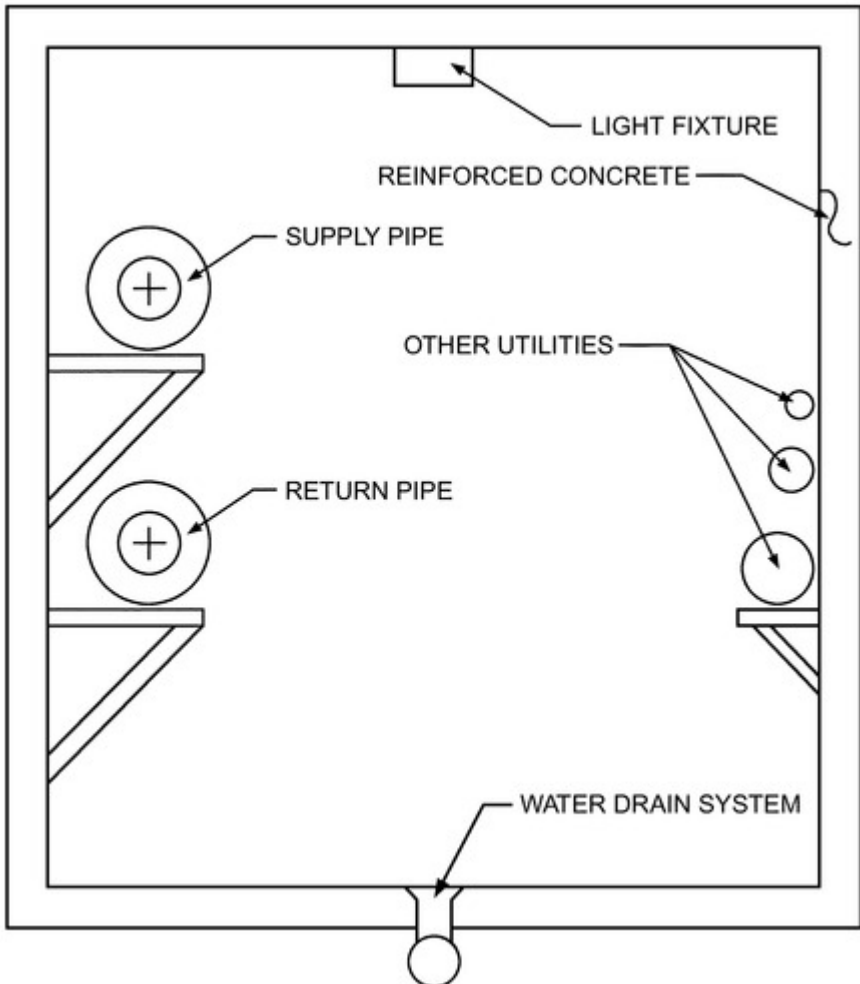


Figure 14. Walk-Through Tunnel

Shallow Concrete Surface Trench. This system (Figure 15) is a partially buried system. The floor is usually about 3 ft below the surface grade. It is only wide enough for the carrier pipes and the pipe insulation plus some additional width to allow for pipe movement and possibly enough room for a person to stand on the floor. The trench usually is about as wide as it is deep. The top is constructed of reinforced concrete covers that protrude slightly above the surface and may also serve as a sidewalk. The floor and walls are usually cast-in-place reinforced concrete and the top is either precast or cast-in-place concrete. Precast concrete floor and wall sections have not been successful because of the large number of oblique joints and nonstandard sections required to follow the surface topography and to slope the floor for drainage. This system is designed to handle stormwater and groundwater that enters the system, so the floor is always sloped toward a drainage point, which typically is a steam trap pit, valve vault, or access port where a sump pump or other positive-drainage method is provided.

Cross beams that attach to the side walls are preferred to support the carrier pipes. This keeps the floor free of obstacles that would interfere with drainage and allows the distribution pipes to be assembled before lowering them on the pipe supports. Also, floor-mounted pipe supports tend to corrode.

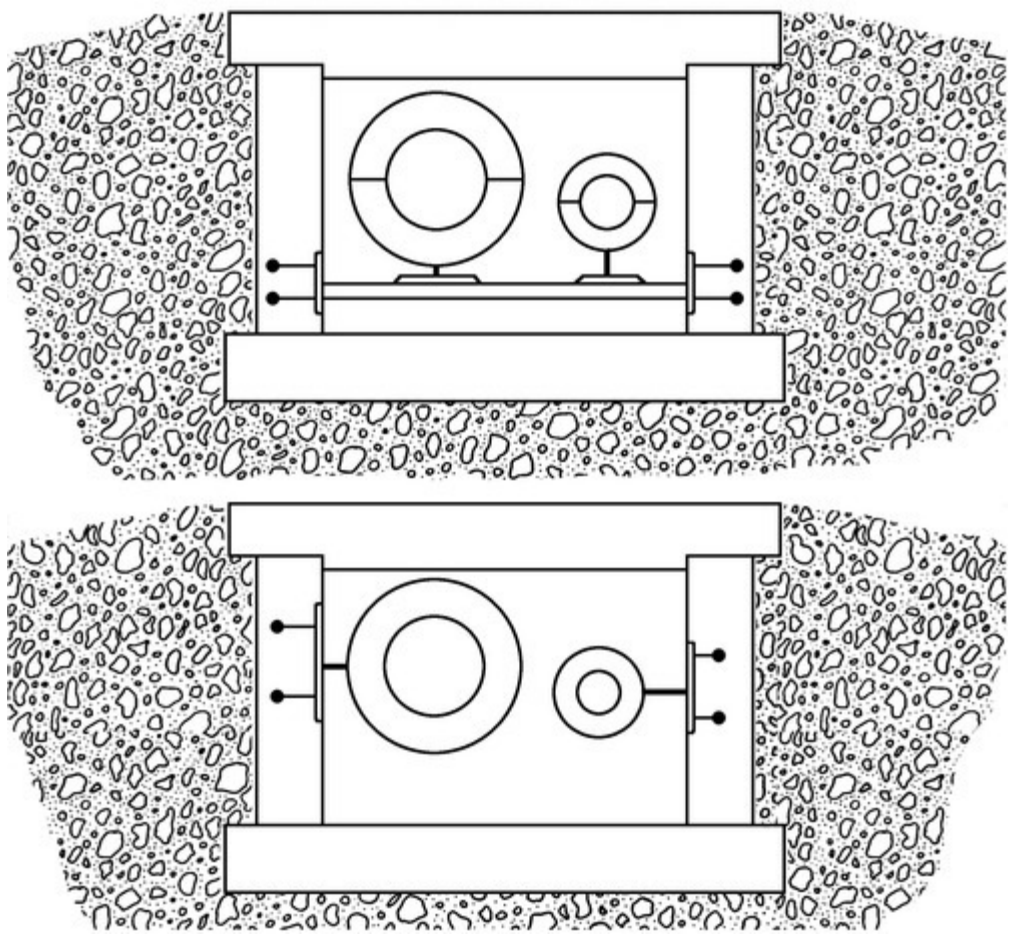


Figure 15. Concrete Surface Trench

The carrier pipes, pipe supports, expansion loops and bends, and insulation jacket are similar to aboveground systems, with the exception of pipe insulation. Experience with these systems indicates that flooding will occur several times during their design life; therefore, the insulation must be able to survive flooding and boiling and then return to near its original thermal efficiency. The pipe insulation is covered with a metal or plastic jacket to protect the insulation from abuse and from stormwater that enters at the top cover butt joints. Small inspection ports of about 12 in. diameter may be cast into the top covers at key locations so the system can be inspected without removing the top covers. All replaceable elements, such as valves, condensate pumps, steam traps, strainers, sump pumps, and meters, are located in valve vaults. The first cost of this system is among the lowest for underground systems because it uses typical construction techniques and materials. The life-cycle cost is often the lowest because it is easy to maintain, correct construction deficiencies, and repair leaks.

Deep-Bury Tunnel. The tunnel in this system (Figure 16) is only large enough to contain the distribution piping, pipe insulation, and pipe supports. One type of deep-bury tunnel is the shallow concrete surface trench covered with earth and sloped independent of the topography. Because the system is covered with earth, it is essentially not maintainable between valve vaults without major excavation. All details of this system must be designed and all materials must be specified by the project design engineer.

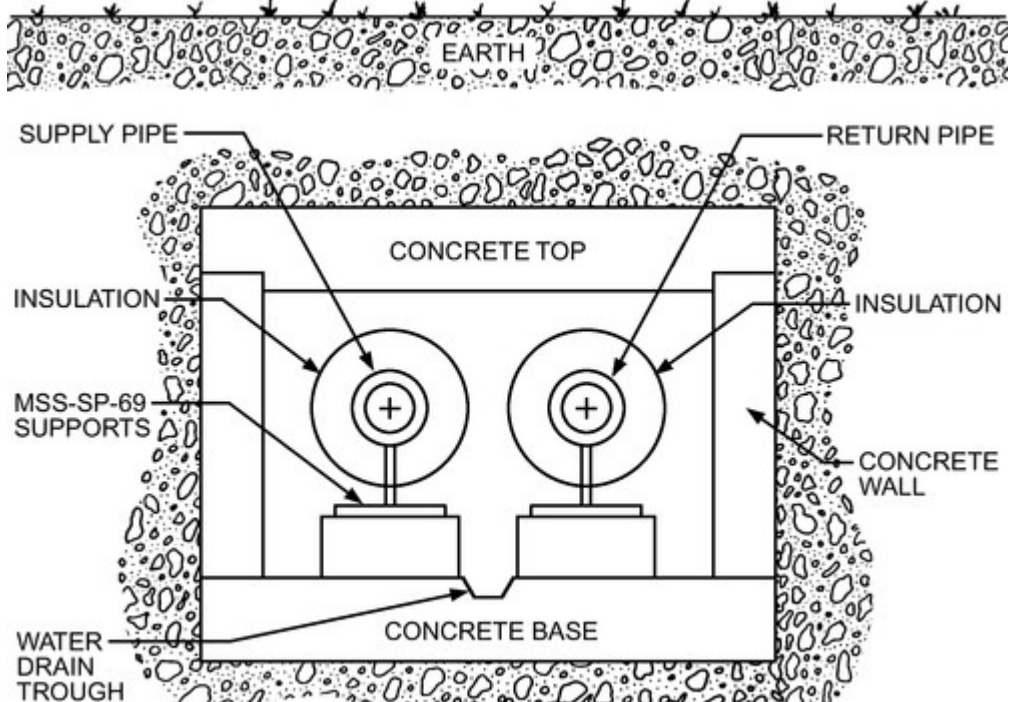


Figure 16. Deep-Bury Small Tunnel (Boxed Conduit)

Because this system is not maintainable between valve vaults, great care must be taken to select materials that will last for the intended life and to ensure that the groundwater drainage system will function reliably. This system is intended to be used on sites where the groundwater elevation is typically lower than the bottom of the tunnel. The system can tolerate some groundwater saturation, depending on the watertightness of the construction and the capacity and reliability of the internal drainage. But even in desert areas, storms occur that expose underground systems to flooding; therefore, as with other types of underground systems, it must be designed to handle groundwater or stormwater that enters the system. The distribution pipe insulation must be of the type that can withstand flooding and boiling and still retain its thermal efficiency.

Construction of this system is typically started in an excavated trench by pouring a cast-in-place concrete base that is sloped so intruding groundwater can drain to the valve vaults. The slope selected must also be compatible with the pipe slope requirements of the distribution system. The concrete base may have provisions for the supports for the distribution pipes, the groundwater drainage system, and the mating surface for the side walls. The side walls may have provisions for the pipe supports if the pipes are not bottom supported. If the upper portion is to have cast-in-place concrete walls, the bottom may have reinforcing steel for the walls protruding upward. The pipe supports, distribution pipes, and pipe insulation are all installed before the top cover is installed.

The groundwater drainage system may be a trough formed into the concrete bottom, a sanitary drainage pipe cast into the concrete bottom, or a sanitary pipe that is located slightly below the concrete base. The cover for the system is typically either of cast-in-place concrete or preformed sections such as precast concrete sections or half-round clay tile sections. The top covers must mate to the bottom and each other as tightly as possible to limit entry of groundwater. After the covers are installed, the system is covered with earth to match the existing topography.

Poured Insulation. This system (Figure 17) is buried with the distribution system pipes encased in an envelope of insulating material and the insulation envelope covered with a thick layer of earth as required to match existing topography. This system is used on sites where the groundwater is typically well below the piping system. Like other underground systems, experience indicates that it will be flooded because the soil will become saturated with water several times during the design life; therefore, the design must accommodate flooding.

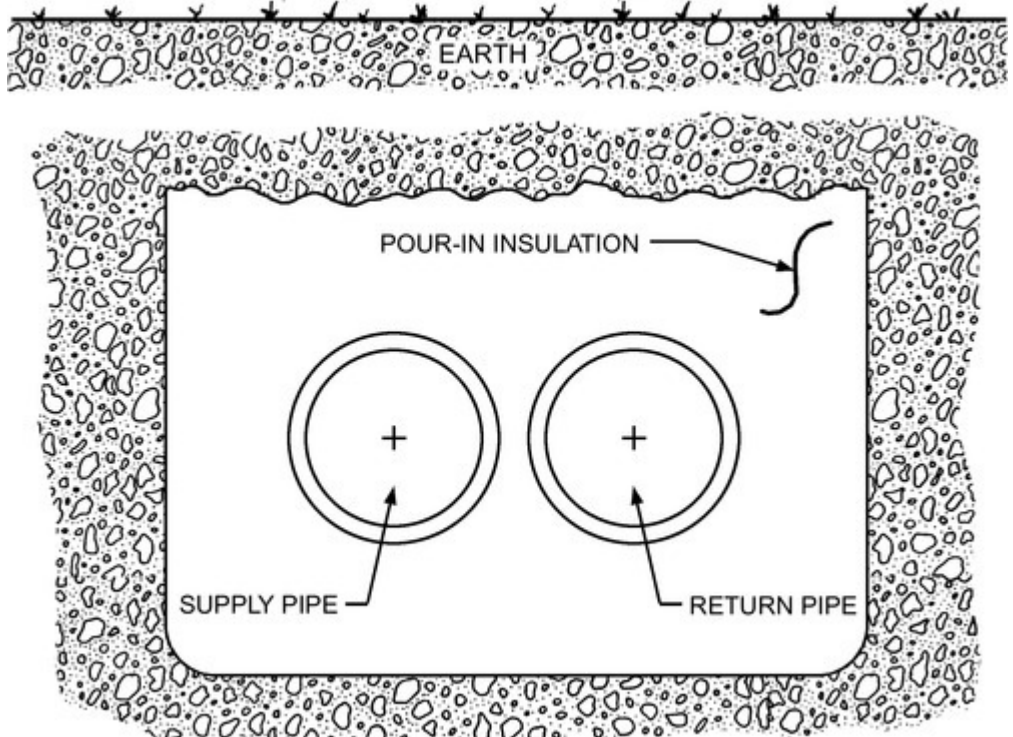


Figure 17. Poured Insulation System

The insulation material serves several functions. It may support the distribution pipes as bedding and backfill, with additional support as recommended by the product manufacturer, and it must support earth loads. The insulation must prevent groundwater from entering the interior of the envelope, and it must have long-term resistance to physical breakdown caused by heat and water. The insulation envelope must allow the distribution pipes to expand and contract axially as the pipes change temperature. In elbows, expansion loops, and bends, the insulation must allow formed cavities for lateral movement of the pipes, or be able to migrate around the pipe without significant distortion of the insulation envelope while still retaining the required structural load-carrying capacity. Pay special attention to corrosion of metal parts and water penetration at anchors and structural supports that penetrate the insulation envelope.

Hot distribution pipes tend to drive moisture out of the insulation as steam; however, pipes used to distribute a cooling medium tend to condense water in the insulation, which reduces the insulation's thermal resistance. A groundwater drainage system may be required, depending on the insulation material selected and severity of the groundwater; however, if such drainage is needed, it is a strong indicator that this is not the proper system for the site conditions.

This system is constructed by excavating a trench with a bottom slope that matches the desired slope of the distribution piping. The width of the bottom of the trench is usually the same as the width of the insulation envelope because it serves as a form. The distribution piping is then assembled in the trench and supported at the anchors and by blocks that are removed as the insulation is poured in place. The form for the insulation can be the trench bottom and sides, wooden forms, or sheets of plastic, depending on the type of insulation used and the site conditions. The insulation envelope is covered with earth to complete the installation.

The project design engineer is responsible for finding an insulation material that fulfills all of the previously mentioned requirements. At present, no standards have been developed for insulation used in this type of application. **Hydrophobic powders**, which are a special type of pulverized rock treated to be water repellent, have been used successfully. The hydrophobic characteristic of this powder prevents water from dampening the powder and has some ability as a barrier for preventing water from entering the insulation envelope. This insulating powder typically has a much higher thermal conductivity than mineral wool or fiberglass pipe insulation; therefore, the thickness of the poured envelope must be significantly greater. In addition, Phetteplace et al. (1998) found installed heat losses to be much higher than would be expected using manufacturer's data for one poured insulation material, and that actual installed dimensions were less than manufacturer's recommendations in the majority of installations. This may have been the result of the design or construction errors, contractors purposely "shorting" on dimensions, or compaction during or after backfilling. The user is cautioned to verify dimensions during construction and also use appropriate in-place densities for the poured insulation material; do not use bulk or loose density. Measure installed density by a meaningful test for noncohesive soils (e.g., ASTM Standard D4253). Furthermore, thermal conductivity is a strong function of density and thus must also be measured at realistic in-place densities. As with any system, the designer should analyze the cost of the alternatives, including preinsulated (i.e., conduit) or field-insulated piping.

Field-Installed, Direct-Buried Rigid Closed-Cell Insulation. In this system (Figure 18), a rigid closed-cell insulation is covered with an asphaltic jacket. The insulation supports the pipe. Oversized loops with internal support elements provide for expansion. The project design engineer must make provisions for pipe movement in the expansion loops. As shown in Figure 18, a drain should be installed to drain groundwater away. A waterproof jacket is recommended for all buried applications.

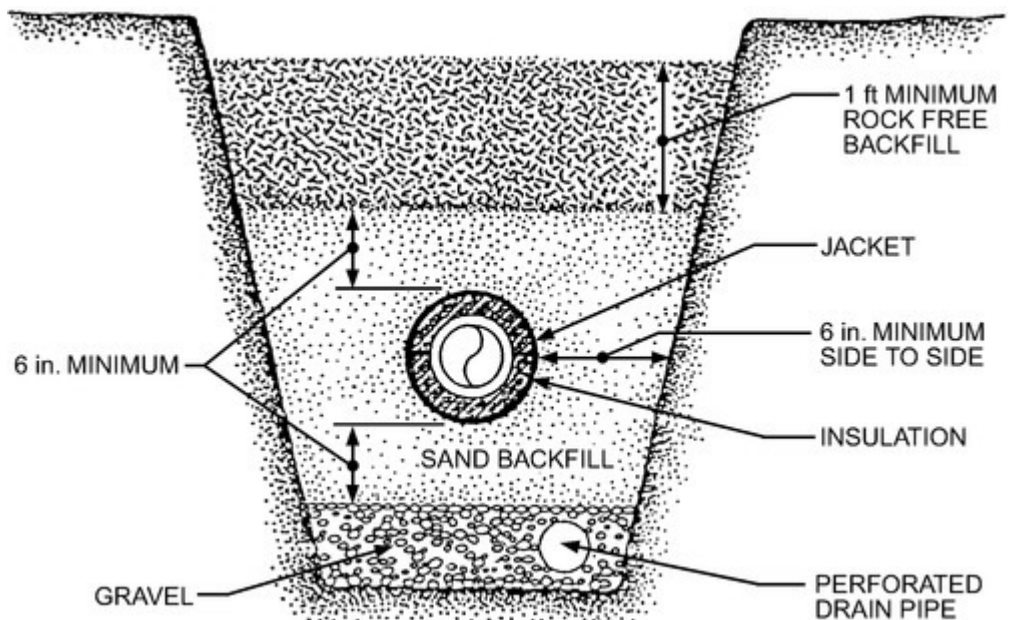


Figure 18. Field-Installed, Direct-Buried Rigid Closed-Cell Insulation System

When used for heating applications, the dry soil condition must be investigated to determine if the temperature of the jacket exceeds the material's allowable temperature (see the section on Methods of Heat Transfer Analysis). This is one of the controlling conditions to determine how high the carrier pipe fluid temperature can be without exceeding the temperature limits of the jacket material. Insulation thickness depends on the thermal operating parameters of the carrier pipe. For maximum system integrity under extreme operating conditions, such as groundwater flooding, the jacketing may be applied to the insulation segments in the fabrication shop with hot asphalt. As with other underground systems, experience indicates that this system may be flooded several times during its projected life.

Conduits

The term conduit denotes an entire assembly, which consists of a carrier pipe, the pipe insulation, the casing, and the exterior casing coating (Figure 19). The conduit is assembled in a factory and shipped as unit called a **conduit section**. The pipe that carries the working medium is called a **carrier pipe** and the outermost perimeter enclosure is called a **casing**. Each conduit section is shipped in lengths up to 40 ft. Elbows, tees, loops, and bends are factory prefabricated to match the straight sections. The prefabricated components are assembled at the construction site; therefore, a construction contract is typically required for trenching, backfilling, connecting to buildings, connecting to distribution systems, constructing valve vaults, and performing some electrical work associated with sump pumps, power receptacles, and lights.

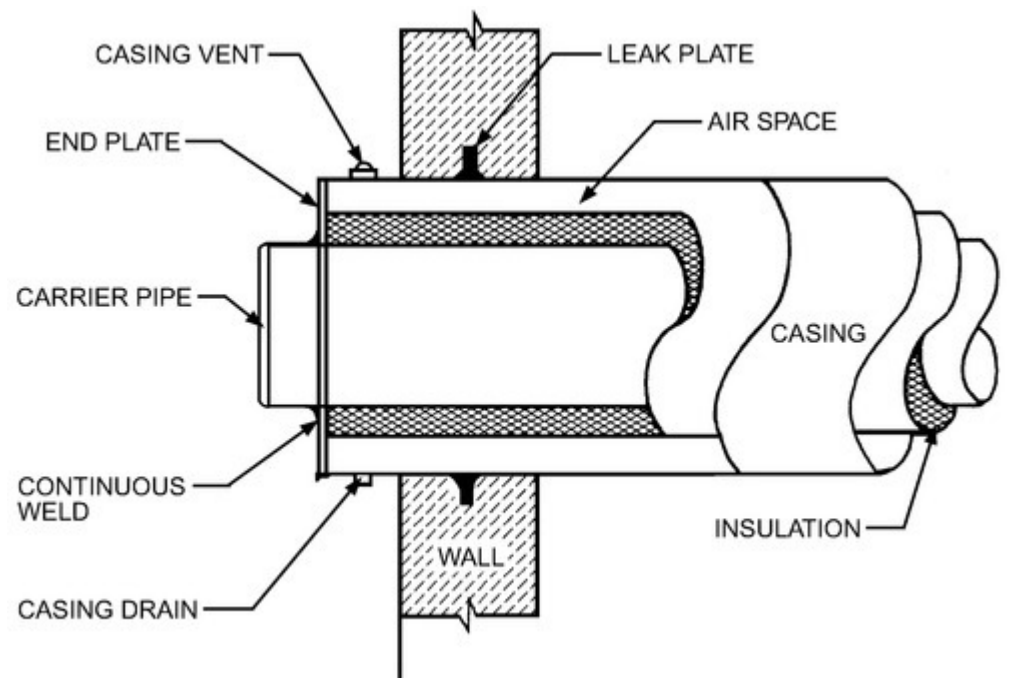


Figure 19. Conduit System Components

Much of the design work is done by the factory that manufactures the prefabricated sections; however, the field work must be designed and specified by the project design engineer or architect. Prefabricated components create a serious problem with accountability. For comparison, when systems are entirely field assembled, the design responsibility clearly belongs to the project design engineer, and system assembly and installation are clearly the responsibility of the construction contractor. When a condition arises where a conduit system cannot be built without modifying prefabricated components, or if the construction contractor does not follow the instructions from the prefabricator, a serious conflict of responsibility arises. The design engineer, as the engineer of record (EOR), assumes the responsibility of review and approval of any design work by others. For these reasons, it is imperative that the project design engineer or architect clearly delineate the responsibilities of the factory prefabricator.

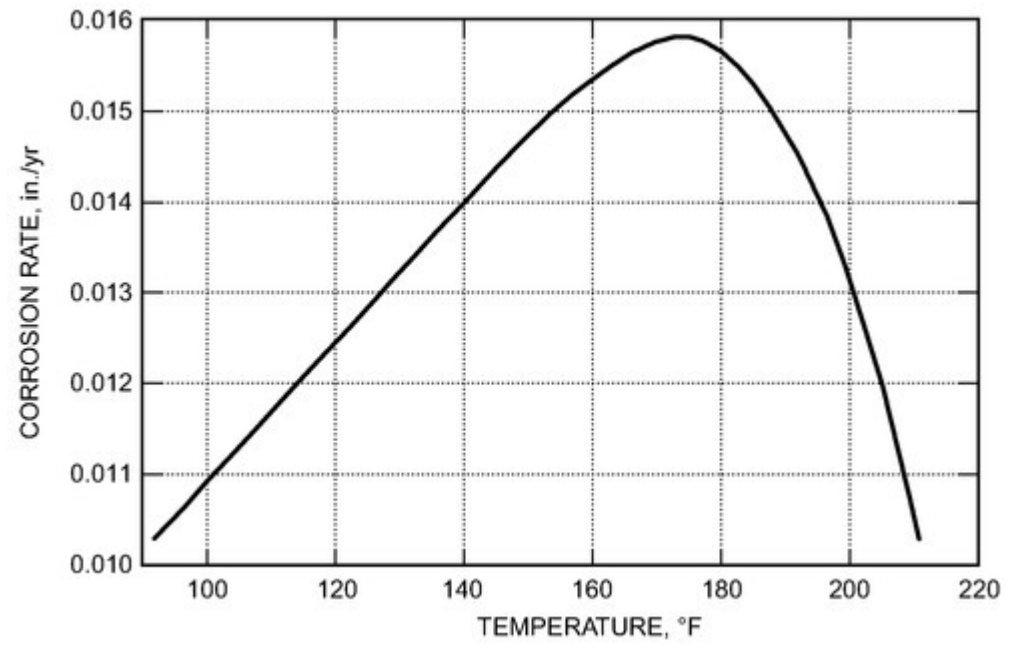


Figure 20. Corrosion Rate in Aggressive Environment Similar to Mild Steel Casings in Soil

Cushing loads have been used (erroneously) to size the casing thickness, assuming that corrosion was not a factor. However, corrosion rate is usually the controlling factor because the casing temperature can range from less than 100°F to more than 300°F, a range that encompasses the maximum corrosion rate of steel (Figure 20). As shown in the figure, the steel casing of a district heating pipe experiences corrosion rates several times that of domestic water pipes. The casing temperature varies with burial depth, soil conditions, carrier pipe temperature, and pipe insulation thickness. The casing must be strong and thick enough to withstand expansion and contraction forces and corrosion degradation.

All insulation must be kept dry for it to maintain its thermal insulating properties; the exception is cellular glass in cold applications. Because underground systems may be flooded several times during their design life, even on sites that are thought to be dry, a reliable water intrusion removal system is necessary in the valve vaults. Two designs are used to ensure that the insulation performs satisfactorily for the life of the system. In the **air space system**, an annular air space between the pipe insulation and the casing allows the insulation to be dried out if water enters. In the **water spread limiting (WSL) system**, which has no air space, the conduit is designed to keep water from entering the insulation. If water enters one section, a WSL system slows or prevents its spread to adjacent sections of piping.

The air space conduit system (Figures 21 and 22) should have an insulation that can survive short-term flooding without damage. The conduit manufacturer usually runs a boiling test with the insulation installed in the typical factory casing. No U.S. standard has been approved for this boiling test; however, the U.S. government uses a Federal Agency Committee 96 h boiling test for conduit insulation [see Phetteplace et al. (2013b) for protocol]. The insulation must have demonstrated that it can be dried with air flowing through an annular air space, and it must retain nearly new thermal insulating properties when dried.

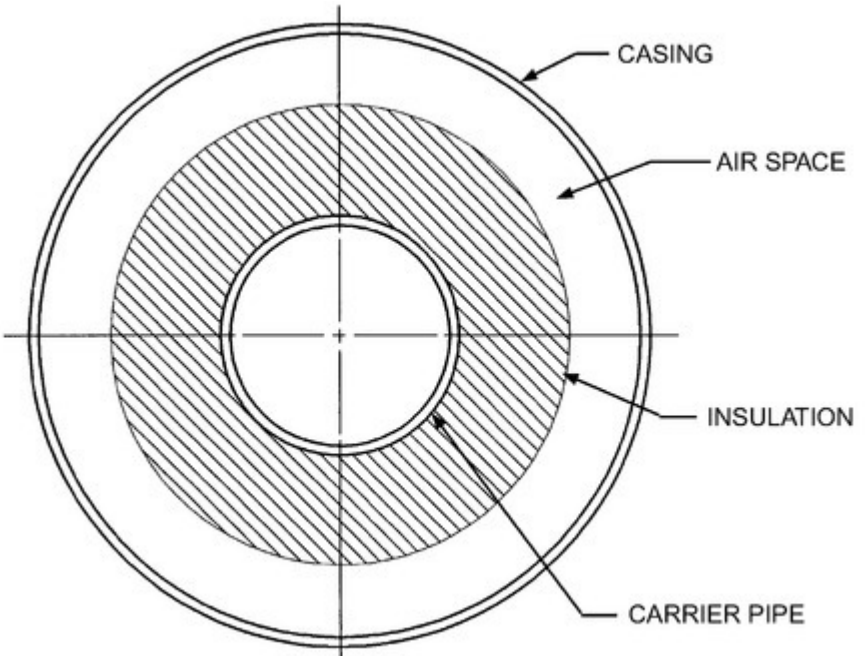


Figure 21. Conduit System with Annular Air Space and Single Carrier Pipe

Insulation fails because the bonding agents, called binders, that hold the principal insulation material in the desired shape degrade. The annular air space around the insulation, typically more than 1 in. wide, allows air to flow outside the insulation to dry it. Unfortunately, the air space has a serious detrimental side effect: it allows unwanted water to flow freely to other parts of the system.

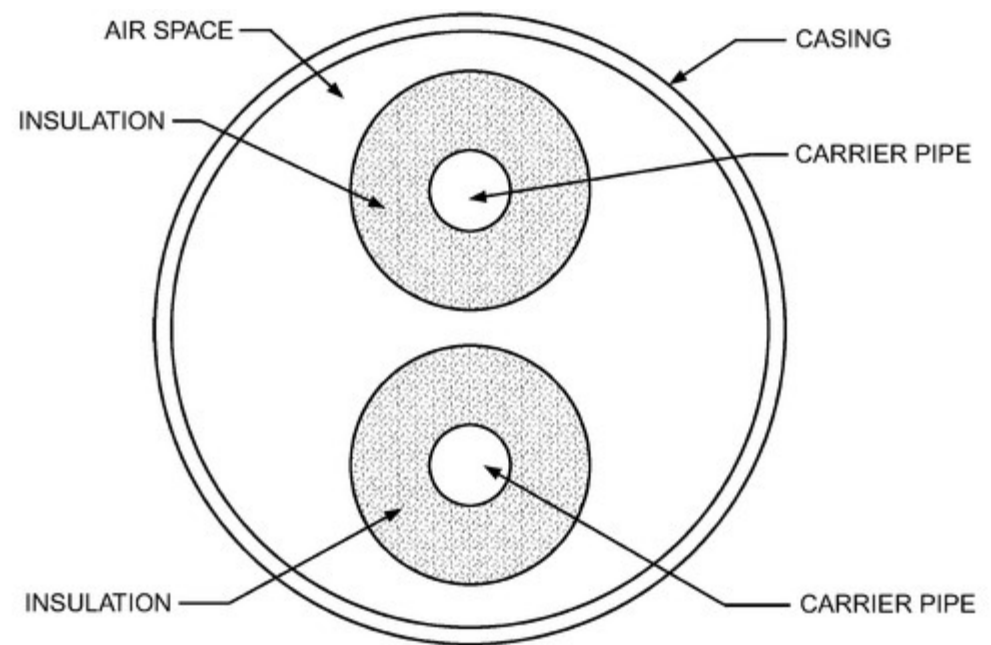


Figure 22. Conduit System with Two Carrier Pipes and Annular Air Space

The WSL system (Figure 23) encloses the insulation in an envelope that will not allow water to contact the insulation. The typical insulation is polyurethane foam, which will be ruined if excess water infiltration occurs. Polyurethane foam is limited to a temperature of about 250°F for a service life of 30 years or more. Europe has the most successful of the WSL systems, which are typically used in low-temperature water applications. These systems, available in the United States as well, meet CEN European Standard EN 353 with regard to all major construction details. Standards also have been established for fittings (EN 448), preinsulated valves (EN 488), field joint assemblies (EN 489), and system design (EN 13941). With this system, the carrier pipe, insulation, and casing are bonded together to form a single unit. Forces caused by thermal expansion are passed as shear forces to the mating component and ultimately to the soil; thus, no additional expansion provisions are required. Movement at changes in direction (elbows) may be accommodated for by cushions.

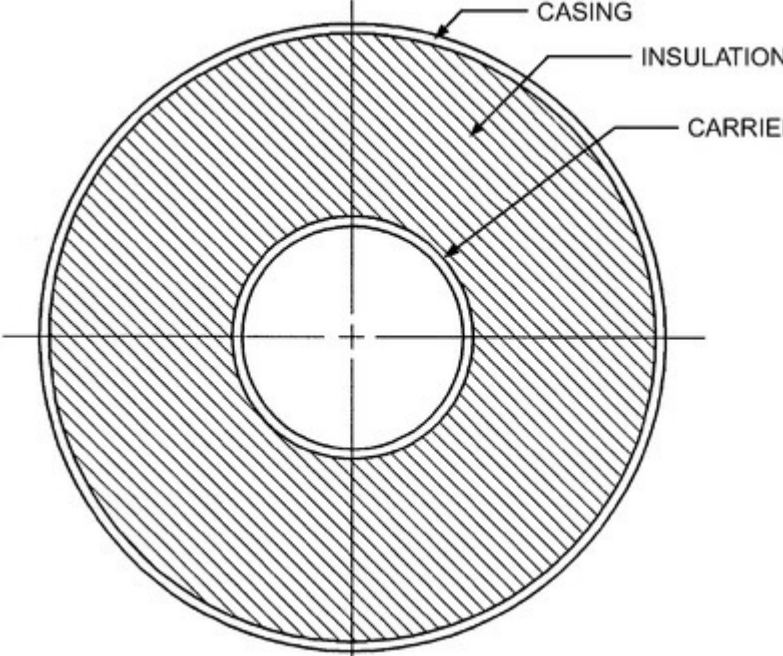


Figure 23. Conduit System with Single Carrier Pipe and No Air Space (WSL)

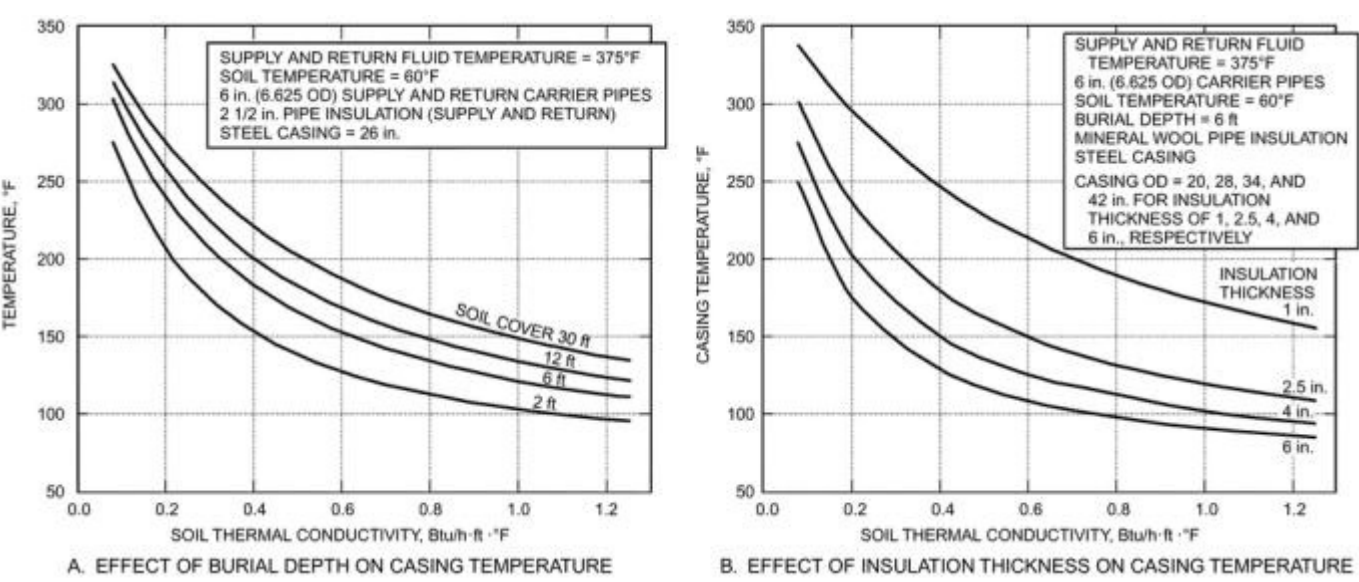
This type of WSL system is feasible because of the small temperature differential and because the thinner carrier pipe wall creates smaller expansion forces because of its low cross-sectional area. Some systems rely on a watertight field joint between joining casing sections to extend the envelope to a distant envelope termination point where the casing is sealed to the carrier pipe. Other systems form the waterproof insulation envelope in each individual prefabricated conduit section, using the casing and carrier pipe to form part of the envelope and a waterproof bulkhead to seal the casing to the carrier pipe. In another type of construction, a second pipe fits tightly over the carrier pipe and seals the insulation between the second pipe and casing to achieve a watertight insulation envelope.

Conduit Design Conditions. The following three design conditions must be addressed to have reasonable assurance that the system selected will have a satisfactory service life:

- **Maximum heat loss** occurs when the soil is wettest and the conduit is shallowly buried (minimum burial depth), usually with about 2 ft of earth cover. This condition represents the highest gross heat transfer and is used to size the distribution piping and equipment in the central heating plant. For heating piping, because the casing is coldest during start-up, the relative movement of the carrier pipe with respect to the casing may be maximum during this condition.
 - A **dry-soil condition** may occur when the conduit is buried deep. The soil plays a more significant role in the heat transfer than the pipe insulation because of the soil's thickness (and thus, insulating value). The highest temperature of the insulation, casing, and casing coating occur during this condition. Paradoxically, the minimum heat loss occurs during this condition because the soil acts as a good insulator. This condition is used to select temperature-sensitive materials and to design for casing expansion. The relative movement of the carrier pipe with respect to the casing may be at minimum during this condition; however, if the casing is not restrained, its movement with respect to the soil will be at maximum. If restrained, the casing axial stresses and axial forces will be highest and the casing allowable stresses will be lowest because of the high casing temperature. Buckling of the casing is possible in extreme situations.
 - **Figure 24a** shows the effect of burial depth on casing temperature as a function of soil thermal conductivity for a typical system. **Figure 24b** shows the effect of insulation thickness on casing temperature, again as a function of soil thermal conductivity. Analysis of **Figure 24a** and **24b** suggests some design solutions that could lower the effects of the dry-soil condition: reducing carrier pipe temperature, using thicker carrier pipe insulation, providing a device to keep the soil wet, or minimizing burial depth. However, if these solutions are not feasible or cost effective, a different type of material or an alternative system should be considered.
- Although it is possible that the soil will never dry out, given the variability of climate in most areas, it is likely that a drought will occur during the life of the system. Only one very dry condition can cause permanent damage to the insulation or other system components if the soil thermal conductivity drops below the assumed design value. On-site measurement of the driest soil condition likely to cause insulation damage is not feasible. As a result, the designer is left with the conservative choice of using the lowest thermal conductivity from **Table 9** to calculate the highest temperature to be used to select materials.
- The **nominal or average condition** occurs when the soil is at its average water content and the conduit is buried at the average depth. This condition is used to compute the yearly energy consumption from heat loss to the soil (see **Table 9** for soil thermal conductivity and the previous section on Thermal Properties).

Cathodic Protection of Direct-Buried Conduits

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 because of differences in their natural electrical potentials, a current flows from anode to cathode. When 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 makes one section 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.



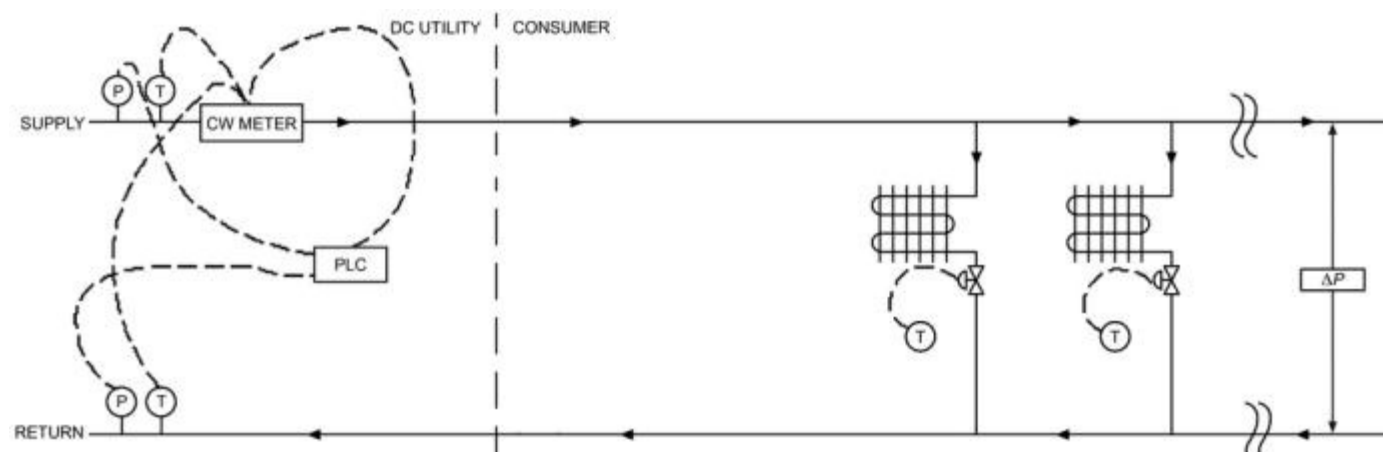


Figure 26. Direct Connection of Building System to District Chilled Water Without Building Pumps (Phetteplace et al. 2013a)

Particular attention must be paid to connecting high-rise buildings because they induce a static head that affects the design pressure of the entire system. Pressure control devices should be investigated carefully. It is not unusual to have a water-based district heating or cooling system with a mixture of direct and indirect connections in which heat exchangers isolate the systems hydraulically based on the location of the ETS in the building and the number of floors served. In a direct connected 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 should be installed in the buildings. Special attention should be given to potential noise problems at the control valves. These valves must correspond to the design pressure differential in a system that has constantly varying distribution pressures because of load shifts. Multiple valves may be required to serve the load under all flow and pressure ranges. Industrial-quality valves and actuators should be used for this application.

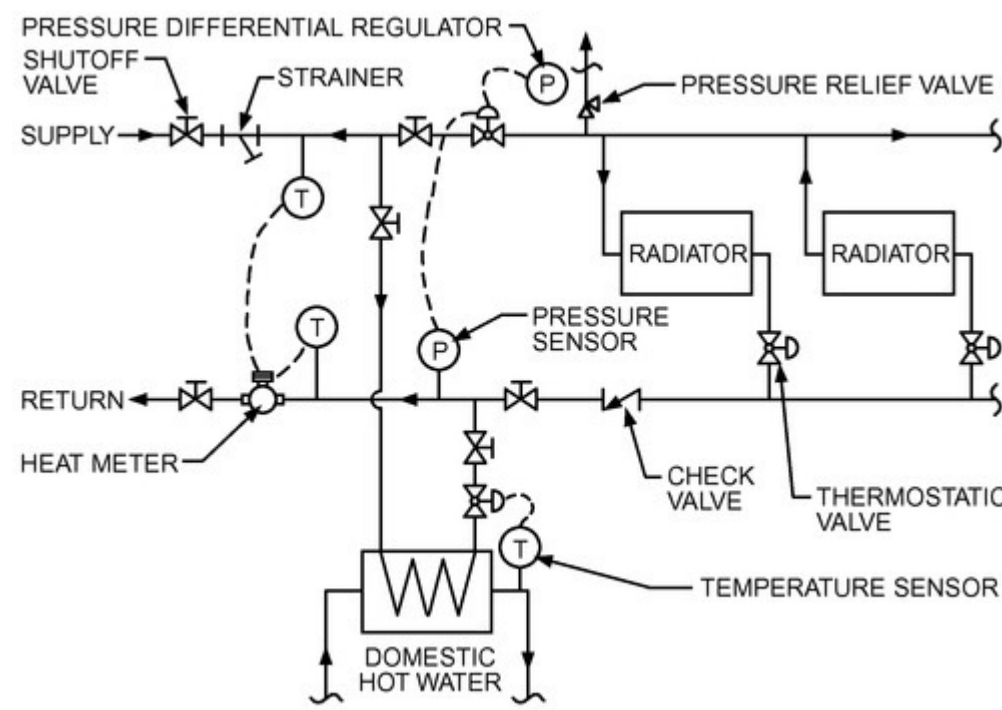


Figure 27. Direct Connection of Building System to District Hot Water

If the temperature supplied from 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. The consumer's desired supply temperature can be attained by mixing return water with 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 (e.g., 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 bridge between return and supply piping, decouplers, and bypass lines). This ensures that terminal unit two-way control valves are subjected only to the differential pressure established by the customer's building (tertiary) pump. Figure 25 shows a primary/secondary connection using an in-building pumping scheme. When tertiary pumps are used, all series connections between the district system pumps must be removed. A series connection can 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. For similar reasons, booster pumps in chilled-water return lines back to the plant are discouraged.

4.2 INDIRECT CONNECTIONS

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 buffer between potentially different quality water treatments. Identical to the direct connection, the rate of energy extraction in the heat exchanger is governed by a control valve that reads 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. 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. The primary disadvantages of using an indirect connection are the (1) additional cost of the heat exchanger, (2) temperature loss caused by the approach of the heat exchanger, and (3) increased pumping pressure because of the addition of another heat transfer surface. Figure 28 shows a typical district chilled-water indirect connection, and Figure 29 shows a cascading district heating indirect connection where the domestic hot-water heat exchanger (HX) is supplied from the comfort HX outlet piping for greater Δt across the connection.

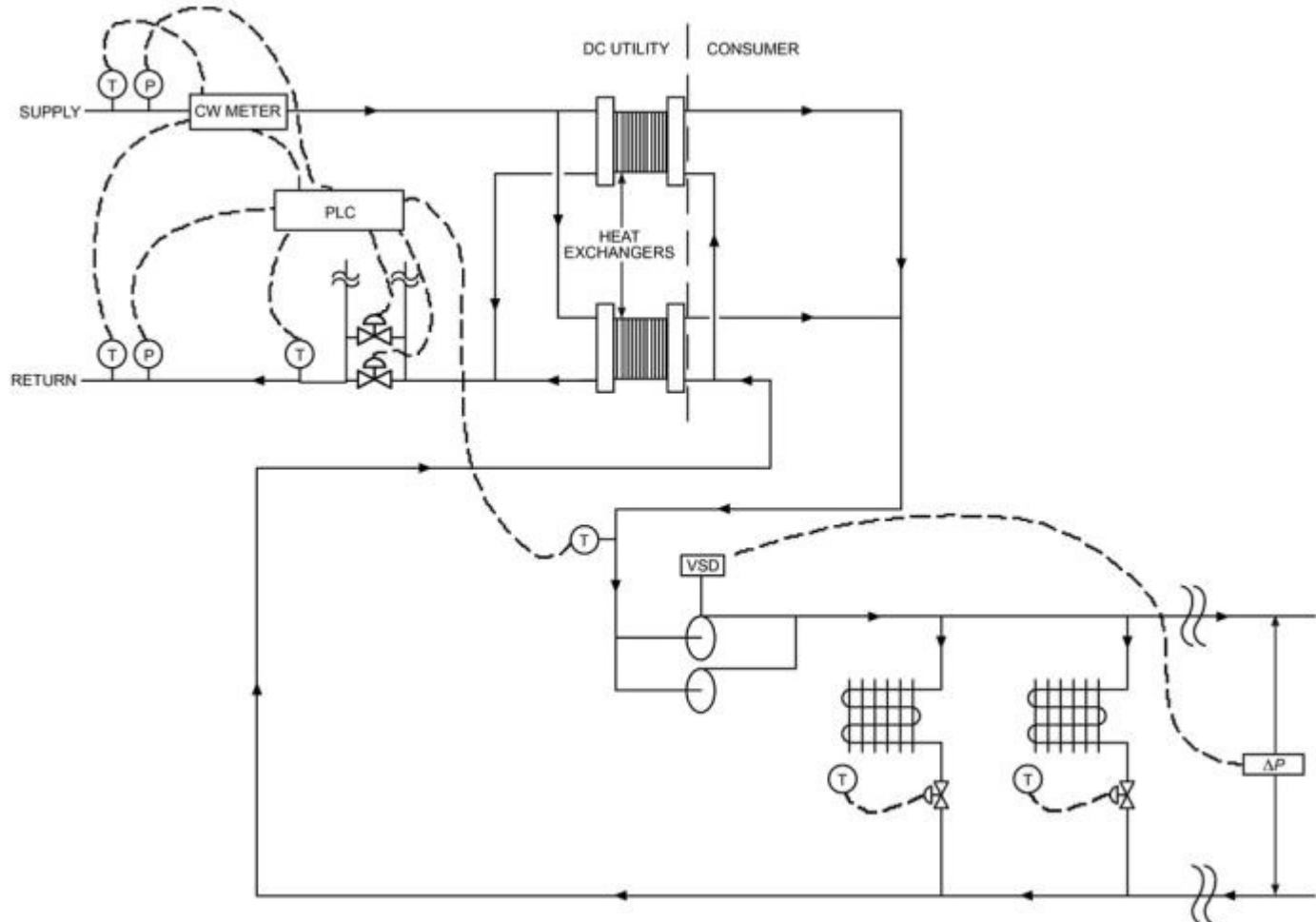


Figure 28. Indirect Connection of Building System to District Chilled Water (Phetteplace et al. 2013a)

4.3 STEAM CONNECTIONS

Although higher pressures and temperatures are sometimes used, most district heating systems supply saturated steam at pressures between 5 and 250 psig to customers' facilities. The steam is pretreated to maintain a neutral pH, and the condensate is cooled and discharged to the building sewage system (not preferred) or returned back to the central plant for recycling (preferred). Many consumers run the condensate through a heat exchanger to heat the domestic hot-water supply of the building before returning it to the central plant or, sometimes, to the building drains if the district system does not have a condensate return pipe. This energy-saving process extracts the maximum amount of energy out of the delivered steam. Again, it is best to return the condensate back to the plant whenever the system configuration dictates, but there still are some vintage single-pipe steam systems in operation that are 100% makeup water systems. The designer should coordinate with the district energy provider to confirm the requirements.

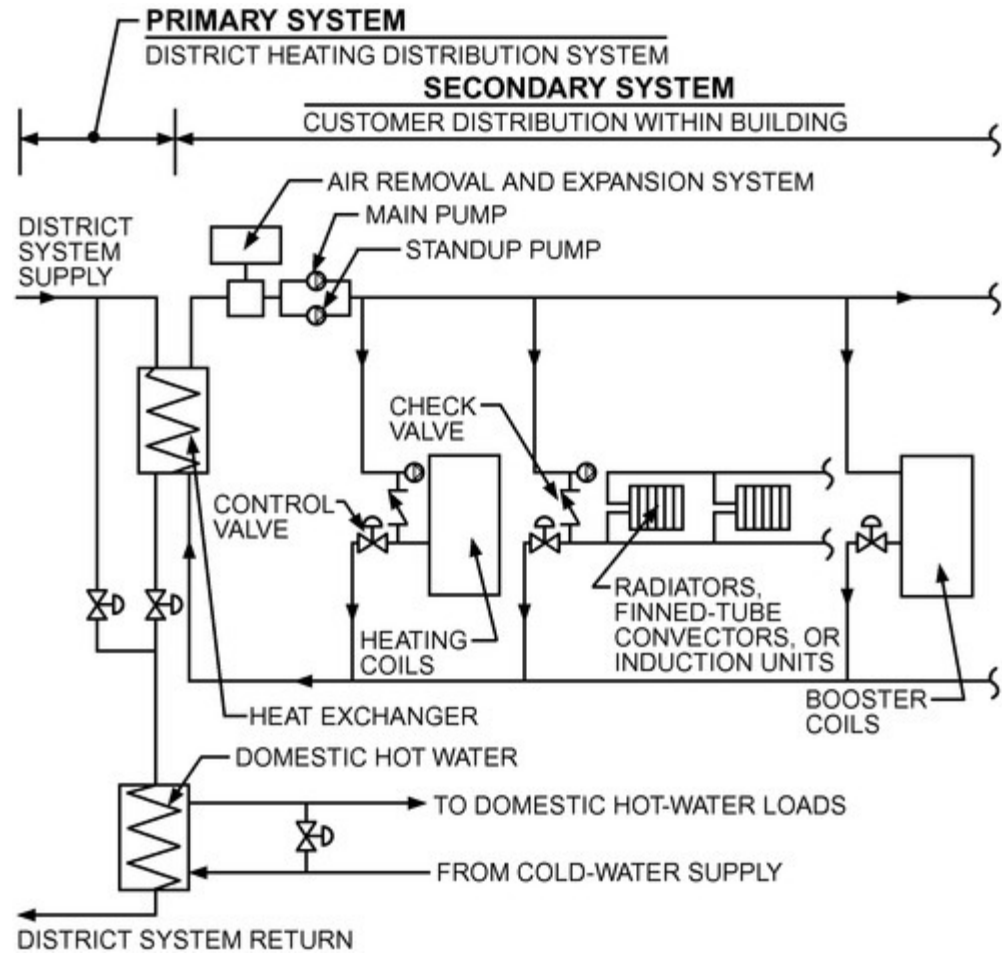


Figure 29. Basic Cascading Indirect Heating-System Schematic

Interconnection between the district and the building is simple when the building uses the steam directly in heating coils or radiators or for process loads (humidification, kitchen, laundry, laboratory, steam absorption chillers, or turbine-driven devices). Other buildings extract the energy from the district steam via a steam-to-water heat exchanger to generate hot water and circulate it to the air-side terminal units. Typical installations are shown in Figures 30 and 31. Chapter 11 has additional information on building distribution piping, valving, traps, and other system requirements. The type of steam chemical treatment should be considered in applications for the food industry and for humidification (Tredinnick 2008b).

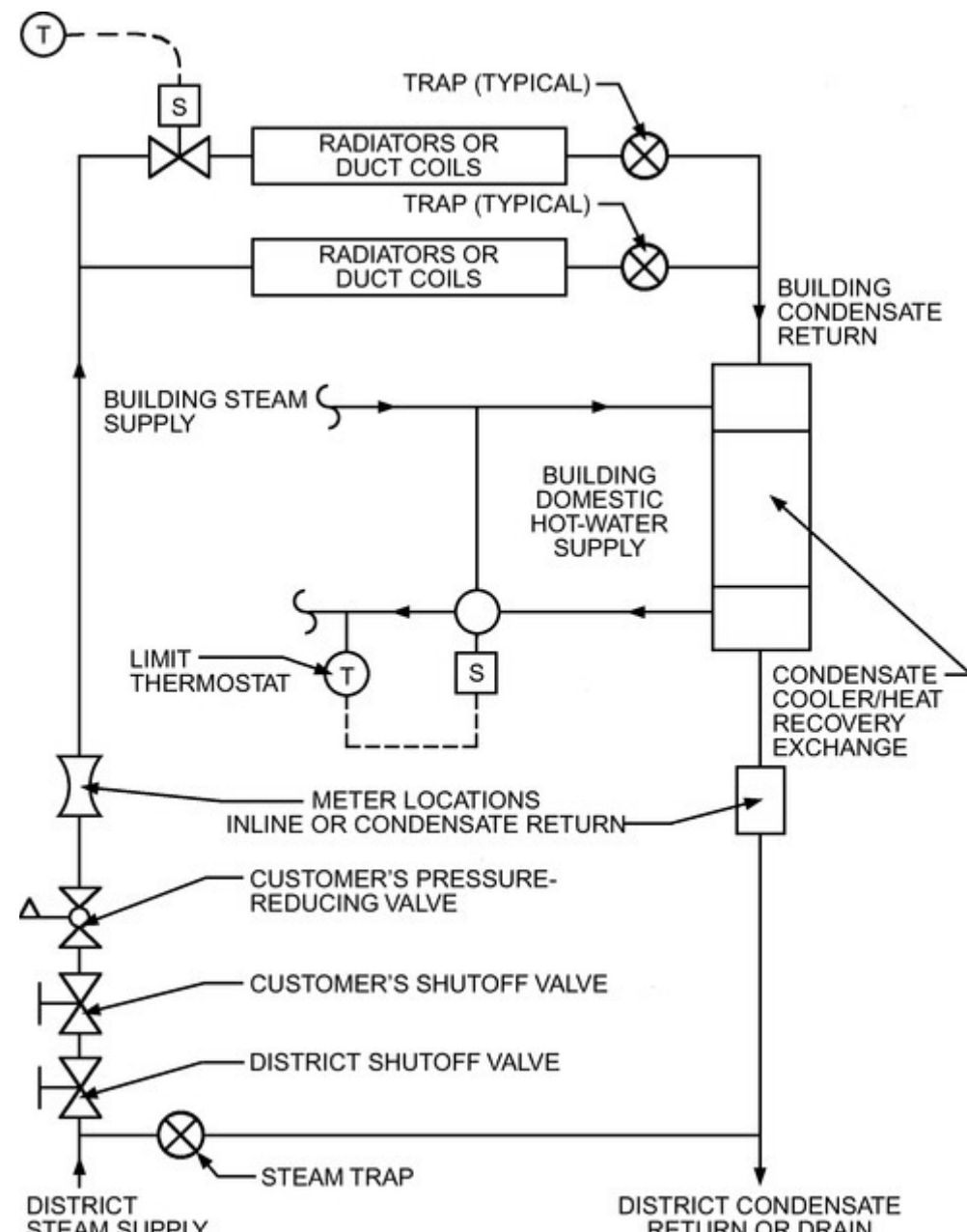


Figure 30. District/Building Interconnection with Heat Recovery Steam System

Other components of the steam connection may include condensate pumps, flowmeters (steam and/or condensate), and condensate conductivity probes, which may dump condensate if contaminated by unacceptable debris. Often, energy meters are installed on both the steam and condensate pipes to allow the district energy supplier to determine how much energy is used directly and how much energy (condensate) is not returned back to the plant. Using customer energy meters for both steam and condensate is desirable for the following reasons:

- Offers redundant metering (if the condensate meter fails, the steam meter can detect flow or vice versa)
- Bills customer accordingly for makeup water and chemical treatment on all condensate that is not returned or is contaminated
- Meter is in place if customer requires direct use of steam in the future
- Assists in identifying steam and condensate leaks
- Improves customer relations (may ease customer's fears of overbilling because of a faulty meter)
- Provides a more accurate reading for peak demand measurements and charges

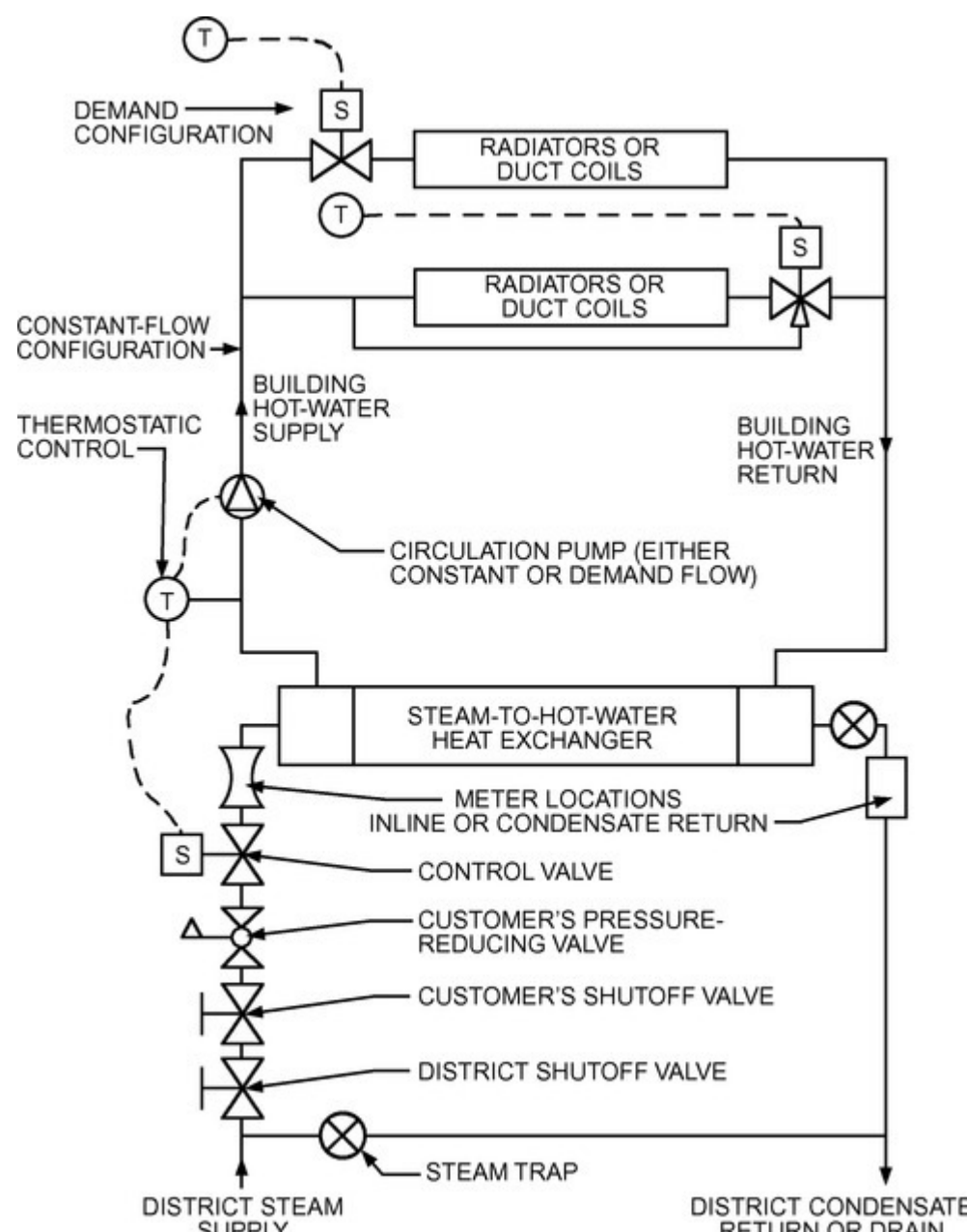


Figure 31. District/Building Interconnection with Heat Exchange Steam System

Monitor each level of steam pressure reduction as well as the temperature of the condensate. Where conductivity probes are used to monitor the quality of water returned to the steam plant, adequate drainage and cold-water quenching equipment may be required to satisfy local plumbing code requirements (temperature of fluid discharging into a sewer). The probe status should also be monitored at the control panel, to communicate high-conductivity alarms to the plant and, when condensate is being dumped, to notify the plant that a conductivity problem exists at a customer.

Table 14. Conversion Suitability of Heating System by Type			
Type of System	Low	Medium	High
Steam equipment			
One-pipe cast iron radiation	X		
Two-pipe cast iron radiation		X	
Finned-tube radiation		X	
Air-handling unit coils		X	
Terminal unit coils	X		
Hot-water equipment			
Radiators and convectors			X
Radiant panels			X
Unitary heat pumps			X
Air-handling unit coils			X
Terminal unit coils			X
Gas/oil-fired equipment			
Warm-air furnaces	X		
Rooftop units	X		
Other systems	X		
Electric equipment			
Warm-air furnaces	X		
Rooftop units	X		
Air-to-air heat pumps	X		
Other systems	X		

Source: Sleiman et al. (1990).

Building Conversion to District Heating
[Table 14](#) (Sleiman et al. 1990) summarizes the suitability or success rate of converting various heating systems to be served by a district hot-water system. As shown, the probability is high for water-based systems, lower for steam, and lowest for fuel oil or electric systems. Low-suitability systems usually require expensive replacement of the entire heating terminal and generating units with suitable water-based equipment, including piping, pumps, controls, and heat transfer media.

4.4 COMPONENTS

Heat Exchangers
Aside from their isolation function, heat exchangers act as the line of demarcation between ownership responsibilities of the different components of an indirect system. They transfer thermal energy and act as pressure interceptors for the water pressure in high-rise buildings. They also keep fluids from each side (which may have different chemical treatments) from mixing. [Figure 29](#) shows a basic building schematic including heat exchangers and secondary systems. Reliability of the installation is increased if multiple heat exchangers are installed. The number selected depends on the types of loads present and how they are distributed 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 turnaround as close as possible (oversized units may not perform as desired at maximum turnaround; therefore, using several smaller units optimizes the installation).
- Assess the critical nature of the load/operation/process to address reliability and redundancy. For example, if a building has 24 h 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 at 50% load each to allow one unit to be cleaned without interrupting building service. Separate heat exchangers should be capable of automatic isolation during low-load conditions to increase part-load performance.
- Determine customer's temperature and pressure design conditions. Some gasket materials for plate heat exchangers have limits for low pressure and temperature.
- Evaluate customer's water quality (i.e., use appropriate fouling factor).
- Determine available space and structural factors of the mechanical room.
- Quantify design temperatures. The heat exchanger may require rerating at a higher inlet temperature during off-peak hours.
- Calculate 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 or below) if existing pumps are to be reused. For chilled-water connections, investigate the existing chiller evaporator pressure drop to assist in this evaluation.

Size all heat exchangers with future expansion in mind. During selection, be aware that closer approach temperatures or low pressure drop require more heat transfer area and hence cost more and take up more space. Install strainers in front of any heat exchanger and control valve to keep debris from fouling surfaces or orifices.

Plate, shell-and-coil, and shell-and-tube heat exchangers are all used for indirect connection. Whatever heat transfer device is selected must meet the appropriate temperature and pressure duty, and be stamped/certified accordingly as pressure vessels. Whether using heat exchangers for heating or cooling, it is advantageous and recommended that if one side of the unit has variable flow, the other side should as well. This ensures that minimum pump and thermal energy is used to satisfy a load (Tredinnick 2007). See [Chapter 48](#) for more information on heat exchangers.

Plate Heat Exchangers (PHEs). These exchangers, which are used for steam, hot-water, and chilled-water applications, are available as gasketed units and in two gasket-free designs (brazed and all- or semiwelded construction). All gasketed PHEs, also known as plate-and-frame heat exchangers (HXs), consist of multiple gasketed embossed metal plates bolted together 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 because hot fluid flows on one side of the plate and cool fluid 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 gaps and the other in the even-numbered gaps.

Gaskets between the plates contain the two media in the plates and act as a boundary. Gasket failure does not cause the two media to mix; instead, the media leaks to the atmosphere. Gaskets can be either glued or clipped on. Gasketed PHEs are suitable for steam-to-liquid and liquid-to-liquid applications. 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.

Because PHEs require turbulent flow for proper heat transfer, pressure drops may be higher than those for comparable shell-and-tube models. 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 keeps surfaces clean from fouling. However, larger particles may become lodged in fine cavities between the plates and choke flow; automatic back-flushing valves may be used to address this issue. 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 are also capable of closer approach temperatures.

PHEs are typically used for district heating and cooling with water and for cooling lower water heat recovery (free cooling). Double-wall plates are also available for potable-water heating, chemical processes, and oil quenching. PHEs have three to five times greater heat transfer coefficients than shell-and-tube units and can achieve 1°F approach, but for economic reasons the approach is traditionally 2°F. Gasketed PHEs can be disassembled in the field to clean the plates and replace the gaskets. Typical applications go up to 365°F and 400 psig. Plates are typically made from stainless steel, but are available in titanium for more corrosive uses such as seawater cooling.

Brazed PHEs are suitable for steam, vapor, or water solutions. They feature a close approach temperature (within 2°F), large temperature drop, compact size, and a high heat transfer coefficient. Construction materials are stainless steel plates and sometimes frames brazed together with copper or nickel. Tightening bolts are not required as with 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 usually peak at a capacity of under 200,000 Btu/h (about 200 plates and 600 gpm) and are suitable for 435 psig and 435°F. Typical applications are district heating using hot water and refrigeration process loads. Double-wall plates are also available for domestic hot-water use. Avoid applications where the PHE may be exposed to large, sudden, or frequent changes in temperature and load, because of risk of thermal fatigue.

Welded PHEs can be used in any application for which shell-and-tube units are used that are outside the accepted range of gasketed PHE units, in liquid-to-liquid, steam-to-liquid, gas-to-liquid, gas-to-gas, and refrigerant applications. Construction is very similar to gasketed units except gaskets are replaced with laser welds. Materials are typically stainless steel, but titanium, monel, nickel, and various alloys are available. Models have design ratings that range from 500°F at 150 psig to 1000°F at 975 psig; however, they are available only in small sizes. Normally, these units are used in ammonia refrigeration and aggressive process fluids. They are more suitable to pressure pulsation or thermal cycling because they are thermal fatigue resistant. A semiwelded PHE is a hybrid of the gasketed and the all-welded units in which the plates are alternatively sealed with gaskets and welds.

Shell-and-Coil Heat Exchangers. These European-designed heat exchangers are suitable for steam-to-water and water-to-water applications and feature an all-welded and brazed construction. This counter/cross-flow heat exchanger consists of a hermetically sealed (no gaskets), carbon-steel pressure vessel with hemispherical heads. Copper or stainless helical tubes within are installed in a vertical configuration. This type of heat exchanger offers a high temperature drop and close approach temperature. Its vertical arrangement requires less floor space than other designs and has better heat transfer characteristics than shell-and-tube units.

Shell-and-Tube Heat Exchangers. These exchangers are usually a multiple-pass design. The shell is usually constructed from steel and the tubes are often of U-bend construction, usually 3/4 in. (nominal) OD copper, but other materials are available. These units are ASME U-1 stamped for pressure vessels.

Heat Exchanger Load Characteristics. To provide high ΔT under load conditions, variable flow is required on both sides of the heat exchanger (between Andro 1999; Skogestad and Milderstein 2002; Tredinnick 2007). Without variable flow on the customer side, more water flow is required on the district side during reduced load. This condition results in both increased pumping for the district energy provider as well as reduced ΔT . In addition, the customer side also experiences increased pumping costs without the use of variable flow. The specific degradation in ΔT and the increases in flow depend on the actual heat exchanger selection, and can easily be determined for a specific heat exchanger available from the heat exchanger manufacturer.

An example provided by Skogestad and Milderstein (2002) for a 427 ton design load indicates that, at 50% load and constant flow on the consumer side, 75% of the design flow would be required on the district cooling system side, compared to 45% if the consumer side used variable flow. However, consumer-side constant flow reduces the ΔT from the design value of 15°F to just 10°F at 50% load; when variable flow is used on the consumer's side of the PHE, the ΔT actually increased from the design value of 15°F to 16.7°F.

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 32](#), the consumer side of the heat exchanger has constant flow with the consumer-side design supply temperature of 42°F. The PHE has been sized such that, at 100% of design load, the district cooling return temperature is 54°F; thus, at design conditions, the ΔT is 14°F, again assuming 2°F approach. However, at 50% load with variable flow on the consumer's side, only 51% of the design flow is required on the district cooling side, and the return temperature actually increases to 56°F. Thus, under this load condition, the ΔT for the DHC increases from 9.6°F with constant flow to 16°F for variable consumer-side flow.

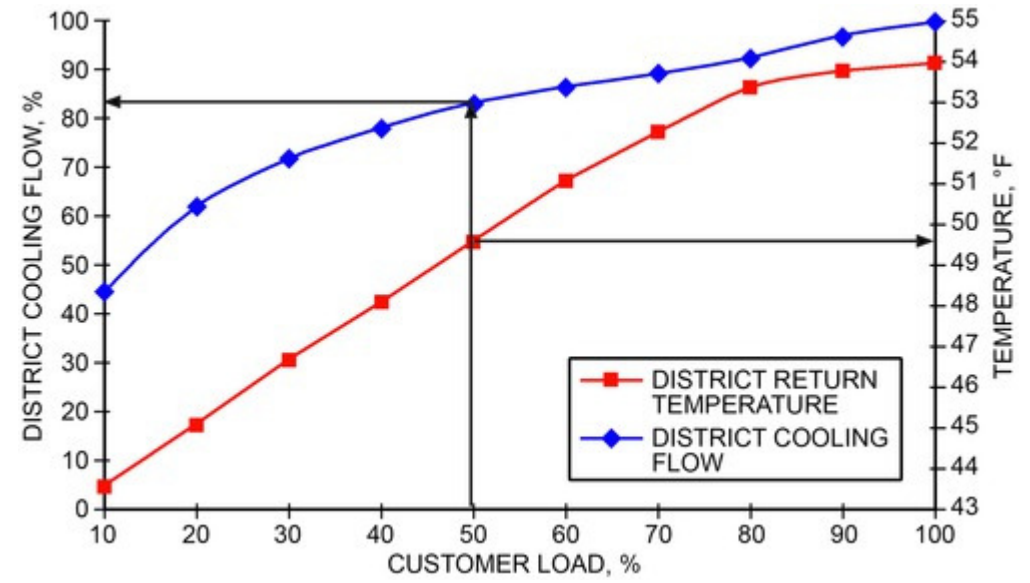


Figure 32. Plate Heat-Exchanger Performance with Constant Flow on Customer Side and Customer-Side Supply Temperature of 42°F (Tredinnick 2007)

[Figure 33](#), also from Tredinnick (2007), shows 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 and at 100% of design load, the PHE has been sized to yield a district cooling return temperature of 54°F; thus, at design conditions, the ΔT is 14°F, again assuming 2°F approach. However, at 50% load with variable flow on the consumer's side, only 51% of the design flow is required on the district cooling side, and the return temperature actually increases to 56°F. Thus, under this load condition, the ΔT for the DHC increases from 9.6°F with constant flow to 16°F for variable consumer-side flow.

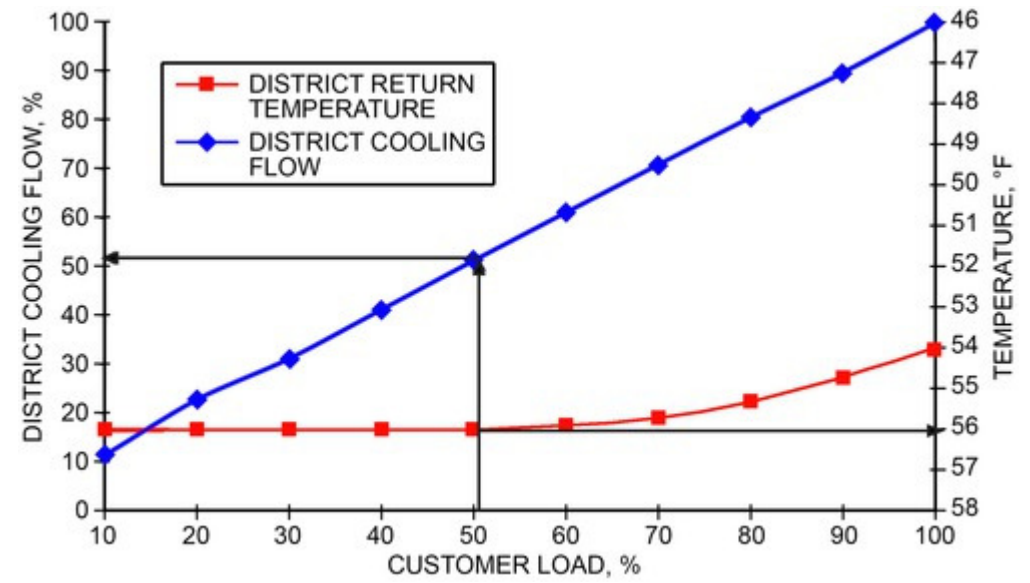


Figure 33. Plate Heat-Exchanger Performance with Variable Flow on Customer Side and Customer-Side Supply Temperature of 42°F (Tredinnick 2007)

Variable flow also saves electrical pump energy and aids in controlling comfort. These examples, as well as others (e.g., Perdue and Andro 1999) should make clear 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 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. Carefully analyze the valve actuator, because the shutoff requirements and control characteristics are totally different for a two-way valve than for a three-way valve. For more information on building conversion, see Skogestad and Milderstein (2002).

In theory, a cooling coil should have higher return water temperature when partially loaded than at full load, because the coil is oversized for the duty and thus has closer approach temperatures. In many real systems, as the load increases, the return water temperature tends to rise, and under low loads, 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 systems, after the flowmeter, control valves are the most important element in the interface with the district energy system because proper valve adjustment and calibration save energy. 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 slam closed. 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.

The wide range of flows and pressures expected makes selection of control valves difficult. Typically, only one control valve is required; however, for optimal response to load fluctuations and to prevent cavitation, two valves in parallel are often needed. 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 oversizing control valves complicates the selection process, so accurate method of 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 are typically specified.

Electronic control valves should remain in a fixed position when a power failure occurs and should be manually operable. Pneumatic control valves should close upon loss of air pressure. A manual override on the control valves allows the operator to control flow. All chilled-water control valves must fall in the closed position. Then, when any secondary in-building systems are deenergized, the valves close and will not bypass chilled water to the return system. All steam pressure-reducing valves should close as well.

Oversizing results in reduced valve and actuator life span and causes 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 hot-water systems, control valves are normally installed in the return line because the lower temperature in the line reduces the risk of cavitation and increases valve life. In chilled-water systems, control valves can be installed in either location; typically, however, they are installed the return line to reduce the potential for condensation on exposed external surfaces and to minimize water turbulence upstream of the flowmeter.

Instrumentation
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 pump control (water systems) or valve control for pressure-reducing stations (steam and water). Detailed recommendations on pressure, temperature, flow, and power transducers may be found in IDEA (2008a).

Table 15. Measuring Points and Derivative Parameters for Remote Monitoring and Control of Indirect Consumer Interconnection		
Measured Point/Parameter	Location	
	District Cooling Side	Consumer Side
Temperature	Supply, return	Supply, return
Pressure	Supply, return	Supply, return
Differential pressure	At building entrance, heat exchanger(s), control valve(s), strainer	Heat exchanger(s)
Flow rate	Supply or return	
Energy transfer	Supply and return	
Position of control valve(s)	Yes	Optional
Variable-speed drive percentage(s)	Yes (not typical)	Yes

Temperature sensors need to be located by the exchangers being controlled rather than in the common pipe. Improperly located sensors cause one control valve to open and others to close, resulting in unequal loads in the exchangers. Temperature sensors used in energy metering applications should be matched-pair, four-wire, 1000 Ω for increased accuracy. [Table 15](#) lists 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 or recommended if the district heating utility assumes some responsibility for operation of the building system or if the building and district cooling system belong to a single owner.

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 (often) control. Pressure transducers should be specified to provide accuracy of $\pm 1\%$ of full scale and typically resolution of 0.1 psi (IDEA 2008a).

Controller
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, energy calculations, alarming for parameter outside normal, and monitoring and control of all components. Often, the controller has a small battery power supply to preserve settings and billing information. 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 for cooling, the effect on the customer's ability to dehumidify properly with an elevated entering coil temperature should be investigated carefully.

Pressure Control Devices
If the steam or water pressure delivered to the customer is too high for direct use, it must be reduced. Similarly, pressure-reducing or pressure-sustaining valves may be required if building height creates a high static pressure and influences the district system's return water pressure. Water pressure can also be reduced by control valves or regenerative turbine pumps. The risk of using pressure-regulating devices to lower pressure on the return line is that if they fail, the entire distribution system is exposed to their pressure, and overpressure will occur.

Table 16. Flowmeter Characteristics			
Meter Type	Accuracy	Range of Control	Straight Piping Requirements (Length in Pipe Diameters)
Orifice plate	$\pm 1\%$ to 5% full scale	3:1 to 5:1	High (>5 psi)
Electromagnetic	$\pm 0.15\%$ to 1% rate	30:1 to 100:1	Low (<3 psi)
Vortex	$\pm 0.5\%$ to 1.25% rate	0.1 to 25:1	Medium (3 to 5 psi)
Turbine	$\pm 0.15\%$ to 0.5% rate	10:1 to 50:1	Medium (3 to 5 psi)
Ultrasonic	$\pm 1\%$ to 5% rate	>10:1 to 100:1	Low (<3 psi)

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 effect of the pressure. Provide vacuum vents at the top of the building's water risers to introduce air into the piping in case the vertical water column collapses.

Flow and Energy Metering
All thermal energy or power delivered to customers or end users for billing or revenue by a commercially operated district energy system must be metered. The type of meter selected depends on the fluid to be measured, accuracy required, and expected turndown of flow to meet the low-flow and maximum-flow conditions. It is important that the meter be sized accurately for the anticipated loads and not oversized, because this will lead to inaccuracies. Historical, metered, or benchmarked data should be used when available if the actual load is not accurately known.

Steam may be used for direct comfort heating or to power absorption chillers for cooling. It is typically measured by using the differential pressure across calibrated orifices, nozzles, or venturi tubes; or by pitot tubes, vortex-shedding meters, or condensate meters. In the United States, the customary commercial unit is pounds of steam per hour with a heat equivalent typically assumed to be 1000 Btu/lb, but for more precise thermal metering, the meters are coupled with steam quality (temperature and pressure) differential heat content measuring devices. Care must be taken to deliver dry steam (superheated or saturated without free water) to the customer. Dry steam is delivered by installing an adequate trap just ahead of the customer's meter (or ahead of the customer's process when condensate meters are used).

When condensate meters are used, care must be taken to ensure that all condensate from the customer's process, but only such condensate, goes to the condensate meter. This may not be possible if the steam is used directly for humidification purposes and hence no condensate is returned. In this case, steam meters are preferable.

IDHA (1969) and Sultz and Kitto (1992) have more information on steam metering. For steam, as with hot- and chilled-water system metering, electronic and computer technology provide direct, integrating, and remote input to central control/measurement energy management systems.

Hot- and chilled-water systems are metered by measuring the temperature differential between the supply and return lines and the flow rate of the energy transfer medium. Thermal (RtU or kWh_{th}) meters compensate for the actual volume and heat content characteristics of the energy transfer medium. 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.

Water flow can be measured with a variety of meters, usually pressure differential, turbine or propeller, or displacement meters. [Chapter 36 of the 2013 ASHRAE Handbook—Fundamentals](#), the *District Heating Handbook* (IDHA 1983), 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 16](#). The data in the table only provide general guidance; meter manufacturers should be contacted for data specific to their products.

The meter should be located upstream of the heat exchanger and the control valve(s) should be downstream from the heat exchanger. This installation minimizes the possible formation of bubbles in the flow stream and provide 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 $\pm 1\%$ 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. ASHRAE Standard 125 describes a test method for rating liquid thermal meters. Several European countries have developed performance standards and/or test methods for thermal meters, and CEN Standard EN 1434, 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).

For district energy cogeneration systems that send out and/or accept electric power to or from a utility grid, demand and usage meters must meet the existing utility requirements. For district energy systems that send out electric power directly to customers, the electric demand and usage meters must comply with local and state regulations. American National Standards Institute (ANSI) standards are established for all customary electric meters.

4.5 TEMPERATURE DIFFERENTIAL CONTROL

Maintaining a high water system temperature differential ΔT between supply and return lines is most cost effective because it allows smaller pipes to be used in the primary distribution system. These savings must be weighed against higher building conversion costs that may result from the need for a low primary return temperature.

For district heating and cooling providers, hydronic system efficiency is usually measured in terms of the temperature differential. Proper control of the district heating and cooling temperature differential is not dictated at the plant but at the consumer. If the consumer's system is not compatible with the temperature parameters of the DHC system, operating efficiency suffers unless components in the consumer's system are modified. Low system ΔT requires additional equipment to be energized, thus using more energy to satisfy the flow needs than the actual load demand requires. Therefore, the customer's ΔT must be monitored and controlled.

To optimize the ΔT , meet the customer's chilled-water demand, and save pump energy, flow from the plant should vary. Chilled-water flow in the customer's side must be varied as well. Terminal units in the building connected to the chilled-water loop (e.g., air-handling units, fan-coils) may require modifications (change 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 to 14 fins-per-inch coils are the minimum size coil applied to central station air-handling units to provide adequate performance. With this type of coil, the return water temperature rise should range from 12 to 16°F at full load. Coil performance at reduced loads should be considered, as well; therefore, 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 are sized for an entering water temperature several degrees above the main chilled-water plant supply temperature. In theory, a cooling coil should have higher return water temperature when partially loaded than at full load because the coil is oversized for the duty and hence has closer approach temperatures.

The system designer must consider the aspect of return water temperature control, especially on chilled-water systems where the ΔT is smaller than heating systems, so only a degree or two has a huge effect on system performance. Obtaining high temperature differentials is the single most important factor in operating an efficient plant. Active return water control requires adding a modulating control valve that closes if the temperature returns from the customer coils below the set point for cooling or above the set point for heating. Typically, these valves are industrial-grade ball valves because of their turndown and leaktight characteristics. Butterfly valves should not be used for this function.

[Figure 25](#) shows the proper location of the ΔT control valve. It is controlled from the customer return water temperature. If the water comes back too cool, the valve closes to achieve a higher return water temperature back to the plant. Although experience shows that this may increase the part-load system ΔT by 1 to 2°F, it does come with risks. If the customer's building loop supply water temperature is not monitored, there is a risk that the temperature will become too high to adequately cool or dehumidify; hence, the secondary control feature is typically added to set an upper limit. In addition, the customer's cooling coil control valves may be pulled by the building automation system (BAS). If a certain percentage of cooling control valves are wide open and the leaving air temperature on the air-handling unit is too high, the ΔT control valve allows more cold water from the plant to enter the building.

4.6 OPERATION AND MAINTENANCE

As with any major capital equipment, care must be exercised in operating and maintaining district heating and cooling systems. Both central plant equipment and terminal equipment located in the consumer's building must be operated within intended parameters and maintained on a schedule as recommended by the manufacturer. Thermal distribution systems, especially buried systems, which are out of sight, may suffer from inadequate maintenance. To maintain the distribution system's thermal efficiency as well as its reliability, integrity, and service life, periodic preventative maintenance is strongly recommended. In the case of steam and hot-water distribution systems, it may also be a matter of due diligence on the part of the owner/operator to ensure system integrity, to avoid thermal damage to adjacent property or harm to individuals coming in contact with the system or its thermal effects. Establishing an adequate preventive maintenance program for the distribution system requires consulting the system manufacturer's recommendations (where applicable) as well as considering the type, age, and condition of the system and its operating environment. See Phetteplace et al. (2013a, 2013b) for additional information on operation and maintenance for district cooling and heating systems, respectively.

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