Related Commercial Resources

CHAPTER 11

DISTRICT HEATING AND COOLING

DISTRICT heating and cooling (DHC) 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.

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.

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 is high. A high load density is needed to cover the capital investment for the transmission and distribution system, which usually constitutes most of the capital cost for the overall system, often ranging from 50 to 75% 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. Low-density residential areas have usually not been attractive markets for district heating, although there have been some successful applications. District heating is best suited to areas with a high building and population density in relatively cold climates. District cooling applies in most areas that have appreciable concentrations of cooling loads, usually associated with tall buildings.

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 source, solar energy, 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.

Chilled water can be produced by

- Absorption refrigeration machines

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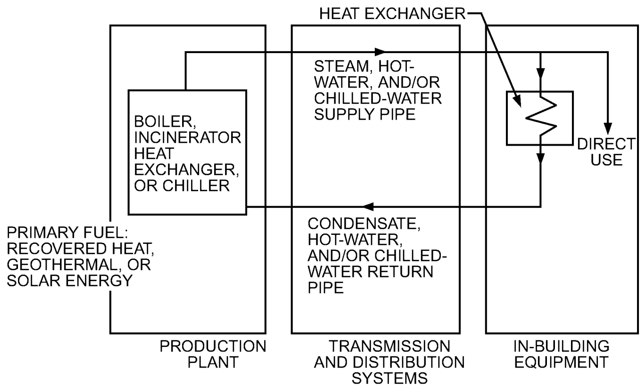


Fig. 1 Major Components of District Heating System

- 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

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, it is important to optimize use.

The third component is the **consumer system**, which includes 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, 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 systems or isolated by a heat exchanger (see the section on Consumer Interconnections).

BENEFITS

Environmental Benefits

Emissions from central plants are easier to control than those from individual plants and, in aggregate, are lower because of higher quality of equipment, seasonal efficiencies and level of maintenance, and lower system heat loss. A central plant that burns high-sulfur coal can economically remove noxious sulfur emissions, where individual combustors could not. Similarly, the thermal energy from municipal wastes can provide an environmentally sound system. Cogeneration of heat and electric power allows for combined efficiencies of energy use that greatly reduce emissions and also allow for fuel flexibility. In addition, refrigerants and other CFCs 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 ammonia systems for cooling.

Consumer Economic Benefits

A district heating and cooling system offers the following economic benefits. Even though the basic costs are still borne by the central plant owner/operator, because the central plant is large the customer can realize benefits of economies of scale.

Operating Personnel. One of the primary advantages 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 other use.

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 can several smaller units. When strict regulations must be met, additional pollution control equipment is also more economical for larger plants. Cogeneration of heat and electric power results in much higher overall efficiencies than are possible from separate heat and power 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. Central plants generally have efficient base-load units and less costly peaking equipment for use in extreme loads or emergencies.

PRODUCER ECONOMICS

Available Fuels. Smaller heating plants are usually designed for one type of fuel, which is generally gas or oil. Central DHC plants can operate on less expensive coal or refuse. Larger facilities can often be designed for more than one fuel (e.g., coal and oil), and combined with power generation (see Chapter 7 for information on combined heat and power systems).

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 tradeoff 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. 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.

The selection of temperature and pressure is crucial because it 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 the four components of (1) concept planning, (2) design, (3) construction, and (4) consumer interconnections.

Concept Planning. In concept planning, three areas are generally reviewed. First, the **technical feasibility** of a DHC system must be considered. Conversion of an existing heat source, for example, usually requires the services of an experienced power plant or DHC engineering firm.

Financial feasibility is the second consideration. For example, a municipal or governmental body must consider availability of bond financing. **Alternative energy choices** for potential customers must be reviewed because consumers are often asked to sign long-term contracts in order to justify a DHC system.

Design. The distribution system accounts for a significant portion of the initial investment. Distribution design depends on the heat transfer medium chosen, its operating temperature and pressure, and the routing. Failure to consider these key variables results in higher-than-planned installation costs. An analysis must be done to optimize insulating properties. The section on Economical Thickness for Pipe Insulation discusses determining insulation values.

Construction. The construction costs of the central plant and distribution system depend on the quality of the concept planning and design. Although the construction cost usually accounts for most of the initial capital investment, neglect in any of the other three areas could mean the difference between economic success and failure. Field changes usually increase the final cost and delay start-up. Even a small delay in start-up can adversely affect both economics and consumer confidence. It is extremely important that the contractors have experience commensurate with the project. 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 piping system
- Type of distribution system installation (e.g., direct buried, tunnel)
- Depth of bury and restoration of existing conditions (e.g., city streets, green areas)
- Below-grade conflict resolutions
- Economies of scale

Sample construction cost unit pricing is as follows, but the designer is cautioned that cost can vary widely:

- Cooling plant (building, chillers, cooling towers, pumps, piping, controls) = \$1500 to \$2600 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 = \$500 to \$1000 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, 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 a system being installed in an established business district.

Consumer Interconnection. These costs are usually borne by the consumer. 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 to a hot-water DHC system may be too costly, even though the cost of energy is lower.

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 so-called satellite plants are sometimes used in situations where a customer's building is located in an area where distribution piping is not yet installed.

HEATING AND COOLING PRODUCTION

Heating Medium

In plants serving hospitals, industrial customers, or 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 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.

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) condensed and cooled to 180°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 (Aamot 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.

Return System. Condensate return systems require more maintenance than hot-water return systems because hot-water systems function as a closed loop with very low makeup water requirements. For condensate return systems, corrosion of piping and other components, particularly in areas where feedwater is high in bicarbonates, is a problem. Nonmetallic piping has been used successfully in some applications, such as systems with pumped returns, where it has been possible to isolate the nonmetallic piping from live steam.

Similar concerns are associated with condensate drainage systems (steam traps, condensate pumps, and receiver tanks) for steam supply lines. Condensate collection and return should be carefully considered 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 consumer. 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 the use of a higher pressure class of piping and/or booster pumps.

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. 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 materials such as polyurethane foam insulation and nonmetallic conduits.

Hot-water systems are divided into three temperature classes:

- High-temperature systems supply temperatures over 350°F.
- Medium-temperature systems supply temperatures in the range of 250 to 350°F.
- Low-temperature 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.

In many instances, existing equipment and processes require the use of steam. See the section on Consumer Interconnections for further information.

Heat Production

Fire-tube and water-tube boilers are available for gas/oil firing. 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 underfeed 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.

Cooling Supply

Chilled water may be produced by an absorption refrigeration machine or by vapor-compression equipment driven by electric, turbine (steam or combustion), or internal combustion engines. The chilled-water supply temperature for a conventional system ranges from 40 to 44°F. A 12°F temperature difference (Δt) results in a flow rate of 2 gpm/ton of refrigeration. Because of the cost of the distribution system piping, large chilled-water systems are sometimes operated at lower supply 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, temperatures as low as 34°F have been used.

Multiple air-conditioning loads interconnected with a central chilled-water system provide some economic advantages and energy conservation opportunities. In addition, central plants afford the opportunity to consider the use of refrigerants such as ammonia that may be impractical for use in individual buildings. The size of air-conditioning loads served, as well as the 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 are required to meet the loads economically and provide redundancy. In some areas, plants over 20,000 tons are common for systems exceeding 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) within the plant as well as large distribution headers in streets already congested with other utilities, making piping layout problematic.

An economic evaluation of piping and pumping costs versus chiller power requirements can establish the most suitable supply water temperature. 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 the plant design and loading, thermal storage can reduce chiller equipment requirements and lower operating costs. By shifting a part of the chilling load, chillers can be sized closer to the average load than the peak load. Shifting the entire refrigeration load to off peak requires the same (or slightly larger) chiller machine capacity, but removes all of the electric load from the peak period. Because many utilities offer lower rates during off-peak periods, operating costs for electric-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 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.

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. 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 (Andrepont 1995); furthermore, the cost of above-grade tanks is usually less than below-grade tanks. Chapter 50 has information on thermal storage.

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. Figure 2 presents a layout for a large hot-water/chilled-water plant.

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 though the piping. 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.

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.

Emission Control. Environmental 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

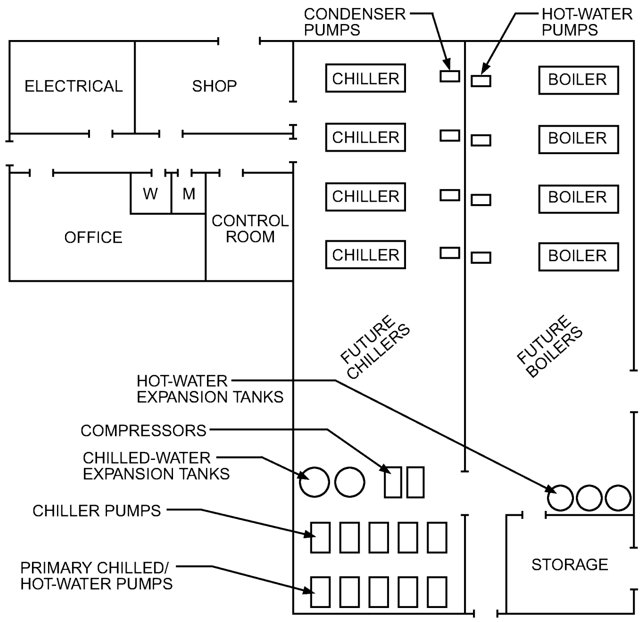


Fig. 2 Layout for Hot-Water/Chilled-Water Plant

for continuous online use, they are less suited for cyclic operation. Heating and cooling 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 18 of the 2005 *ASHRAE Handbook—Fundamentals* and Chapter 29 have information on air pollution and its control.

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.

Constant Flow

Constant flow is generally applied only to smaller systems where simplicity of design and operation are important and where distribution pumping costs are low. Chillers may be arranged in series to handle higher temperature differentials. Flow volume through a full-load distribution system depends on the type of constant-flow system used. One technique connects the building and its terminals across the distribution system. The central plant pump circulates chilled water through three-way valve controlled air-side terminal units (constant-volume direct primary pumping). Balancing problems may occur in this design when many separate flow circuits are interconnected (Figure 3).

Constant-flow distribution is also applied to in-building circuits with separate pumps. This arrangement isolates the flow balance problem between buildings. In this case, flow through the distribution system can be significantly lower than the sum of the flows needed by the terminal if the in-building system supply temperature is higher than the distribution system supply temperature (Figure 3).

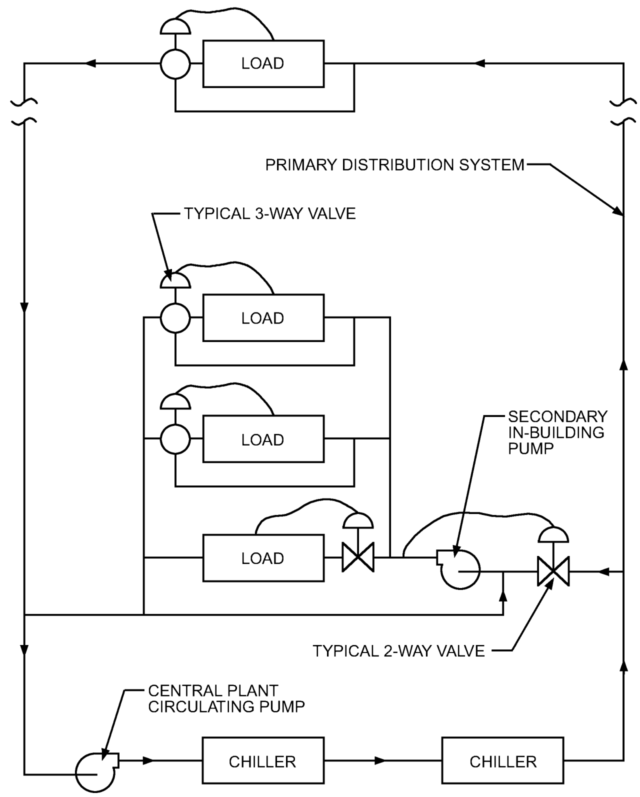


Fig. 3 Constant-Flow Primary Distribution with Secondary Pumping

The water temperature rise in the distribution system is determined by the connected in-building systems and their controls.

In constant-flow design, chillers arranged in parallel have decreased entering water temperatures at part load; thus, several machines may need to run simultaneously, each at a reduced load, to produce the required chilled-water flow. In this case, chillers in series are better because constant flow can be maintained though the chilled-water plant at all times, with only the chillers required for producing chilled water energized. 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. Modern control systems mitigate the reason for constant flow through the chillers, and variable flow should be investigated for larger systems.

Variable Flow

Variable-flow design can improve 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. Flow-throttling (two-way) valves 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

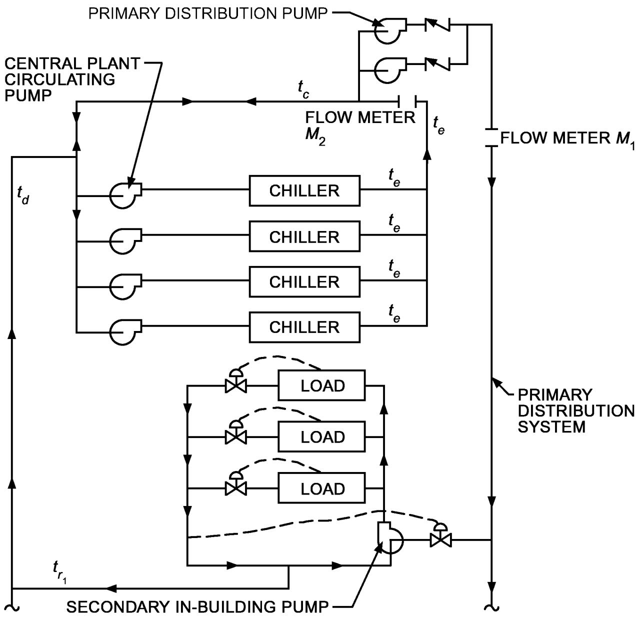


Fig. 4 Variable-Flow Primary/Secondary Systems

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 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. If the designer has control over the design of each in-building system, this pumping method can be achieved in a reasonable manner. In retrofit situations 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, hydraulic isolating piping and pumping design should be used to ensure that two-way control valves are subjected only to the differential pressure established by the in-building pump. Figure 4 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 must 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.

In very large systems, a design known as **distributed pumping** may be used. Under this approach, the distribution pumps in the central plant are eliminated. Instead, the distribution system pumping load is borne by the pumps in the user buildings. In cases 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 the addition of a large load anywhere in the system affects the pressure drop of each distribution pump.

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 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.

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 for a minimum temperature differential of 12 to 16°F. A 12 to 20°F maximum temperature differential with a 10 to 12°F minimum temperature differential can be achieved with this design.
- Limit the use of constant-flow systems to relatively small central chilled-water plants. Investigate chillers arranged in series.
- Larger central chilled-water plants can benefit from primary/secondary or primary/secondary/tertiary pumping with constant flow in central plant and variable flow in the distribution system. 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. Minimum flows on chiller evaporators should be investigated with the manufacturer 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. See Chapter 42 for more information on valves.
- The lower practical limit for chilled water supply temperatures is 39°F. Temperatures below that should be carefully analyzed, although systems with thermal energy storage may operate advantageously at lower temperatures.

DISTRIBUTION SYSTEM

HYDRAULIC CONSIDERATIONS

Objectives of Hydraulic Design

Although 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; COWIconsult 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 10, as well as in IDHA (1983). For water systems, consult Chapter 12 and IDHA (1983).

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 and sudden valve closures. A simplified method to determine maximum resultant pressure may be found in Chapter 36 of the 2005 *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

Friction 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. The calculation of pressure loss is discussed in Chapters 2 and 36 of the 2005 *ASHRAE Handbook—Fundamentals*.

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 done 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 36 of the 2005 *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 fps 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 (Bøhm 1988). Other studies indicate that higher levels of pressure loss may be acceptable (Stewart and Dona 1987) and warranted from an economic standpoint (Bøhm 1986; Koskelainen 1980; Phetteplace 1989).

When establishing design flows for thermal distribution systems, the diversity of consumer demands should be considered (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.

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.

Condensate Drainage and Return

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. If it is not possible to slope the steam pipe in the direction of steam flow, 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 will 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.

Schedule 80 steel piping is recommended for condensate lines because of the extra allowance for corrosion its 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 10. Information on sizing condensate return piping may be found in Chapter 36 of the 2005 *ASHRAE Handbook—Fundamentals*.

THERMAL CONSIDERATIONS

Thermal Design Conditions

Three thermal design conditions must be met to ensure satisfactory system performance:

1. The "normal" condition used for the life-cycle cost analysis determines appropriate insulation thickness. Average values for the temperatures, burial depth, and thermal properties of the materials are used for design. If the thermal properties of the insulating material are expected to degrade over the useful life of the system, appropriate allowances should be made in the cost analysis.
2. 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, the thermal conductivity of each component must be taken at its maximum value, and the temperatures must be assumed to take on their extreme values, which would result in the greatest temperature difference between the carrier medium and the soil or air. The burial depth will normally be at its lowest value for this calculation.
3. 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 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, Bottorf (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.

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 1. Material properties such as thermal conductivity, density, compressive strength, moisture absorption, dimensional stability, and combustibility are typically reported in ASTM standard for the respective material.

Table 1 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 parenthesis 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/ft³	15/22		6.7 to 9.2	8 to 11
Maximum temperature, °F	1200	250	800	850
Compressive strength (min),[b] kPa	700 at 5% deformation		450	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 available. They may not represent installed performance of insulation under actual conditions that differ substantially from test conditions. The manufacturer should have design values.
[b]Compressive strength for cellular glass shown is for flat material, capped as per ASTM C240.

Table 2 Effect of Moisture on Underground Piping System Insulations

Characteristics	Polyurethane[a]	Cellular Glass	Mineral Wool[b]	Fibrous Glass
Heating Test	Pipe temp. 35°F to 260°F Water bath 46°F to 100°F	Pipe temp. 35°F to 420°F Water bath 46°F to 100°F	Pipe temp.35°F to 450°F Water bath 46°F to 100°F	Pipe temp. 35°F to 450°F Water bath 46°F to 100°F
Length of submersion time to reach steady-state *k*-value	70 days	See Note C	10 days	2 h
Effective *k*-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 (Note C). 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	See Note C	Conduction and convection	Conduction and convection
Length of time for specimen to return to dry steady-state *k*-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 *k*-value	16 days	Data recorded at 4 days constant at 12 days	6 days	1/2 h
Effective *k*-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 *k*-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/ft^3.
[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 96 h boiling water test.
[c]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; the period of dynamic process was about 20 min. Cracks had negligible effect on the thermal conductivity of dry cellular glass insulation before and after submersion. No cracks formed during cooling test.

Table 3 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

Some properties have more than one associated standard. For example, thermal conductivity for insulation material in block form may be reported using ASTM C177, C518, or C1114. Thermal conductivity for insulation material fabricated or molded for use on piping is reported using ASTM 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 2. 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 unjacketed to simulate installation in a conduit.

Painting chilled-water piping before insulating is recommended in areas of high humidity. Insulations used today for chilled water include polyurethane and polyisocyanurate cellular plastics, phenolics, and fiberglass. With the exception of fiberglass, the rest can form acidic solutions (pH 2 to 3) once they hydrolyze in the presence of water. The acids emanate from the chlorides, sulfates, and halogens added during manufacturing to increase fire retardancy or expand the foam. Phenolics can be more than six times more corrosive than polyurethane because of acids used in their manufacture and can develop environments to pH 1.8. The easiest way to mitigate corrosion is to paint the pipe exterior with a strong rust-preventative coating (two-coat 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 barrier is required.

Soil. If an analysis of the soil 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 3 may be used as an estimate when detailed information on the soil is not known. Because dry soil is rare in most areas, a low moisture content should be assumed only 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 will migrate 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.

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 earth and fluid temperatures and the thermal insulation. 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.

To compute transient heat gains or losses in underground piping systems, numerical methods that approximate any physical problem and include such factors as the effect of temperature on thermal properties must be used. For most designs, numerical analyses may not be warranted, except where the potential exists to thermally damage something adjacent to the distribution system. Also, complex geometries may require numerical analysis. Albert and Phetteplace (1983), Minkowycz et al. (1988), and Rao (1982) have further information on numerical methods.

Steady-state calculations are appropriate for determining the annual heat 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·ft^2·°F).

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 Considerations. For example, if the purpose of the calculation is to determine if 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:

1. Use the average annual air temperature to approximate the average annual soil temperature. This estimate is appropriate when the objective of the calculation is to yield the average heat loss over the yearly weather cycle. Mean annual air temperatures may be obtained from various sources of climatic data (e.g., CRREL 1999).
2. 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 the information found in Chapter 28 of the 2005 *ASHRAE Handbook—Fundamentals*.
3. For systems that are buried at other than shallow depths, maximum/minimum undisturbed soil temperatures may be estimated as a function of depth, soil thermal properties, and prevailing climate. This estimate is appropriate when checking the temperatures in a system to determine if 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, ft^2/day
t_{ms} = mean annual surface temperature, °F
A_s = surface temperature amplitude, °F

CRREL (1999) lists values for the climatic constants t_{ms} and A_s for various regions of the United States.

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/ft^3
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.

4. 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 [i.e., Equations (6) and (7)] when estimates of heat loss/gain as a function of time of year are desired. The substitution of the undisturbed soil temperatures at the pipe depth allows the steady-state equations to be used as a first approximation to the solution to the actual transient heat transfer problem with its annual temperature variations at the surface. The following equation may be used to estimate the undisturbed soil temperature at any depth at any point during the yearly weather cycle (ASCE 1996). (*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

CRREL (1999) lists values for the climatic constants t_{ms}, A_s, and θ_{lag} for various regions of the United States. Equation (3) may be used to calculate soil thermal diffusivity.

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, the following method may be used 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, McCabe et al. (1995) observed significant temperature variations caused by the type of surfaces over district heating and cooling systems.

As a first approximation, an **effectiveness 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 = effectiveness 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·ft^2·°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).

Single Uninsulated Buried Pipe

For this case (Figure 5), 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.

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

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

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

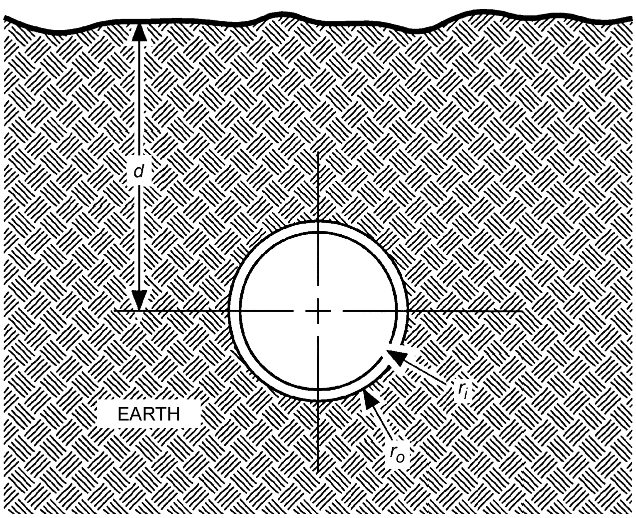


Fig. 5 Single Uninsulated Buried Pipe

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

$$R_p = \frac{\ln(r_o/r_i)}{2\pi k_p} \tag{8}$$

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 1. 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 within close proximity. Assume the average annual soil temperature is 60°F.

Solution: Calculate the thermal resistance of the pipe using Equation (8):

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

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).]

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

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

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

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_f = 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.

Single Buried Insulated Pipe

Equation (8) can be used to calculate the thermal resistance of any concentric circular region of material, including an insulation layer (Figure 6). When making calculations using insulation thickness, actual thickness rather than nominal thickness should be used to obtain the most accurate results.

Example 2. 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 1. Calculate the thermal resistance of the insulation layer from Equation (8) as follows:

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

For the PVC jacket material, use Equation (8) again:

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

The thermal resistance of the soil as calculated by Equation (7) decreases slightly to R_s = 0.51 h·ft·°F/Btu because of the increase in the outer radius of the piping system. The total thermal resistance is now

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

The 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.

Single Buried Pipe in Conduit with Air Space

Systems with air spaces may be treated by adding an appropriate resistance for the air space. For simplicity, assume a heat transfer coefficient of 3 Btu/h·ft^2·°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

$$R_a = 1/(3 \times 2\pi \times r_{oi}) = 0.053/r_{oi} \tag{9}$$

where

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

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). The effect of radiation in the annulus should also be considered when high temperatures are expected within the air space. For the treatment of radiation, refer to Siegel and Howell (1981).

Example 3. 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_i = 0.026 Btu/h·ft·°F at 200°F and k_i = 0.030 Btu/h·ft·°F at 300°F.

The pipe will be encased in a steel conduit with a concentric air gap of 1 in. The steel conduit will be 0.125 in. thick and will have a corrosion resistant coating approximately 0.125 in. thick. The pipe will be 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. The 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 temperature of the insulation 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, the insulation thermal conductivity k_i = 0.028 Btu/h·ft·°F. Then calculate insulation thermal resistance using Equation (8):

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

Calculate the thermal resistance of the air space using Equation (9):

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

Calculate the thermal resistance of the soil from Equation (7):

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

The total thermal resistance is

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

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_{po} = 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 h·ft·°F/Btu. The 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°F, and no further calculations are needed.

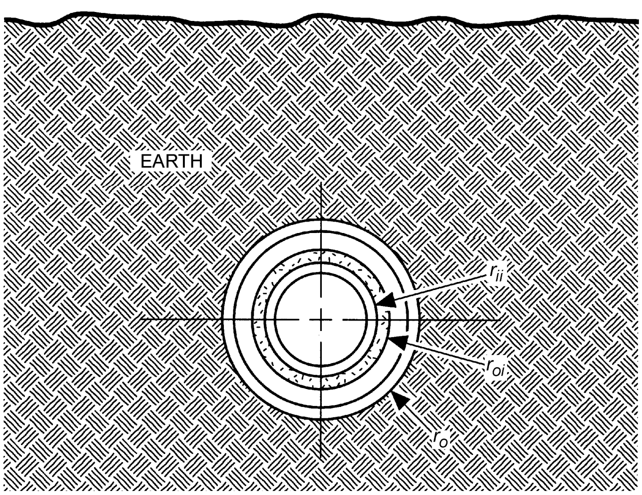


Fig. 6 Single Insulated Buried Pipe

Single Buried Pipe with Composite Insulation

Many systems are available that 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). The calculation of the heat loss and/or temperature in a composite system is straightforward using the equations presented previously.

Example 4. A high-temperature water line operating at 400°F is to be 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 will be placed around the calcium silicate insulation. The polyurethane insulation will be encased in a 0.5 in. thick fiberglass-reinforced plastic (FRP) jacket. The piping system will be 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/ft^3

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 soil temperature rather than the average annual soil temperature must be found. Also, the lowest anticipated soil thermal conductivity and the 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). CRREL (1999) shows the values for climatic constants for this region as t_{ms} = 71.8°F and A_s = 15.0°F. Soil thermal diffusivity may be estimated using Equation (3).

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

Then Equation (1) is used to calculate the maximum soil temperature at the installation depth as follows:

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

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

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

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

Calculate the thermal resistance of the soil from Equation (7):

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

The total thermal resistance is

$$R_t = 1.42 + 2.14 + 2.89 = 6.45 \text{ h·ft·°F/Btu}$$

The first estimate of the heat loss is then

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

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.

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

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

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

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°F for the calcium silicate and (328 + 220)/2 = 274°F for the polyurethane foam. Thus, the insulation thermal conductivity for the calcium silicate would be interpolated to be 0.045 Btu/h·ft·°F, and the resulting thermal resistance is $R_{i,1}$ = 1.32 h·ft·°F/Btu.

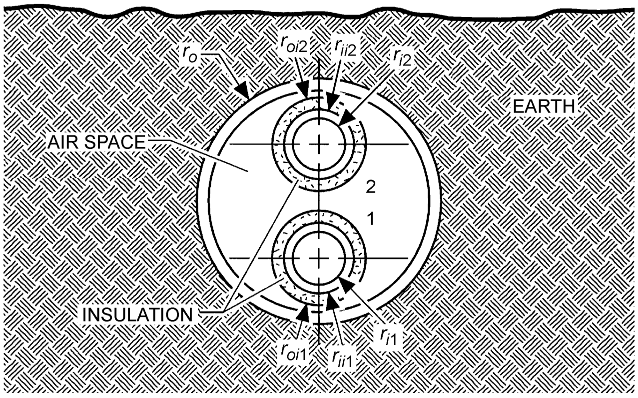


Fig. 7 Two Pipes Buried in Common Conduit with Air Space

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 h·ft·°F/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°F, and the outer surface temperature of the polyurethane foam is calculated to be $t_{io,2}$ = 226°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 (EuHP 1991). 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.

Two Pipes Buried in Common Conduit with Air Space

For this case (Figure 7), make the same assumption as made in the previous section, Single Buried Pipe in Conduit with Air Space. For convenience, add some of the thermal resistances as follows:

$$R_1 = R_{p1} + R_{i1} + R_{a1} \tag{10}$$

$$R_2 = R_{p2} + R_{i2} + R_{a2} \tag{11}$$

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

$$q = \frac{[(t_{f1} - t_s)/R_1] + [(t_{f2} - t_s)/R_2]}{1 + (R_{cs}/R_1) + (R_{cs}/R_2)} \tag{12}$$

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:

$$t_a = t_s + qR_{cs} \tag{13}$$

Then calculate the insulation outer surface temperature:

$$t_{i1} = t_a + (t_{f1} - t_a)(R_{a1}/R_1) \tag{14}$$

$$t_{i2} = t_a + (t_{f2} - t_a)(R_{a2}/R_2) \tag{15}$$

The heat flows from each pipe are given by

$$q_1 = (t_{f1} - t_a)/R_1 \tag{16}$$

$$q_2 = (t_{f2} - t_a)/R_2 \tag{17}$$

When the insulation thermal conductivity is a function of its temperature, as is usually the case, an iterative calculation is required, as illustrated in the following example.

Example 5. 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 3. 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 3 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):

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

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

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

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):

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

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

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

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 8) 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

$$\theta_1 = (t_{p2} - t_s)/(t_{p1} - t_s) \tag{18}$$

$$\theta_2 = 1/\theta_1 = (t_{p1} - t_s)/(t_{p2} - t_s) \tag{19}$$

$$P_1 = \frac{1}{2\pi k_s} \ln\left[\frac{(d_1 + d_2)^2 + a^2}{(d_1 - d_2)^2 + a^2}\right]^{0.5} \tag{20}$$

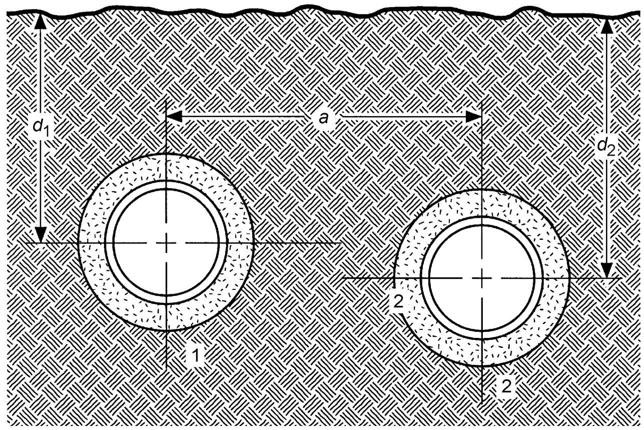


Fig. 8 Two Buried Pipes or Conduits

$$P_2 = \frac{1}{2\pi k_s} \ln\left[\frac{(d_2 + d_1)^2 + a^2}{(d_2 - d_1)^2 + a^2}\right]^{0.5} \tag{21}$$

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

The thermal resistance for each pipe or conduit is given by

$$R_{e1} = \frac{R_{t1} - (P_1^2/R_{t2})}{1 - (P_1\theta_1/R_{t2})} \tag{22}$$

$$R_{e2} = \frac{R_{t2} - (P_2^2/R_{t1})}{1 - (P_2\theta_2/R_{t1})} \tag{23}$$

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_t = total thermal resistance of one pipe/conduit if buried separately, h·ft·°F/Btu

Heat flow from each pipe is then calculated from

$$q_1 = (t_{p1} - t_s)/R_{e1} \tag{24}$$

$$q_2 = (t_{p2} - t_s)/R_{e2} \tag{25}$$

Example 6. 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:

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

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

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

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

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

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

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

Calculate the effective total thermal resistances as

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

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

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 will 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 9) require several assumptions to obtain approximate solutions for the heat transfer. First, assume that the 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 within a tunnel or trench do not cause significant errors for normal operating temperatures when using the following calculation methods.

Unless numerical methods are used, an approximation must be made for the resistance of a rectangular region such as the walls of a trench or tunnel. 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):

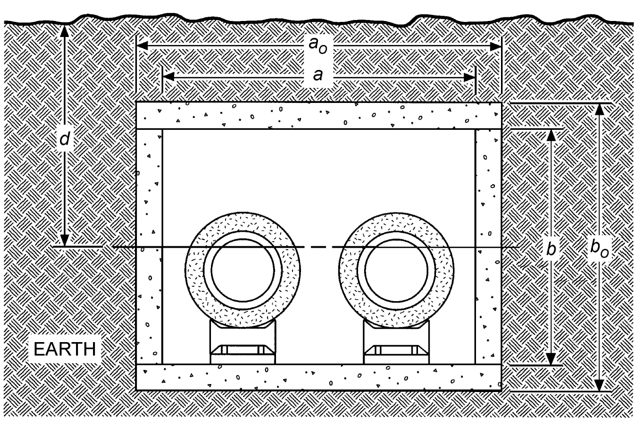


Fig. 9 Pipes in Buried Trenches or Tunnels

$$R_w = \frac{x_w}{2k_w(a+b)} \tag{26}$$

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):

$$R_{ts} = \frac{\ln[3.5d/(b_o^{0.75}a_o^{0.25})]}{k_s[(a_o/2b_o) + 5.7]} \qquad a_o > b_o \tag{27}$$

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. For example, Equation (9) may be used to calculate an approximate resistance for the air space.

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):

$$R_{aw} = 1/[2h_t(a+b)] \tag{28}$$

where

R_{aw} = thermal resistance of air/trench wall interface, h·ft·°F/Btu
h_t = 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:

$$q = \frac{(t_{p1} - t_s)/R_1 + (t_{p2} - t_s)/R_2}{1 + (R_{ss}/R_1) + (R_{ss}/R_2)} \tag{29}$$

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 within the trench/tunnel may be found as

$$t_{ta} = t_s + qR_{ss} \tag{30}$$

where t_{ta} is the air temperature within trench/tunnel.

The individual heat flows for the two pipes within the trench/tunnel are then

$$q_1 = (t_{p1} - t_{ta})/R_1 \tag{31}$$

$$q_2 = (t_{p2} - t_{ta})/R_2 \tag{32}$$

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 7. 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 5.

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 5, or

$$R_1 = 4.61 \text{ h} \cdot \text{ft} \cdot °\text{F/Btu}$$

$$R_2 = 4.26 \text{ h} \cdot \text{ft} \cdot °\text{F/Btu}$$

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

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

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

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

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

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

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):

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

Refined estimates of the pipe insulation surface temperatures are then calculated using Equations (14) and (15):

$$t_{i1} = 84.0 + [(325-84.0)(0.13/4.61)] = 90.8°\text{F}$$

$$t_{i2} = 84.0 + [(225-84.0)(0.15/4.26)] = 89.0°\text{F}$$

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

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

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 (27) and (29) to (32) may be used for shallow trench systems.

Example 8. Consider a shallow trench having the same physical parameters and operating conditions as the buried trench in Example 7, except that the top of the trench is at grade level. Calculate the thermal resistance of the shallow trench using Equation (27):

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

Use thermal resistances for the pipe/insulation systems from Example 7, 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):

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

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

$$t_{i1} = 72.2 + [(325-72.2)(0.13/4.61)] = 79.3°\text{F}$$

$$t_{i2} = 72.2 + [(225-72.2)(0.15/4.26)] = 77.6°\text{F}$$

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 the 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 clay 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 clay 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.

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. The thermal resistances for cylindrical systems can be found from heat transfer coefficients derived using the equations that follow. Generally, the piping systems used for thermal distribution have sufficiently low surface temperatures to preclude any significant heat transfer by thermal radiation.

Example 9. 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 $\varepsilon = 0.26$. 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 3, 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 1995):

$$h_{cv} = 1.016\left(\frac{1}{d}\right)^{0.2}\left(\frac{2}{t_{oi}+t_a}\right)^{0.181}(t_{oi}-t_a)^{0.266}(1+1.277V)^{0.5}$$

$$= 1.016\left(\frac{1}{11.625}\right)^{0.2}\left(\frac{2}{160}\right)^{0.181}(40)^{0.266}(6.11)^{0.5}$$

$$= 1.86\ \text{Btu/h}\cdot\text{ft}^2\cdot{}^\circ\text{F}$$

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 1995):

$$h_{rad} = \varepsilon\sigma\frac{(T_a^4 - T_s^4)}{T_a - T_s}$$

$$h_{rad} = 0.26 \times 1.713 \times 10^{-9}\frac{(560^4 - 520^4)}{560 - 520} = 0.28\ \text{Btu/h·ft}^2\text{·°F}$$

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

$$R_{surf} = \frac{1}{2\pi r_{oi}h_t} = \frac{1}{2\pi \times 0.484 \times 2.14} = 0.15\ \text{h·ft·°F/Btu}$$

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_{oi} = 375 − (92.6 × 3.25) = 74°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 9, 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.

The life-cycle cost of a system 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:

$$\text{LCC} = \text{CC} + (qt_uC_h\text{PWF}) \tag{33}$$

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_h = 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:

$$\text{CRF} = \frac{i(1+i)^n}{(1+i)^n - 1} \tag{34}$$

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 10. Consider a steel conduit system with an air space. The insulation is mineral wool with thermal conductivity as given in Examples 3 and 4. 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_h 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, 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_a, 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_s, Eq. (7), h·ft·°F/Btu	0.46	0.46	0.43	0.43	0.42	0.40
R_t, 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
LCC, Eq. (33), $/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.

EXPANSION PROVISIONS

All piping moves 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, expansion loops are positioned at appropriate locations. If loops are required, additional right-of-way may be required. If field conditions allow, the flexibility of the piping should be used 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 45 covers the design of the pipe bends, loops, and the use of expansion joints. However, the chapter uses conservative stress values. Computer-aided 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 when the pipe will provide the required flexibility. For example, Table 11 in Chapter 45 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 computer-aided program recommends a 13 ft wide by 26 ft high loop; and 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 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, in the direction 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. However, the manufacturer of the expansion joint should be contacted 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 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. The manufacturer of the expansion device should be consulted 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. In addition, these surface finishes do not corrode or increase sliding resistance in aboveground installations. Guides should be selected to handle twice the expected movement, so they may be installed in a neutral position without the need for cold-springing the pipe.

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 distribution systems. The materials needed to ensure long life and low heat loss further classify DHC systems into low-temperature, medium-temperature, and high-temperature systems. The temperature range for medium-temperature systems is usually too high for the materials that are used in low-temperature systems; however, the same materials that are 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 selection of efficient insulation materials and inexpensive pipe materials that resist corrosion is 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 materials that are readily available. 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 the walk-through tunnel (Figure 10), the concrete surface trench (Figure 11), the deep-burial small tunnel (Figure 12), and underground systems that use poured insulation (Figure 13) or cellular glass (Figure 14) 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, may be the only system that can be used in flooded sites. 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. 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 can be high enough to reduce the strength of plastic pipe to an unacceptable level. Rigid, extruded polystyrene insulation 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. A numerical analysis of the thermal problem may be required to ensure that the desired effect is achieved.

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 liability issues are addressed. In laying out the main supply and return piping, redundancy of supply and return should be considered. If a looped system is used to provide redundancy, flow rates under all possible failure modes must be addressed when sizing and laying out the piping.

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. Special attention must be given to the safety of personnel who come in contact with distributions systems or who must enter spaces occupied by underground systems. The regulatory authority's definition of a confined space and the possibility of exposure to high-temperature or high-pressure piping can have a significant impact on the access design, which must be addressed by the designer. Gravity venting of tunnels is good practice, and access ports and tunnels 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 A53 seamless or ERW, Grade B; or ASTM A106 seamless, Grade B. Care should be taken to exclude ASTM A53, Type F, because of its lower allowable stress and because of the method by which 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 be considered when choosing between a high-temperature and a low-temperature system. Welding pipes with thinner walls requires extra care and may require additional inspection. Also, extra care must be taken 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 corrosive nature 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. Fiberglass-reinforced plastic (FRP) or glass-reinforced plastic (GRP) pipe used for condensate return has not performed well. Failed steam traps, pipe resin solubility, deterioration at elevated temperatures, and thermal expansion are thought to be the cause of premature failure.

Chilled-Water Distribution. For chilled-water systems, a variety of pipe materials, such as steel, ductile iron, 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. The soil temperature is usually highest when the chilled-water loads are highest; therefore, it is usually life-cycle cost-effective to insulate the chilled-water pipe. A life-cycle cost analysis often favors a factory prefabricated product that is insulated with a plastic foam with a waterproof casing or a field-fabricated system that is insulated with ASTM C552 cellular glass insulation. If a plastic product is selected, care must be taken 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. 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 may be used to insulate adjacent chilled-water lines from the impacts 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, when transitioning from ductile or plastic piping to steel at the buildings, flanged connections are usually best, but should be located inside the building and not buried. Proper gasket selection and bolt torque are also critical.

Aboveground Systems

An aboveground system consists of a distribution pipe, insulation that surrounds the pipe, and a protective jacket that surrounds the insulation. The jacket may have an integral vapor retarder. 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 storm water out of the insulation. The jacket material can be aluminum, stainless steel, galvanized steel, or plastic sheet; a multilayered fabric and organic cement composite; or a combination of these. Plastics and organic cements exposed to sunshine must be ultraviolet-light resistant.

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 rest 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. It requires an order of magnitude more design effort and construction inspection accuracy when compared to gas, water, and sanitary 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, cellular glass, and conduit system.

Walk-Through Tunnel. This system (Figure 10) consists of a field-erected tunnel large enough for a someone 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.

Shallow Concrete Surface Trench. This system (Figure 11) 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 to drain. This system is designed to handle the storm water and groundwater that enters the system, so the floor is always sloped toward a drainage point. A drainage system is required at all floor low points.

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.

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 storm water that enters at the top cover butt joints. Small inspection ports of about 12 in. diameter

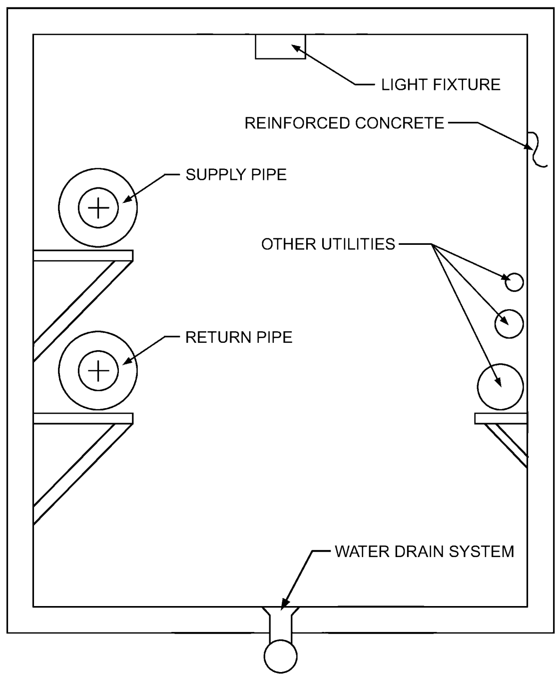


Fig. 10 Walk-Through Tunnel

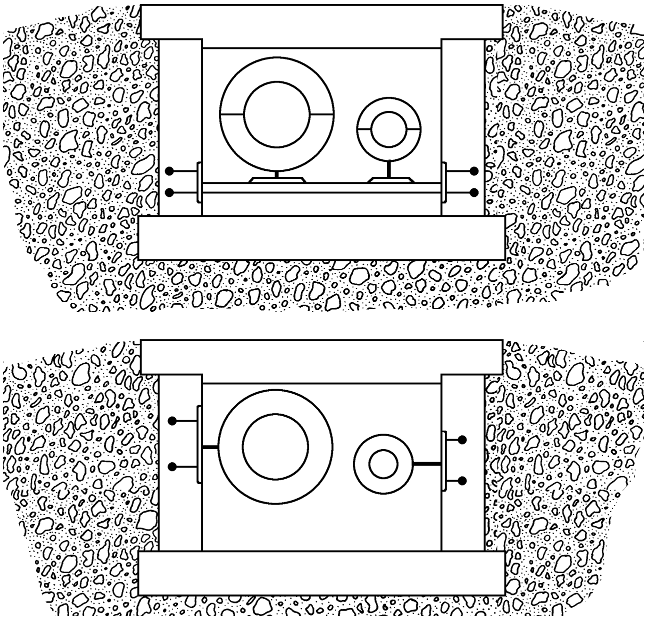


Fig. 11 Concrete Surface Trench

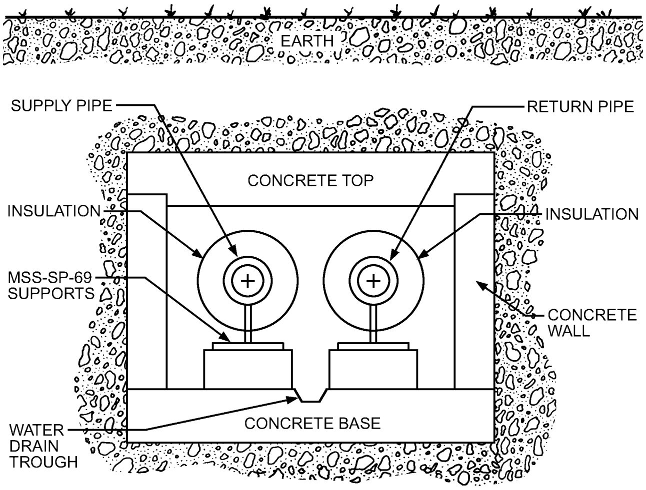


Fig. 12 Deep-Bury Small Tunnel

may be cast into the top covers at key locations so that 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 12) 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.

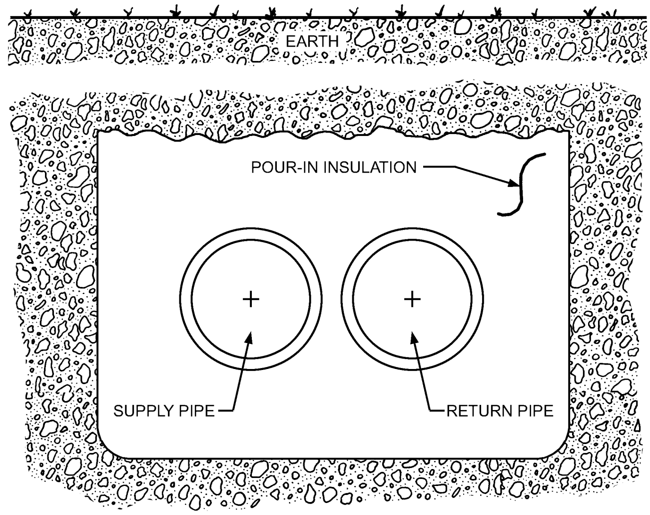


Fig. 13 Poured Insulation System

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 storm water 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, the distribution pipes, and the 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 the entry of groundwater. After the covers are installed, the system is covered with earth to match the existing topography.

Poured Insulation. This system (Figure 13) 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 below the 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 flooded condition.

The insulation material serves several functions. It may support the distribution pipes, 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. Special attention must be given 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 thermal resistance. A groundwater drainage system may be required, depending on the insulation material selected and the severity of the groundwater; however, if such a 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 that is treated to be water repellent, have been used successfully. The hydrophobic characteristic of this powder prevents water from dampening the powder and has some capability 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.

Field-Installed, Direct-Buried Cellular Glass. In this system (Figure 14), cellular glass 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 movement of the pipes in the expansion loops. As shown in Figure 14, a drain should be installed to drain groundwater away. A waterproof jacket is recommended for all buried applications.

When used for heating applications, the dry soil condition must be investigated to determine if the temperature of the jacket exceeds the material 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 will be allowed to be without exceeding the temperature limits of the jacket material. The thickness of insulation 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 15). 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**.

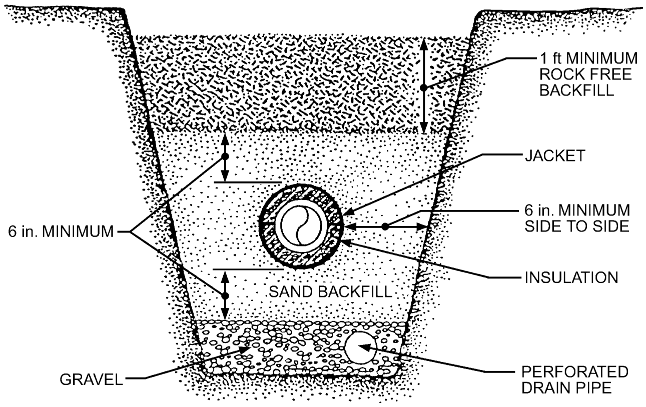


Fig. 14 Field-Installed, Direct-Buried Cellular Glass Insulated System

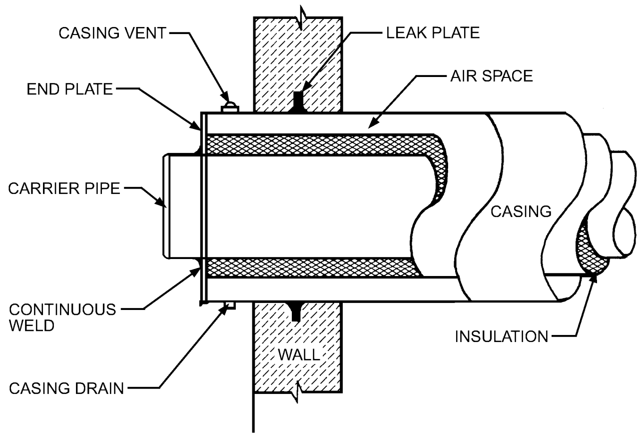


Fig. 15 Conduit System Components

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.

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 is 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. For these reasons, it is imperative that the project design engineer or architect clearly delineate the responsibilities of the factory prefabricator.

Crushing 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

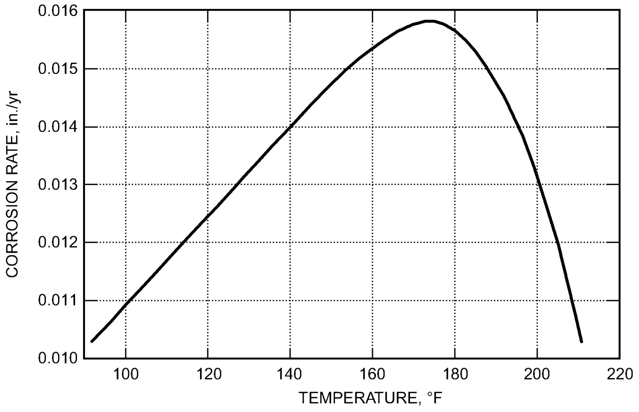


Fig. 16 Corrosion Rate in Aggressive Environment Similar to Mild Steel Casings in Soil

that encompasses the maximum corrosion rate of steel (Figure 16). As shown in the figure, the steel casing of a district heating pipe experiences corrosion rates several times that of domestic water pipes. The temperature of the casing 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. In the event that water enters one section, a WSL system prevents its spread to adjacent sections of piping.

The air space conduit system (Figures 17 and 18) 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 has been using a Federal Agency Committee 96 h boiling test for conduit insulation. 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.

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.

The WSL system (Figure 19) 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 European *Standard* EN 253 with regard to all major construction and design details. Standards also have been established for fittings (EN 448), preinsulated valves (EN 488), and the field joint assemblies (EN 489). With this system, the carrier pipe, insulation, and casing are bonded together to form a single unit. Forces caused by thermal expansion are

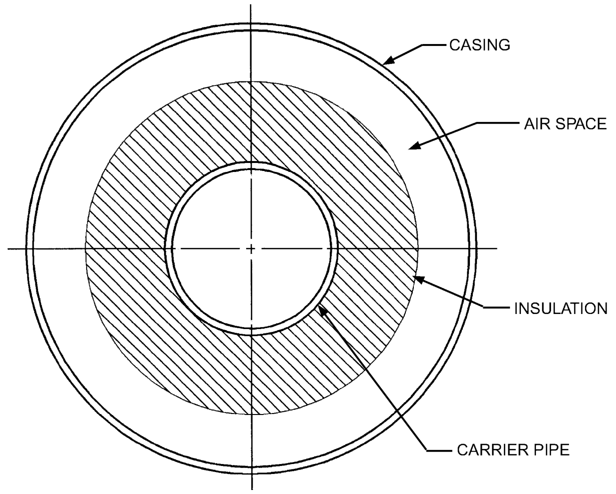


Fig. 17 Conduit System with Annular Air Space and Single Carrier Pipe

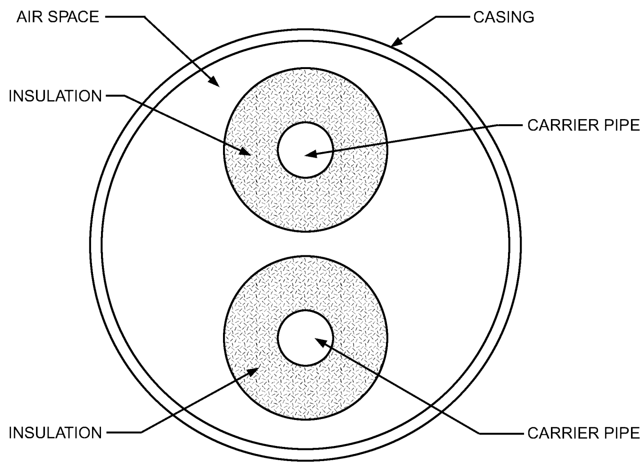


Fig. 18 Conduit System with Two Carrier Pipes and Annular Air Space

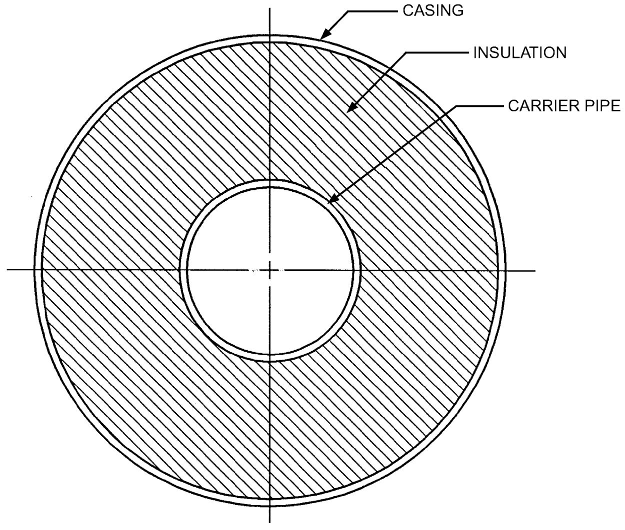


Fig. 19 Conduit System with Single Carrier Pipe and No Air Space (WSL)

passed as shear forces to the mating component and ultimately to the soil; thus, no additional expansion provisions are required.

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 is acting 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 minimum during this condition; however, if the casing is not restrained, its movement with respect to the soil will be 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.

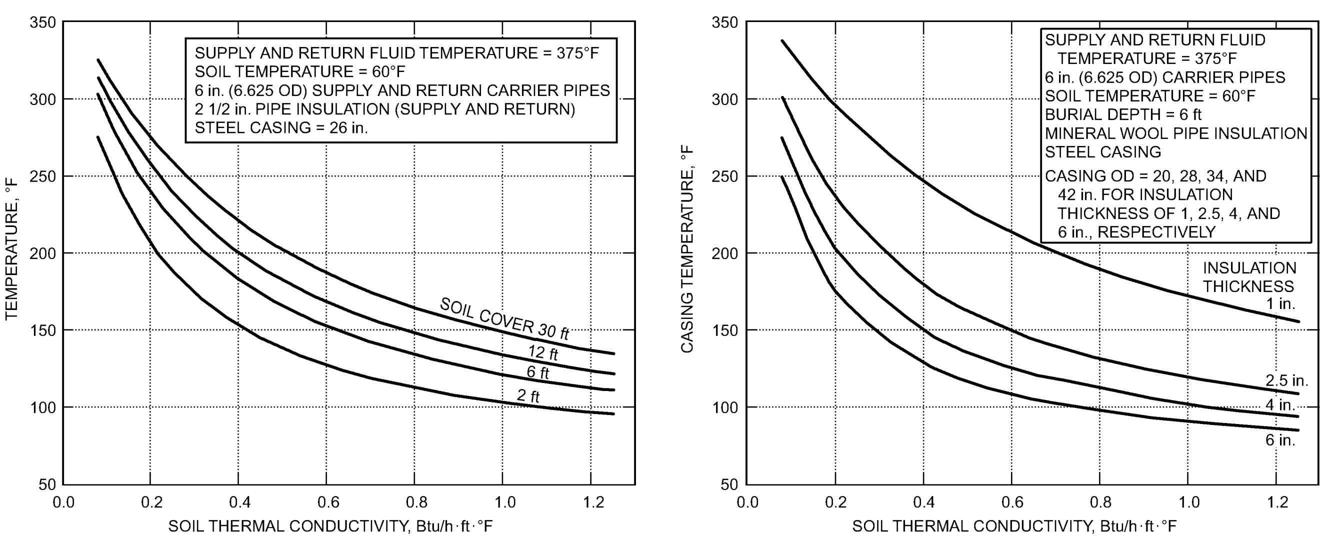
  Figure 20A shows the effect of burial depth on casing temperature as a function of soil thermal conductivity for a typical system. Figure 20B shows the effect of insulation thickness on casing temperature, again as a function of soil thermal conductivity. Analysis of Figure 20A and 20B suggests some design solutions that could lower the effects of the dry-soil condition. Possible solutions are to lower the carrier pipe temperature, use thicker carrier pipe insulation, provide a device to keep the soil wet, or minimize the 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 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 3 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 3 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.

Cathodic protection is a standard method used by the underground pipeline industry to further protect coated steel against



A. EFFECT OF BURIAL DEPTH ON CASING TEMPERATURE B. EFFECT OF INSULATION THICKNESS ON CASING TEMPERATURE

Fig. 20 Conduit Casing Temperature Versus Soil Thermal Conductivity

corrosion. Cathodic protection systems are routinely designed for a minimum life of 20 years. Cathodic protection may be achieved by the sacrificial anode method or the impressed current method.

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

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

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

A cathodically protected system must be electrically isolated at all points where the pipe is connected to building or access port piping and where a new system is connected to an existing system. Conduits are generally tied to another building or access piping with flanged connections. Flange isolation kits, including dielectric gaskets, washers, and bolt sleeves, electrically isolate the cathodically protected structure. If an isolation flange is not used, any connecting piping or metallic structure will be in the protection system, *but protection may not be adequate*.

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

NACE *Standard* RPO169 has further information on control of external corrosion on buried metallic structures.

Leak Detection

The conduit may require excavation to repair construction errors after burial. Various techniques are available for detecting leaks in district heating and cooling piping. They range from performing periodic pressure tests on the piping system to installing a sensor cable along the entire length of the piping to continuously detect and locate leaks. Pressure testing should be performed on all piping to verify integrity during installation and the life of the piping. Chilled-water systems should be pressure tested during the winter, and hot-water and steam systems tested during the summer.

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

Chilled- and Hot-Water Systems. Chilled-water piping systems are usually insulated with urethane foam with a vaporproof jacket (HDPE, urethane, PVC, CPVC, etc.). Copper wires can be installed during fabrication to aid in detecting and locating liquid leaks. The wires may be insulated or uninsulated, depending on the manufacturer. Some systems monitor the entire wire length, whereas others only monitor at the joints of the piping system. The detectors either look for a short in the circuit using Ohm's law or monitor for impedance change using time domain reflectometry (TDR).

Steam, High-Temperature Hot-Water, and Other Conduit Systems. Air gap designs, which have a gap between the inner wall of the outer casing and the insulation, can have probes installed at the low points of drains or at various points to detect leaks. Leaks can also be detected with a continuous cable that monitors liquid leakage. The cable is installed at the bottom of the conduit with a minimum air gap required, typically 1 in. Pull points or access ports are installed every 400 to 500 ft on straight runs, with changes in direction reducing the length between pull points. Systems monitor either by looking for a short on the cable using Ohm's law or by sensing the impedance on a coaxial cable using resistance temperature devices (RTD). During installation, care must be taken to keep the system clean and dry to keep any contamination from the leak detection system that might cause it to fail. The system must be sealed airtight to prevent condensation from accumulating in the piping at the low points.

Valve Vaults and Entry Pits

Valve vaults allow a user to isolate problems to one area rather than analyze the entire line. This feature is important if the underground distribution system cannot be maintained between valve vaults without excavation. The optimum number of valve vaults is that which affords the lowest life-cycle cost and still meets all design requirements, usually no more than 500 ft apart. Valve vaults provide a space in which to put valves, steam traps, carrier pipe drains, carrier pipe vents, casing vents and drains, condensate pumping units, condensate cooling devices, flash tanks, expansion joints, groundwater drains, electrical leak detection equipment and wiring, electrical isolation couplings, branch line isolation valves, carrier pipe isolation valves, and flowmeters. Valve vaults allow for elevation changes in the distribution system piping while maintaining an acceptable slope on the system; they also allow the designer to better match the topography and avoid unreasonable and expensive burial depths.

Ponding Water. The most significant problem with valve vaults is that water ponds in them. Ponding water may be from either carrier pipe leaks or intrusion of surface or groundwater into the valve vault. When the hot- and chilled-water distribution systems share the same valve vault, plastic chilled-water lines often fail because ponded water heats the plastic to failure. Water gathers in the valve vault irrespective of climate; therefore, design strives to eliminate the water for the entire life of the underground distribution system. The most successful water removal systems are those that drain to sanitary or storm drainage systems; this technique is successful because the system is affected very little by corrosion and has no moving parts to fail. Backwater valves are recommended in case the drainage system backs up.

Duplex sump pumps with lead-lag controllers and a failure annunciation system are used when storm drains and sanitary drains are not accessible. Because pumps have a history of frequent failures, duplex pumps help eliminate short cycling and provide standby pumping capacity. Steam ejector pumps can be used only if the distribution system is never shut down because the carrier pipe insulation can be severely damaged during even short outages. A labeled, lockable, dedicated electrical service should be used for electric pumps. The circuit label should indicate what the circuit is used for; it should also warn of the damage that will occur if the circuit is deenergized.

Electrical components have experienced accelerated corrosion in the high heat and humidity of closed, unventilated vaults. A pump that works well at 50°F often performs poorly at 200°F and 100% rh. To resolve this problem, one approach specifies components that have demonstrated high reliability at 200°F and 100% rh with a damp-proof electrical service. The pump should have a corrosion-resistant shaft (when immersed in water) and impeller and have demonstrated 200,000 cycles of successful operation, including the electrical switching components, at the referenced temperature and humidity. The pump must also pass foreign matter; therefore, the requirement to pass a 3/8 in. ball should be specified.

Another method drains the valve vaults into a separate sump adjacent to the vault. Then the pumps are placed in this sump, which is cool and more nearly a sump pump environment. Redundant methods may be necessary if maximum reliability is needed or future maintenance is questionable. The pump can discharge to the sanitary or storm drain or to a splash block near the valve vault. Water pumped to a splash block has a tendency to enter the vault, but this is not a significant problem if the vault construction joints have been sealed properly. Extreme caution must be exercised if the bottom of the valve vault has French drains. These drains allow groundwater to enter the vault and flood the insulation on the distribution system during high-groundwater conditions. Adequate ventilation of the valve vault is also important.

Crowding of Components. The valve vault must be laid out in three dimensions, considering standing room for the worker, wrench swings, the size of valve operators, variation between manufacturers in the size of appurtenances, and all other variations that the specifications allow with respect to any item placed in the vault. To achieve desired results, the vault layout must be shown to scale on the contract drawings.

High Humidity. High humidity develops in a valve vault when it has no positive ventilation. Gravity ventilation is often provided in which cool air enters the valve vault and sinks to the bottom. At the bottom of the vault, the air warms, becomes lighter, and rises to the top of the vault, where it exits. In the past, some designers used a closed-top valve vault with an exterior ventilation pipe with an elbow that directs the exiting air down. However, the elbowed-down vent hood tends to trap the exiting air and prevent gravity ventilation from working. Open structural grate tops are the most successful covers for ventilation purposes. Open grates allow rain to enter the vault; however, the techniques mentioned previously in the section on Ponding Water are sufficient to handle the rainwater. Open grates with sump basins have worked well in extremely cold climates and in warm climates. Some vaults have a closed top and screened, elevated sides to allow free ventilation. In this design, the solid vault sides extend slightly above grade; then, a screened window is placed in the wall on at least two sides. The overall above-grade height may be only 18 in.

High Temperatures. The temperature in the valve vault rises when no systematic way is provided to remove heat losses from the distribution system. The gravity ventilation rate is usually not sufficient to transport heat from the closed vaults. Part of this heat transfers to the earth; however, an equilibrium temperature is reached that may be higher than desired. Ventilation techniques discussed in the section on High Humidity can resolve the problem of high temperature if the heat loss from the distribution system is near normal. Typical problems that greatly increase the amount of heat released include

- Leaks from a carrier pipe, gaskets, packings, or appurtenances
- Insulation that has deteriorated because of flooding or abuse
- Standing water in a vault that touches the distribution pipe
- Steam vented to the vault from partial flooding between valve vaults
- Vents from flash tanks
- Insulation removed during routine maintenance and not replaced

To prevent heat release in a new system, a workable ventilation system must be designed. On existing valve vaults, the valve vault must be ventilated properly, all leaks corrected, and all insulation that was damaged or left off replaced. Commercially available insulation jackets that can easily be removed and reinstalled from fittings and valves should be installed. If flooding occurs between valve vaults, portions of the distribution system may have to be excavated and repaired or replaced. Vents from vault appurtenances that exhaust steam into the vault may have to be routed aboveground if the ventilation technique is insufficient to handle the quantity of steam exhausted.

Deep Burial. When a valve vault is buried too deeply, (1) the structure is exposed to groundwater pressures, (2) entry and exit often become a safety problem, (3) construction becomes more difficult, and (4) the cost of the vault is greatly increased. Valve vault spacing should be no more than 500 ft (NAS 1975). If greater spacing is desired, use an accurate life-cycle analysis to determine spacing. The most common way to limit burial depth is to place the valve vaults closer together. Steps in the distribution system slope are made in the valve vault (i.e., the carrier pipes come into the valve vault at one elevation and leave at a different elevation). If the slope of the distribution system is changed to more nearly match the earth topography, the valve vaults will be shallower; however, the allowable range of slope of the carrier pipes restricts this method. In most systems, the slope of the distribution system can be reversed in a valve vault, but not out in the system between valve vaults. The minimum slope for the carrier pipes is 1 in. in 20 ft. Lower slopes are outside the range of normal construction tolerance. If the entire distribution system is buried too deeply, the designer must determine the maximum allowable burial depth of the system and survey the topography of the distribution system to determine where the maximum and minimum depth of burial will occur. All elevations must be adjusted to limit the minimum and maximum allowable burial depths.

Freezing Conditions. Failure of distribution systems caused by water freezing in components is common. The designer must consider the coldest temperature that may occur at a site and not the 99% or 99.6% condition used in building design (as discussed in Chapter 28 of the 2005 *ASHRAE Handbook—Fundamentals*). Drain legs or vent legs that allow water to stagnate are usually the cause of failure. Insulation should be on all items that can freeze, and it must be kept in good condition. Electrical heat tape and pipe-type heat tracing can be used under insulation. If part of a chilled-water system is in a ventilated valve vault, the chilled water may have to be circulated or be drained if not used in winter.

Safety and Access. Some of the working fluids used in underground distribution systems can cause severe injury and death if accidentally released in a confined space such as a valve vault. The shallow valve vault with large openings is desirable because it allows personnel to escape quickly in an emergency. The layout of the pipes and appurtenances must allow easy access for maintenance without requiring maintenance personnel to crawl underneath or between other pipes. The goal of the designer is to keep clear work spaces for maintenance personnel so that they can work efficiently and, if necessary, exit quickly. Engineering drawings must show pipe insulation thickness; otherwise they will give a false impression of the available space.

The location and type of ladder is important for safety and ease of egress. It is best to lay out the ladder and access openings when laying out the valve vault pipes and appurtenances as a method of exercising control over safety and ease of access. Ladder steps, when cast in the concrete vault walls, may corrode if not constructed of the correct material. Corrosion is most common in steel rungs. Either cast-iron or prefabricated, OSHA-approved, galvanized steel ladders that sit on the valve vault floor and are anchored near the top to hold it into position are best. If the design uses lockable access doors, the locks must be operable from inside or have some keyed-open device that allows workers to keep the key while working in the valve vault.

Vault Construction. The most successful valve vaults are those constructed of cast-in-place reinforced concrete. These vaults conform to the earth excavation profile and show little movement when backfilled properly. Leakproof connections can be made with mating tunnels and conduit casings even though they may enter or leave at oblique angles. In contrast, prefabricated valve vaults may settle and move after construction is complete. Penetrations for prefabricated vaults, as well as the angles of entry and exit, are difficult to locate exactly. As a result, much of the work associated with penetrations is not detailed and must be done by construction workers in the field, which greatly lowers the quality and greatly increases the chances of a groundwater leak.

Construction deficiencies that go unnoticed in the buildings can destroy a heating and cooling distribution system; therefore, the designer must clearly convey to the contractor that a valve vault does not behave like a sanitary access port. A design that is sufficient for a sanitary access port will prematurely fail if used for a heating and cooling distribution system.

CONSUMER INTERCONNECTIONS

The thermal energy produced at the plant is transported via the distribution network and is finally transferred to the consumer. When thermal energy (hot water, steam, or chilled water) is supplied, it may be used directly by the building HVAC system or process loads, or indirectly via a heat exchanger that transfers energy from one media to another. When energy is used directly, it may be reduced in pressure that is commensurate to the buildings' systems. The design engineer must perform an analysis to determine which connection type is best.

For commercially operated systems, a contract boundary or point of delivery divides responsibilities between the energy provider and the customer. This point can be at a piece of equipment, as in a heat exchanger with an indirect connection, or flanges as in a direct connection. A chemical treatment analysis must be performed (regardless of the type of connection) to determine the compatibility of each side of the system (district and consumer) before energizing.

Direct Connection

Because a direct connection offers no barrier between the district water and the building's own system (e.g., air-handling unit cooling and heating coils, fan-coils, radiators, unit heaters, and process loads), water circulated at the district plant has the same quality as the customer's water. Direct connections, therefore, are at a greater risk of incurring damage or contamination based on the poor water quality of either party. Typically, district systems have contracts with water treatment vendors and monitor water quality continuously. This may not be the case with all consumers. A direct connection is often more economical than an indirect connection because the consumer is not burdened by the installation of heat exchangers, additional circulation pumps, or water treatment systems; therefore, investment costs are reduced and return temperatures identical to design values are possible.

Figure 21 shows the simplest form of direct connection, which includes a pressure differential regulator, a thermostatic control valve on each terminal unit, a pressure relief valve, and a check valve. Most commercial systems have a flowmeter installed as well as temperature sensors and transmitters to calculate the energy used. The location of each device may vary from system to system, but all of the major components are indicated. The control valve is the capacity regulating device that restricts flow to maintain either a chilled-water supply or return temperature on the consumer's side.

Particular attention must be paid to connecting high-rise buildings because they induce a static head. Pressure control devices should be investigated carefully. It is not unusual to have a water-based district heating or cooling system with a mixture of direct and indirect connections in which heat exchangers isolate the systems hydraulically.

In a direct system, the pressure in the main distribution system must meet local building codes to protect the customer's installation and the reliability of the district system. To minimize noise, cavitation, and control problems, constant-pressure differential control valves 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.

If the temperature in the main distribution system is lower than that required in the consumer cooling systems, a larger temperature differential between supply and return occurs, thus reducing the required pipe size. The consumer's desired supply temperature can be attained by mixing the return water with the district cooling supply water. Depending on the size and design of the main system, elevation differences, and types of customers and building systems, additional safety equipment, such as automatic shutoff valves on both supply and return lines, may be required.

When buildings have separate circulation pumps, primary/secondary piping, and pumping, isolating techniques are used (cross-connection between return and supply piping, decouplers, and bypass lines). This ensures that two-way control valves are subjected only to the differential pressure established by the customer's

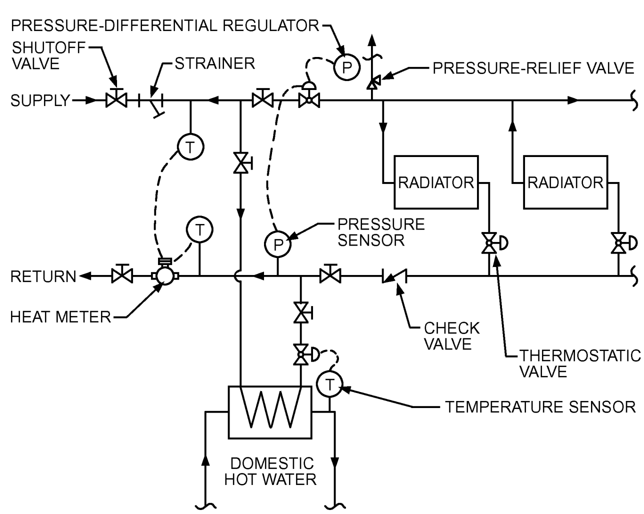


Fig. 21 Direct Connection of Building System to District Hot Water

building (tertiary) pump. Figure 4 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.

Indirect Connection

Many of the components are similar to those used in the direct connection applications, with the exception that a heat exchanger performs one or more of the following functions: heat transfer, pressure interception, and buffer between potentially different quality water treatment.

Identical to the direct connection, the rate of energy extraction in the heat exchanger is governed by a control valve that reacts to the building load demand. Once again, the control valve usually modulates to maintain a temperature set point on either side of the heat exchanger, depending on the contractual agreement between the consumer and the producer.

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 disadvantages of using an indirect connection are the (1) additional cost of the heat exchanger and (2) temperature loss and increased pumping pressure because of the addition of another heat transfer surface.

COMPONENTS

Heat Exchangers

Heat exchangers act as the line of demarcation between ownership responsibility 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 22 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 turndown as close as possible (oversized units may not perform as desired at maximum turndown; therefore, several smaller units will optimize 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 the allowable pressure drop on both sides of heat exchangers. The customer's side is usually the most critical for pressure drop. The higher the pressure drop, the smaller and less expensive the heat exchanger. However, the pressure drop must be kept in reasonable limits (15 psig or below) if the existing pumps are to be reused. Investigate the existing chiller evaporator pressure drop in order to assist in this evaluation.

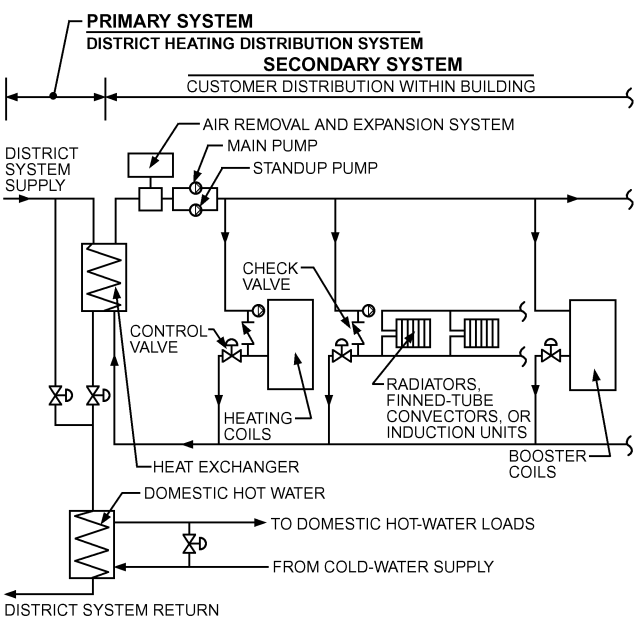


Fig. 22 Basic Heating-System Schematic

All heat exchangers should be sized with future expansion in mind. When selecting heat exchangers, be cognizant that closer approach temperatures or low pressure drop require more heat transfer area and hence cost more and take up more space. Strainers should be installed in front of any heat exchanger and control valve to keep debris from fouling surfaces.

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.

Plate Heat Exchangers (PHEs). These exchangers, which are used for either steam, hot-water, or chilled-water applications, are available as gasketed units and in two gasket-free designs (brazed and all or semiwelded construction). All PHEs consist of metal plates compressed between two end frames and sealed along the edges. Alternate plates are inverted and the gaps between the plates form the liquid flow channels. Fluids never mix as 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 plates and the other in the even-numbered plates.

Because PHEs require turbulent flow for good 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 tends to keep the surfaces clean. PHEs generally have a cost advantage and require one-third to one-half the surface required by shell-and-tube units for the same operating conditions. PHEs are also capable of closer approach temperatures.

Gasketed PHEs (also called plate-and-frame heat exchangers) consist of a number of gasketed embossed metal plates bolted together between two end frames. Gaskets are placed between the plates to contain the two media in the plates and to act as a boundary. Gasket failure will not cause the two media to mix; instead, the media leak to the atmosphere. Gaskets can be either glued or clip-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.

PHEs are typically used for district heating and cooling with water and for cooling tower 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 are capable of achieving 1°F approach. This type of PHE can be disassembled in the field to clean the plates and replace the gaskets. Typical applications go up to 365°F and 400 psig.

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 frames brazed together with copper or nickel. Tightening bolts are not required as in the gasketed design. These units cannot be disassembled and cleaned; therefore, adequate strainers must be installed ahead of an exchanger and it must be periodically flushed clean in a normal maintenance program. Brazed PHEs usually peak at a capacity of under 200,000 Btu/h (about 200 plates and 600 gpm) and 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. Applications where the PHE may be exposed to large, sudden, or frequent changes in temperature and load must be avoided 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 a variety of alloys are available. Models offered 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/crossflow 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. It 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.

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 overstating customer loads complicates the selection process, so accurate load information is important. It is also important that the valve selected operates under the extreme pressure and flow ranges foreseen. Because most commercial-grade valves will not perform well for this installation, industrial-quality valves are 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 fail 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 reduces valve life and causes valve hunting. Select control valves having a wide range of control; low leakage; and proportional-plus-integral control for close adjustment, balancing, temperature accuracy, and response time. Control valves should have actuators with enough force to open and close under the maximum pressure differential in the system. The control valve should have a pressure drop through the valve equal to at least 10 to 30% of the static pressure drop of the distribution system. This pressure drop gives the control valve the "authority" it requires to properly control flow. The relationship between valve travel and capacity output should be linear, with an equal percentage characteristic.

In 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 also installed the return line.

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).

Temperature sensors need to be located by the exchangers being controlled rather than in the common pipe. Improperly located sensors will cause one control valve to open and others to close, resulting in unequal loads in the exchangers.

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 parameters outside normal, and monitoring and control of all components.

Typical control strategies include regulating district flow to maintain the customer's supply temperature (which results in a fluctuating customer return temperature) or maintaining the customer's return temperature (which results in a fluctuating customer supply temperature). When controlling return flow 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 overpressurization will occur.

In high-rise buildings, all piping, valves, coils, and other equipment may be required to withstand higher design pressures. Where system static pressure exceeds safe or economical operating pressure, either the heat exchanger method or pressure-sustaining valves in the return line may be used to minimize the impact of the pressure. Vacuum vents should be provided at the top of the building's water risers to introduce air into the piping in case the vertical water column collapses.

HEATING CONNECTIONS

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 both cooled and discharged to the building sewage system or returned back to the central plant for recycling. 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 to the building drains. This process extracts the maximum amount of energy out of the delivered steam.

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 23 and 24. Chapter 10 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.

Other components of the steam connection may include condensate pumps, flowmeters (steam and/or condensate), and condensate conductivity probes, which may dump condensate if they are contaminated by unacceptable debris. Many times, 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. The use of 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)

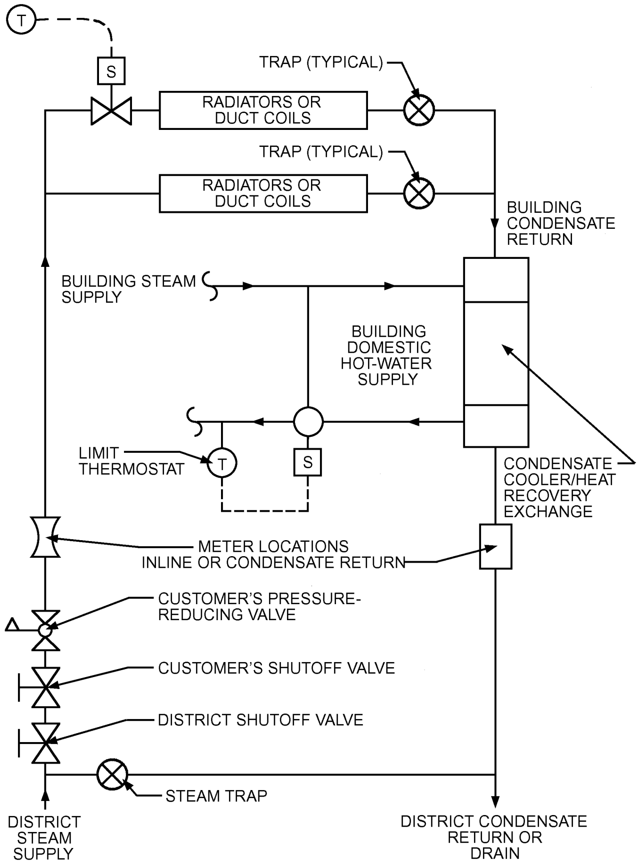


Fig. 23 District/Building Interconnection with Heat Recovery Steam System

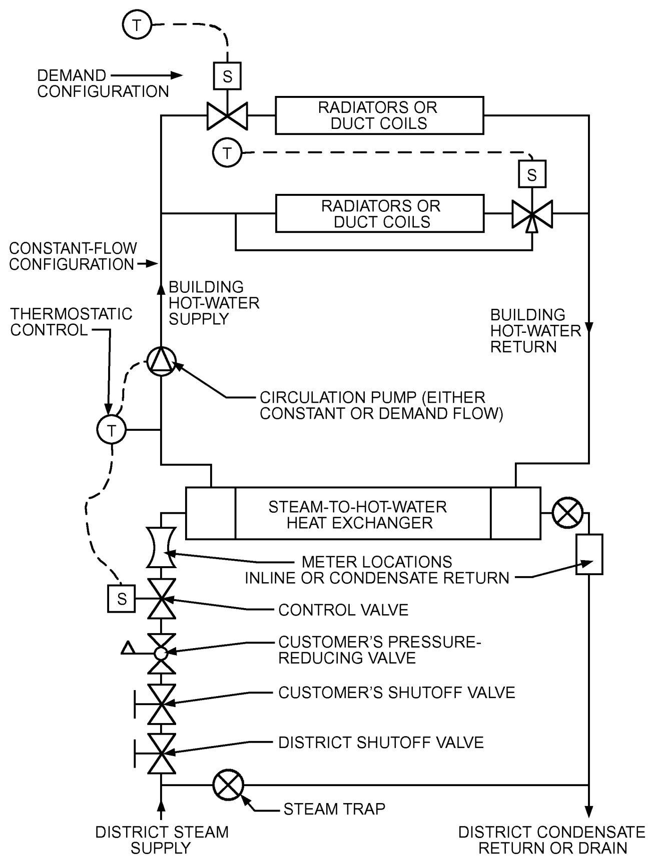


Fig. 24 District/Building Interconnection with Heat Exchange Steam System

- 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

Each level of steam pressure reduction should also be monitored as well as the temperature of the condensate. Where conductivity probes are used to monitor the quality of the 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 a conductivity problem exists at a customer.

Hot-Water Connections

Figure 25 illustrates a typical indirect connection using a heat exchanger between the district hot-water system and the customer's system. It shows the radiator configurations typically used in both constant- and variable-flow systems. Figure 26 shows a typical direct connection between the district hot-water system and the building. It includes the typical configurations for both demand flow and constant-flow systems and the additional check valve and piping required for the constant-flow system. Figure 27 shows an indirect connection for both space and hot-water heating.

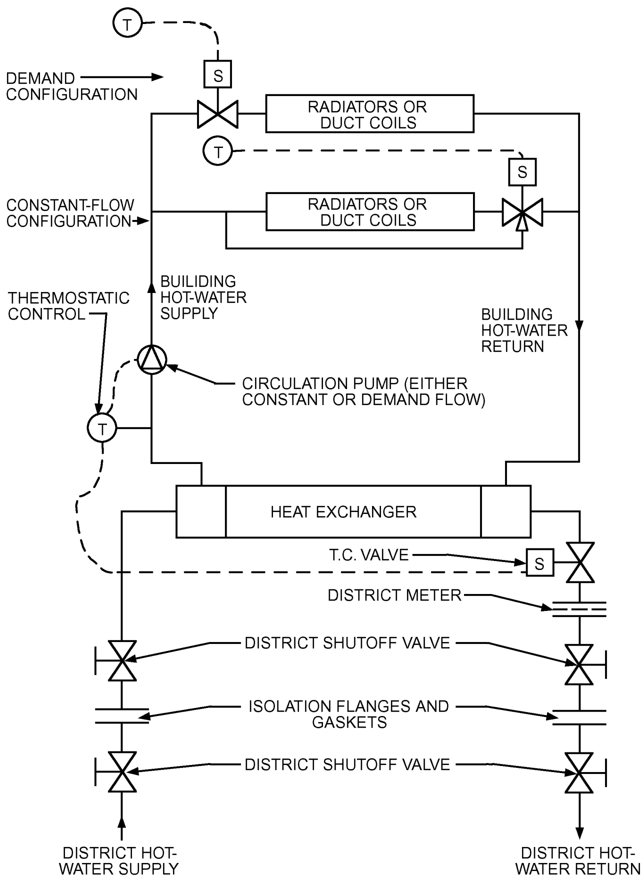


Fig. 25 District/Building Indirect Interconnection Hot-Water System

Lines on these systems must be sized using the same design used for the main feed lines of an in-building power plant. In general, demand flow systems permit better energy transfer efficiency and smaller line size for a given energy transfer requirement. Line sizing should account for any future loads on the building, etc. To keep return temperature low, water flow through heating equipment should be controlled according to the heating demand in the space.

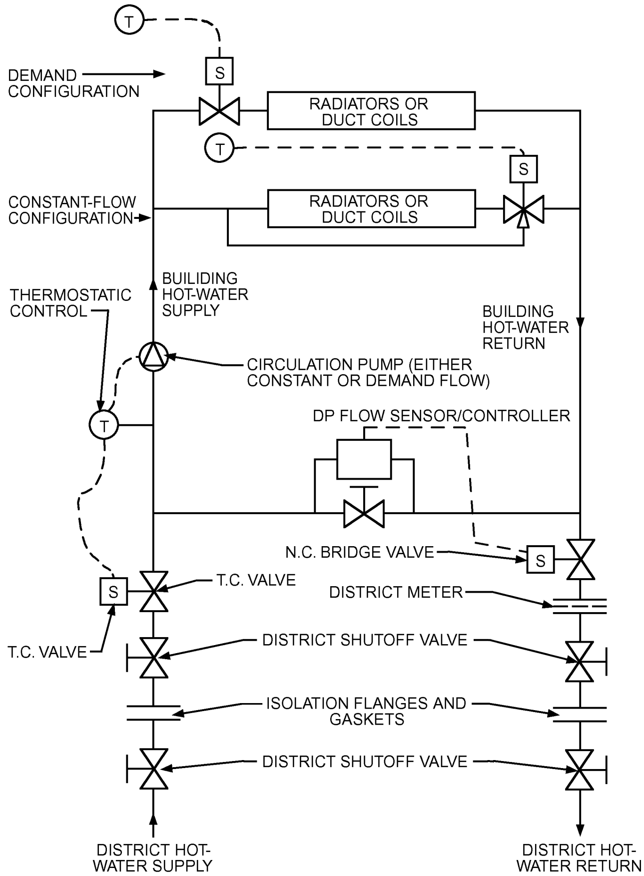


Fig. 26 District/Building Direct Interconnection Hot-Water System

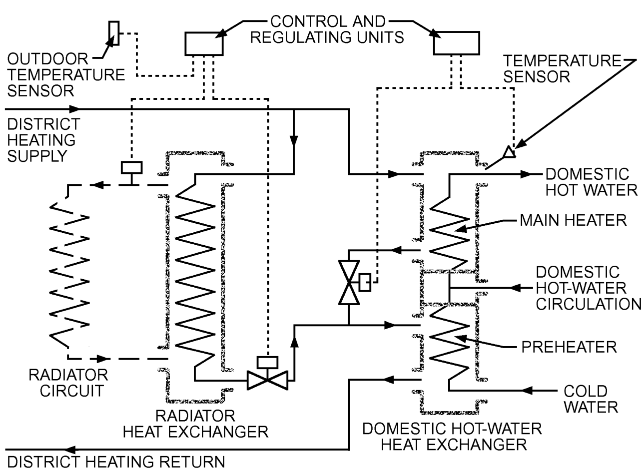


Fig. 27 Building Indirect Connection for Both Heating and Domestic Hot Water

The secondary supply temperature must be controlled in a manner similar to the primary supply temperature. To ensure a low primary return temperature and large temperature difference, the secondary system should have a low design return temperature. This design helps reduce costs through smaller pipes, pumps, amounts of insulation, and pump motors. A combined return temperature on the secondary side of 130°F or lower is reasonable, although this temperature may be difficult to achieve in smaller buildings with only baseboard or radiator space heating.

A hot-water district heating distribution system has many advantages over a steam system. Because of the efficiency of the relatively low temperature, energy can be saved over an equivalent steam system. Low- and medium-temperature hot-water district heating allows a greater flexibility in the heat source, lower-cost piping materials, and more cost-effective ways to compensate for expansion and new customer connections.

Major advantages of a hot-water distribution system include the following:

- Changing both temperature and flow can vary the delivered thermal power.
- Hot water can be pumped over a greater distance with minimal energy loss. This is a major disadvantage of a steam system.
- Operates at lower supply temperatures over the year. This results in lower line losses. Steam systems operate at constant pressure and temperature year round, which increases energy production and system losses while decreasing annual system efficiency.
- Has much lower operating and maintenance costs than steam systems (leaks at steam traps, chemical treatment, spares, etc.).
- Uses prefabricated insulated piping, which has a lower initial capital cost as compared to steam systems.

Building Conversion to District Heating

Table 4 (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 can be seen, the probability is high for water-based systems, lower for steam, and lowest for fuel oil or electric based systems. Systems that are low on suitability usually require the high expense of replacing the entire heating terminal and generating units with suitable water-based equipment, including piping, pumps, controls, and heat transfer media.

Table 4 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		

CHILLED-WATER CONNECTIONS

Similar to district hot-water systems, chilled-water systems can operate with either constant or variable flow. Variable-flow systems can interconnect with either building demand or constant-flow systems. Variable flow is best if dehumidification is required with comfort cooling because the supply temperature remains relatively constant. Figure 28 illustrates a typical configuration for a variable-flow building interconnection using return water control or temperature differential control. However, the district return water side may be satisfied at the expense of an increase in the building supply water temperature. Typically, this connection also monitors the building-side supply water temperature to determine if it increases too much to control building humidity. The best method of connection is the simplest, with no control valve, but with high return water temperature at varying flows. However, this method requires the building design engineer and controls contractor to implement a design that operates per the design intent.

Typical constant-flow systems are found in older buildings and may be converted to simulate a variable-flow system by blocking off the bypass line around the air handler heat exchanger coil three-way control valve. At low operating pressures, this potentially may convert a three-way bypass-type valve to a two-way modulating shutoff valve. Careful analysis of the valve actuator must be undertaken, because shut-off requirements and control characteristics are totally different for a two-way valve compared to a three-way valve.

Peak demand requirements must be determined for the building at maximum design conditions (above the ASHRAE 0.4% design values). These conditions usually include direct bright sunlight on the building, 95 to 100°F dry-bulb temperature, and 73 to 78°F wet-bulb temperature occurring at peak conditions.

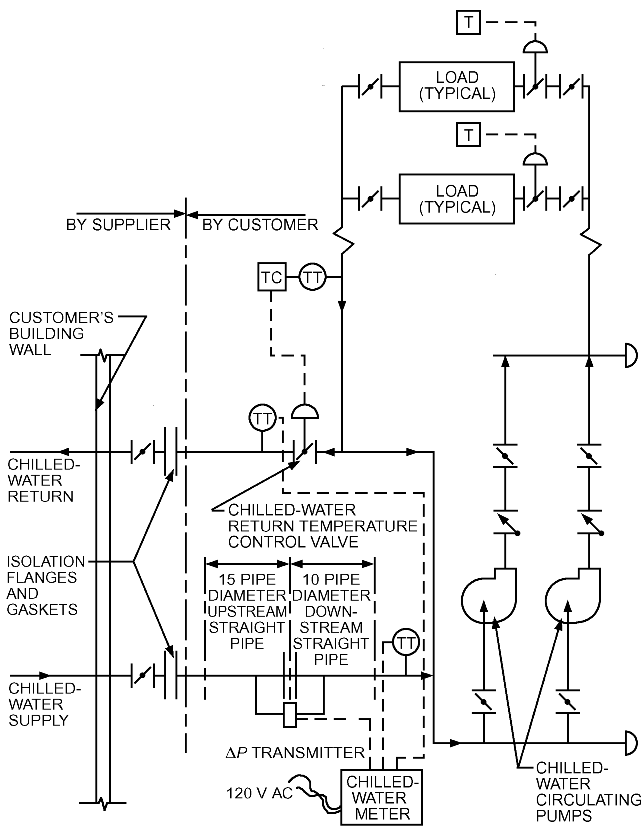


Fig. 28 Typical Chilled-Water Piping and Metering Diagram

The designer must consider the effect of return water temperature control. This is the single most important factor in obtaining a high temperature difference and providing an efficient plant. In theory, a partially loaded cooling coil should have higher return water temperature than at full load because the coil is oversized for the duty and hence has closer approach temperatures. In many real systems, as the load increases, the return water temperature tends to rise and, with a low-load condition, the supply water temperature rises. Consequently, process or critical humidity control systems may suffer when connected to a system where return water temperature control is used to achieve high temperature differentials. Other techniques, such as separately pumping each chilled water coil, may be used where constant supply water temperatures are necessary year-round.

TEMPERATURE DIFFERENTIAL CONTROL

The success of the district heating and cooling system efficiency is usually measured in terms of the temperature differential for water systems. 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 will suffer unless components in the consumer's system are modified.

Generally, maintaining a high 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 any higher building conversion cost that may result from the need for a low primary return temperature. Furthermore, optimization of the Δt is critical to the successful operation of the district energy system. That is the reason the customer's Δt must be monitored and controlled.

To optimize the Δt and meet the customer's chilled-water demand, the flow from the plant should vary. Varying the flow also saves pump energy. Chilled-water flow in the customer's side must be varied as well. Terminal units in the building connected to the chilled-water loop (i.e., air-handling units, fan-coils, etc.) 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. This requires that temperature-actuated diversity control valves be applied to the primary distribution cross connection between the supply and return piping.

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 Stultz 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 (Btu or kWh) 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 14 of the 2005 *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 5. The data in the table only provide general guidance, and the manufacturers of meters 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 orientation 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 ±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

Table 5 Flowmeter Characteristics

Meter Type	Accuracy	Range of Control	Pressure Loss	Straight Piping Requirements (Length in Pipe Diameters)
Orifice plate	±1% to 5% full scale	3:1 to 5:1	High (>5 psi)	10 D to 40 D upstream; 2 D to 6 D downstream
Electromagnetic	±0.15% to 1% rate	30:1 to 100:1	Low (<3 psi)	5 D to 10 D upstream; 3 D downstream
Vortex	±0.5% to 1.25% rate	10:1 to 25:1	Medium (3 to 5 psi)	10 D to 40 D upstream; 2 D to 6 D downstream
Turbine	±0.15% to 0.5% rate	10:1 to 50:1	Medium (3 to 5 psi)	10 D to 40 D upstream; 2 D to D downstream
Ultrasonic	±1% to 5% rate	>10:1 to 100:1	Low (<3 psi)	10 D to 40 D upstream; 2 D to 6 D downstream

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, the 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.

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 thermal efficiency of the distribution system 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 recommendations of the system manufacturer (where applicable) as well as considering the type, age, and condition of the system and its operating environment.

REFERENCES

Aamot, H. and G. Phetteplace. 1978. Heat transmission with steam and hot water. *Publication* H00128. American Society of Mechanical Engineers, New York.

Albert, M.R. and G.E. Phetteplace. 1983. Computer models for two-dimensional steady-state heat conduction. CRREL *Report* 83-10. U.S. Army Cold Regions Research and Engineering Laboratory, Hanover, NH.

Andrepont, J.S. 1995. Chilled water storage: A suite of benefits for district cooling. *District Energy* 81(1). International District Energy Association, Washington, D.C.

ASCE. 1996. *Cold regions utilities monograph.* American Society of Civil Engineers, Reston, VA.

ASHRAE. 1992. Method of testing thermal energy meters for liquid streams in HVAC systems. ANSI/ASHRAE *Standard* 125-1992 (RA 2000).

ASME. 1998. Power piping. *Standard* B31.1. American Society of Mechanical Engineers, New York.

ASTM. 1995. Standard practice for determination of heat gain or loss and the surface temperatures of insulated pipe and equipment systems by the use of a computer program. *Standard* C680-98 (R 1995). American Society for Testing and Materials, West Conshohocken, PA.

Bloomquist, R.G., R. O'Brien, and M. Spurr. 1999. *Geothermal district energy at co-located sites.* WSU-EEP 99007, Washington State University Energy Office.

Bøhm, B. 1986. On the optimal temperature level in new district heating networks. *Fernwärme International* 15(5):301-306.

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

Bottorf, J.D. 1951. *Summary of thermal conductivity as a function of moisture content*. Thesis, Purdue University, West Lafayette, IN.

CEN. 1997. Heat meters. *Standard* EN 1434. Comité Européen de Normalisation.

Chyu, M.-C., X. Zeng, and L. Ye. 1997a. Performance of fibrous glass insulation subjected to underground water attack. *ASHRAE Transactions* 103(1):303-308.

Chyu, M.-C., X. Zeng, and L. Ye. 1997b. The effect of moisture content on the performance of polyurethane insulation on a district heating and cooling pipe. *ASHRAE Transactions* 103(1):309-317.

Chyu, M.-C., X. Zeng, and L. Ye. 1998a. Behavior of cellular glass insulation on a DHC pipe subjected to underground water attack. *ASHRAE Transactions* 104(2):161-167.

Chyu, M.-C., X. Zeng, and L. Ye. 1998b. Effect of underground water attack on the performance of mineral wool pipe insulation. *ASHRAE Transactions* 104(2):168-175.

COWIconsult. 1985. *Computerized planning and design of district heating networks*. COWIconsult Consulting Engineers and Planners AS, Virum, Denmark.

CRREL. 1999. *Regional climatic constants for Equation 6 of the Corps of Engineers Guide Spec 02695*. (Best fit to mean monthly temperatures averaged for the period 1895-1996). U.S. Army Cold Regions Research and Engineering Laboratory, Hanover, NH.

EuHP. 1991. *Preinsulated bonded pipe systems*. EN253, European District Heating Pipe Manufacturers Association, Brussels.

Farouki, O.T. 1981. Thermal properties of soils. CRREL *Monograph* 81-1. U.S. Army Cold Regions Research and Engineering Laboratory, Hanover, NH.

Fox, J.A. 1977. *Hydraulic analysis of unsteady flow in pipe networks*. John Wiley & Sons, New York.

Geiringer, P.L. 1963. *High temperature water heating: Its theory and practice for district and space heating applications*. John Wiley & Sons, New York.

Grober, H., S. Erk, and U. Grigull. 1961. *Fundamentals of heat transfer.* McGraw-Hill, New York.

IDHA. 1969. *Code for steam metering*. International District Energy Association, Washington, D.C.

IDHA. 1983. *District heating handbook*, 4th ed. International District Energy Association, Washington, D.C.

Jeppson, R.W. 1977. *Analysis of flow in pipe networks*. Ann Arbor Science, MI.

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

Lunardini, V.J. 1981. *Heat transfer in cold climates*. Van Nostrand Reinhold, New York.

McCabe R.E., J.J. Bender, and K.R. Potter. 1995. Subsurface ground temperature—Implications for a district cooling system. *ASHRAE Journal* 37(12):40-45.

Minkowycz, W.J., E.M. Sparrow, G.E. Schneider, and R.H. Pletcher. 1988. *Handbook of numerical heat transfer*. John Wiley & Sons, New York.

MSS. 1993. Pipe hangers and supports—Materials, design and manufacture. *Standard* MS-58-1993. Manufacturers Standardization Society of the Valve and Fittings Industry, Vienna, VA.

MSS. 1996. Pipe hangers and supports—Selection and application. *Standard* MS-69-1996. Manufacturers Standardization Society of the Valve and Fittings Industry, Vienna, VA.

NACE. 1996. Control of external corrosion on underground or submerged metallic piping systems. *Standard* RPO169-96. National Association of Corrosion Engineers, Houston, TX.

NAS. 1975. *Technical Report No.* 66. National Academy of Sciences National Research Council Building Research Advisory Board (BRAB), Federal Construction Council.

Ottmer, J.H. and J.B. Rishel. 1993. Airport pumping system horsepower requirements take a nose dive. *Heating, Piping and Air Conditioning* 65(10).

Phetteplace, G.E. 1989. Simulation of district heating systems for piping design. International Symposium on District Heat Simulation, Reykjavik, Iceland.

Phetteplace, G.E., D. Carbee, and M. Kryska. 1991. *Field measurement of heat losses from three types of heat distribution systems*. CRREL SR 631. U.S. Army Cold Regions Research and Engineering Laboratory, Hanover, NH.

Phetteplace, G. and V. Meyer. 1990. Piping for thermal distribution systems. CRREL *Internal Report* 1059. U.S. Army Cold Regions Research and Engineering Laboratory, Hanover, NH.

Phetteplace, G.E., W. Willey, and M.A. Novick. 1981. *Losses from the Fort Wainwright heat distribution system.* CRREL SR 81-14. U.S. Army Cold Regions Research and Engineering Laboratory, Hanover, NH.

Pomroy, J. 1994. Selecting flowmeters. *Instrumentation & control systems*. Chilton Publications.

Rao, S.S. 1982. *The finite element method in engineering.* Pergamon, New York.

Rasmussen, C.H. and J.E. Lund. 1987. *Computer aided design of district heating systems*. District Heating Research and Technological Development in Denmark, Danish Ministry of Energy, Copenhagen.

Reisman, A.W. 1985. *District heating handbook*, vol. 2: *A handbook of district heating and cooling models*. International District Energy Association, Washington, D.C.

Rohsenow, W.M. 1998. *Handbook of heat transfer.* McGraw-Hill, New York.

Segan, E.G. and C.-P. Chen. 1984. Investigation of tri-service heat distribution systems. CERL *Technical Report* M-347. U.S. Army Construction Engineering Research Laboratory (CERL), Champaign, IL.

Siegel, R. and J.R. Howell. 1981. *Thermal radiation heat transfer.* Hemisphere, New York.

Sleiman, A.H. et al. 1990. *Guidelines for converting building heating systems for hot water district heating*. NOVEM, Sittard, The Netherlands.

Stephenson, D. 1981. *Pipeline design for water engineers.* Elsevier Scientific, New York.

Stewart, W.E. and C.L. Dona. 1987. Water flow rate limitations. *ASHRAE Transactions* 93(2):811-825.

Streeter, V.L. and E.B. Wylie. 1979. *Fluid mechanics*. McGraw-Hill, New York.

Stultz, S.C. and J.B. Kitto, eds. 1992. *Steam: Its generation and use*, 40th ed. Chapter 40-13, Measurement of steam quality and purity. Chapter 40-19, Measurement of steam flow. Babcock & Wilcox, Barberton, OH.

Werner, S.E. 1984. *The heat load in district heating systems*. Chalmers University of Technology, Göteborg, Sweden.

BIBLIOGRAPHY

Bowling, T. 1990. *A technology assessment of potential telemetry technologies for district heating*. International Energy Agency and NOVEM, Sittard, The Netherlands.

Holtse, C. and P. Randlov, eds. 1989. *District heating and cooling R&D project review*. International Energy Agency and NOVEM, Sittard, The Netherlands.

Kusuda, T. and P.R. Achenbach. 1965. Earth temperature and thermal diffusivity at selected stations in the United States. *ASHRAE Transactions* 71(1):61-75.

Mørck, O. and T. Pedersen, eds. 1989. *Advanced district heating production technologies*. International Energy Agency and NOVEM, Sittard, The Netherlands.

Ulseth, R., ed. 1990. *Heat meters: Report of research activities*. International Energy Agency and NOVEM, Sittard, The Netherlands.

U.S. Air Force, Army, and Navy. 1978. Engineering weather data. Dept. of the Air Force *Manual* AFM 88-29, Dept. of the Army *Manual* TM 5-785, and Dept. of the Navy *Manual* NAVFAC P-89.

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