# CHAPTER C3
# STEAM SYSTEMS PIPING

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## INTRODUCTION

### General

In this chapter we will first review the basics of piping system design for steam systems applications; we will then consider specifics for underground steam piping and steam piping used in power plants.

### Definitions and Terminology

*Condensate.* Condensed steam.

*Trunk line distribution system.* Distribution system with a large-diameter line leaving the boiler plant; as lateral branches are installed off it for service, the trunk line gradually diminishes in diameter.

*Main and feeder network distribution system.* Distribution system that receives its supply of steam through a high-pressure feeder main leading from the plant through the network; the size of the feeder main required in this case is not as large as in a trunk-line system with the same boiler-plant steam pressure.

*Protective conduits (typically for underground steam lines).* Enclosures for underground steam mains and services to (1) protect the pipe and insulation from damage due to earth pressure and impact loadings, (2) allow free longitudinal expansion and contraction while held in proper alignment, and (3) prevent groundwater seepage or flooding by providing either drains or a completely waterproof structure.

*Light water reactors (LWR).* Nuclear power reactors of either the pressurized water reactor (PWR) or boiling water reactor (BWR) type.

*ASME Class 1[1] piping.* Includes main steam piping up to and including the first stop valve outside the reactor containment for BWRs and it is designated as ASME Class 1.

*ASME Class 2[1] piping.* Includes main steam piping up to and including the first stop valve outside the reactor containment for pressurized water reactors (PWRs) and BWR main steam piping after the first isolation valve outside the reactor containment and it is designated as ASME Class 2.

*ASME B31.1[2] piping.* Includes main steam piping downstream of the first stop valve outside the reactor containment in PWRs and piping external to boilers in fossil power plants, and it is constructed to the ASME B31.1 Code for Pressure Piping.

*ASME B31.3[3] piping.* Includes steam process piping in industrial plants and it is designed to the ASME B31.3 Code for Process Piping.

*BEP.* Boiler external piping.

*DNBR.* Departure from nucleate boiling ratio.

*FFWT.* Final feedwater temperature.

*HARP.* Heater above the reheat point.

*IP.* Intermediate pressure (turbine).

*IV.* (Governor-operated) intercept valve.

*kPa.* Pressure in kilo pascals.

*LP.* Low pressure (turbine).

*MPa.* Pressure in mega pascals.

*NSSS.* Nuclear steam supply system.

*Pa.* Pressure drop or pressure in pascals.

*psi.* pressure drop in pounds (force) per square inch, $lb_f/in^2$.

*psia.* pressure in pounds (force) per square inch, absolute.

*psig.* pressure in pounds (force) per square inch, gauge.

*PWHT.* Postweld heat treatment.

*VWO.* Valves wide open (steam turbine).

## Nomenclature

*A.* An additional thickness to provide for material removed in threading, corrosion or erosion allowance, and material required for structural strength of the pipe, as appropriate, in (mm). See Table C3.1 for selected values of $A$.

*d.* Inside diameter of pipe, in (mm). In using Eq. (C3.4), the value of $d$ is for the maximum possible inside diameter allowable under the purchase specifications.

$D_o$. Outside diameter of pipe, in (mm). For design calculations, the outside diameter of pipe as given in ASME B36.10M and ASME B36.19M[4,5] and specifications shall be used in obtaining the value of $t_m$.

*L.* Length of pipe, ft (m).

*P.* Internal design pressure, psi (Pa).

$P_a$. Calculated maximum allowable internal pressure, psi (Pa), for straight pipe which shall be at least equal the design pressure. $P_a$ may be used for piping products with pressure ratings equal to that of straight pipe.[6] For pipe products where the

**TABLE C3.1** Selected Values of $A$[1,2]

| Type of pipe | $A$, in (mm) |
|---|---|
| Cast-iron pipe centrifugally cast or cast horizontally in green sand molds | 0.14 (3.56) |
| Cast-iron pipe, pit cast | 0.18 (4.57) |
| Threaded steel, wrought-iron or nonferrous ¾ in (19 mm) nominal and smaller | 0.065 (1.65) |
| Threaded steel, wrought-iron or nonferrous 1 in (25 mm) nominal and larger | Depth of thread |
| Grooved steel, wrought-iron, or nonferrous | Depth of groove plus 1/64 in. (0.40 mm) |
| Plain-end steel or wrought-iron pipe or tube for sizes 1 in (25 mm) and smaller | 0.05 (1.27) |
| Plain-end steel or wrought-iron pipe or tube for sizes over 1 in (25 mm) | 0.065 (1.65) |
| Plain-end nonferrous pipe or tube | 0.00 |

pressure rating may be less than that of the pipe (e.g., flanged joints and reinforced branch connections where part of the required reinforcement is in the run pipe), the design pressure shall be used instead of $P_a$.

$\Delta P$. Pressure drop, psi (Pa).

$S$. Maximum allowable stress for the material at the design temperature, psi (Pa).

$t$. Specified or actual wall thickness minus, as appropriate, material removed in threading, corrosion or erosion allowance, material manufacturing tolerances, bending allowance (see Table C3.2), and material to be removed by counterboring, in (mm).

$t_m$. Minimum required wall thickness, in (mm). If pipe is ordered by its nominal wall thickness, the manufacturing tolerance on wall thickness must be taken into account.

$W$. Steam flow, $lb_m$/min (kg/hr).

$Y$. Density of steam, $lb_m$/cu ft (kg/$m^3$) [used in Eqs. (C3.2) and (C3.2M)].

$y$. A coefficient having values as given in Table C3.3. For pipe with a $D_o/t_m$ ratio

**TABLE C3.2** Minimum Thickness Prior to Bending*[18]

| Radius of bend | Furnace bending | Induction and incremental bending | Rotary draw bending | Ram and roll bending |
|---|---|---|---|---|
| 6 Dn | $1.06t_m$ | $1.06t_m$ | $1.09t_m$ | $1.08t_m$ |
| 5 Dn | $1.08t_m$ | $1.08t_m$ | $1.14t_m$ | $1.10t_m$ |
| 4 Dn | $1.14t_m$ | $1.10t_m$ | $1.20t_m$ | $1.13t_m$ |
| 3 Dn | $1.25t_m$ | $1.14t_m$ | $1.26t_m$ | $1.17t_m$ |
| 2 Dn | | $1.22t_m$ | | |
| 1.5 Dn | | $1.30t_m$ | | |

* $t_m$ is determined by Eq. (C3.3) or (C3.4).

**TABLE C3.3** Values of Coefficient $y$[1,2]

| Temperature °F (°C) | 900 (482) and below | 950 (510) | 1000 (538) | 1050 (566) | 1100 (593) | 1150 (621) and above |
|---|---|---|---|---|---|---|
| Ferritic steels | 0.4 | 0.5 | 0.7 | 0.7 | 0.7 | 0.7 |
| Austenitic steels | 0.4 | 0.4 | 0.4 | 0.4 | 0.5 | 0.7 |

***Note:*** The value of $y$ may be interpolated between the 50°F (28°C) values shown in the table. For nonferrous materials and cast-iron, $y$ equals 0.4.

less than 6, the value of $y$ for ferritic and austenitic steels designed for temperatures of 900°F (482°C) shall be as calculated from Eq. (C3.1), as follows:

$$y = \frac{d}{d + D_o} \tag{C3.1}$$

## Types of Systems

***Steam Distribution Systems.*** While there are no fixed standards for the design of steam distribution piping systems, most of the systems fall into one of two general classes: (1) a trunk-line distribution network system and (2) a main and feeder distribution network system.

In Case 1, the diameter of the trunk line leaving the boiler plant is large, and as lateral branches are installed off it for service, the diameter of the trunk line is gradually reduced as the needs for carrying capacity are diminished.

In Case 2, the main and feeder network distribution system receives its supply of steam through a high-pressure feeder main leading from the plant through the network. Advantage is taken of the pressure drop available for the transportation of large volumes of steam to the low-pressure network.

Figures C3.1 and C3.2 show typical steam distribution systems.

Since piping is the largest individual factor in the selection and design of a steam distribution system, the major items that must be resolved are as follows: (1) pipe size, (2) wall thickness, (3) materials selection, (4) types of joints, (5) proper insulation, (6) a protective conduit for the pipe and insulation from water and mechanical damage, (7) drainage of condensate, (8) provision for thermal expansion with controlling anchorage, and (9) safety provisions.

***Underground Steam Piping.*** Underground piping for steam distribution has been a highly specialized field of engineering peculiar to the district-heating industry; today underground steam piping is also used to carry process steam. With the advent of groups of buildings such as in housing developments, institutions, and industrial plants, central heating systems and steam distribution problems are no longer restricted to the district-heating industry. Steam piping systems may be buried and not readily available for enlargement, replacement, and repair. Such piping must be protected from ground elements and excessive heat losses; thus it is important that before such a system is installed every phase of its design and operation be understood. Figures C3.1 and C3.2 show typical steam distribution systems which would be fed by underground steam lines.

See “Protective Conduits for Underground Lines” for more information about

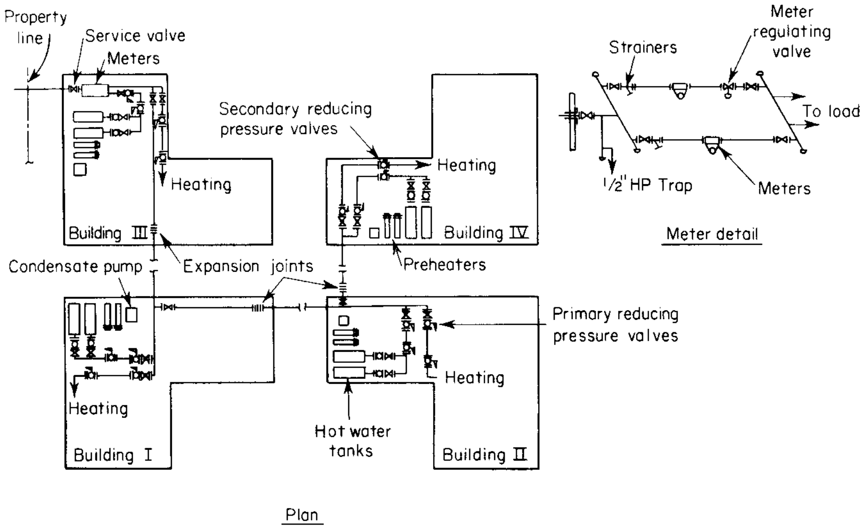


**FIGURE C3.1** Typical steam distribution system for a housing development.

protective enclosures and "Drainage of Condensate" for condensate drainage considerations for underground piping.

***Fossil-Fueled Power Plants.*** The main steam system conducts superheated steam from the steam generator (economizer plus evaporator and superheater) to the turbines as shown in Fig. B1.1 in Part B of this handbook. Figure C3.3 shows a simplified schematic of a main steam system. The main steam system may also provide steam for various auxiliary services.

The piping layout, for a steam power plant, in addition to paying due respect to economic factors, should consider (1) the layout's mechanical design or ability to function properly and efficiently with respect to the mechanical equipment which it serves, (2) the layout's convenience from an operating standpoint, and (3) the layout's appearance as a coordinated part of the plant. Although the relative importance of these basic points obviously falls in the order named, each has an important bearing on the acceptability of any arrangement, as will be discussed in the following sections.

B31.1[1] states that when boilers are connected to a common header, the connection from each boiler having a manhole opening shall be fitted with two stop valves having an ample free-blow drain between them. The discharge of this drain shall be visible to the operator while manipulating the valve. The stop valves shall consist preferably of one automatic nonreturn valve (see next to the boiler) and a second valve of the outside-screw-and-yoke type, or two valves of the outside-screw-and-yoke type shall be used. When a second stop valve or valves is required, it shall have a pressure rating at least equal to that required for the expected steam temperature and pressure at the valve, or the pressure rating at least equal to 85 percent of the lowest set pressure of any safety valve on the boiler drum and for the expected temperature of the steam at the valve, whichever is greater. All valves and fittings on steam lines shall have a pressure rating of at least 100 psig [700 kPa (gauge)] in accordance with the applicable American National Standard.

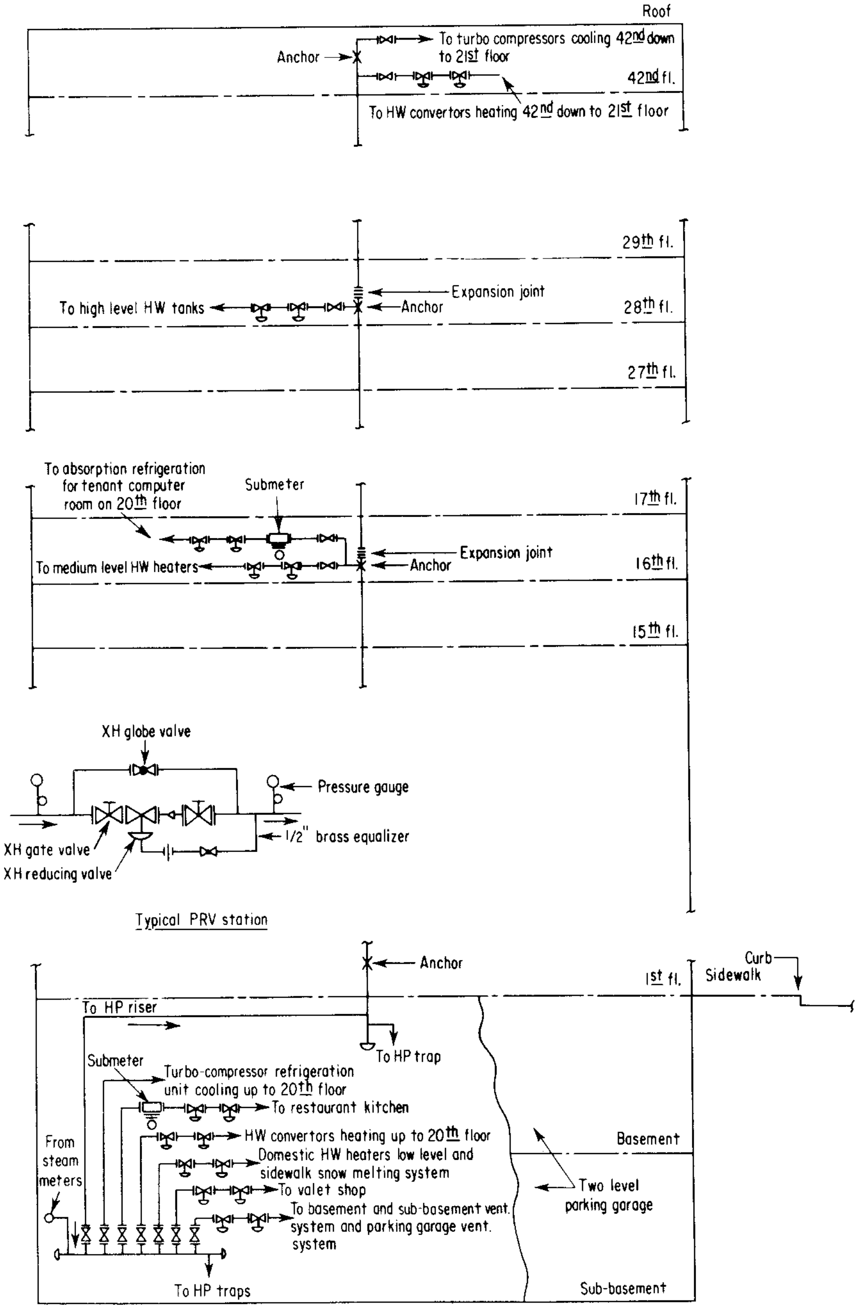


**FIGURE C3.2** Typical high-pressure steam-distribution system in a tower office building.

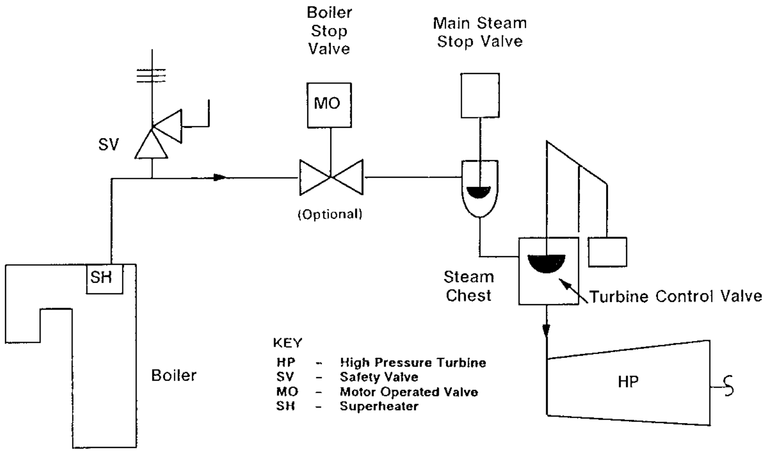


**FIGURE C3.3** Main steam system.

Availability of steam generators has increased to the extent that single units are used almost universally to supply the steam to turbines of sizes up to approximately 1000 MW. In fossil-fueled power plants exhaust steam from the high-pressure turbine is piped back to the steam generator, where it is reheated in a special section before being returned to the inlet of the intermediate pressure or reheat turbine.

The cold reheat system (CRS) and the hot reheat steam (HRS) system shown in Fig. C3.4 are treated here in one section because they are parts of the same overall system used for reheating steam. The CRS conducts steam from the outlet

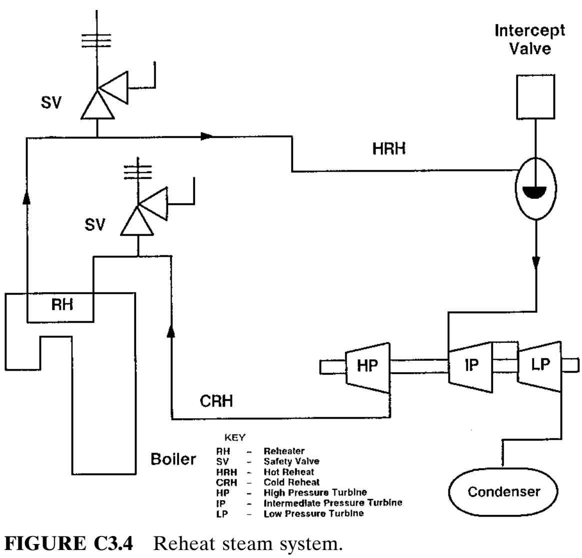


**FIGURE C3.4** Reheat steam system.

of the high-pressure turbine to the reheater inlet and provides steam for the auxiliary steam and extraction steam systems. The CRS is the normal auxiliary steam source, but at low loads auxiliary steam is usually augmented from the main steam system via a pressure-reducing station.

The exhaust steam from the high-pressure turbine is transported in single or multiple leads into the reheater inlet. Each reheater may have a desuperheater for reheater outlet temperature control, and safety valves at the reheater inlet. A cross-connection (cross-tie) may be used between the two cold reheat leads to provide steam to the associated feedwater heater and the auxiliary steam system without creating a pressure imbalance at the reheater inlet. This system has provisions for isolation of the reheater for hydrostatic testing.

The extraction steam system conducts steam from the high-pressure turbine, cold reheat line, intermediate-pressure turbine, and the low-pressure turbines to the feedwater heaters. This extraction steam is required for feedwater heating. Feedwater heating increases cycle efficiency; in addition, the extraction steam system may provide steam for the feedwater pump turbine. In nuclear units, the system also removes moisture from the turbine to provide protection for the lower-pressure turbine blades and to increase turbine efficiency.

The hot reheat system conducts superheated steam from the reheater outlet header to the intermediate-pressure turbine or to the inlet of the low-pressure turbine. One or more hot reheat leads may be furnished. Each lead has a safety valve for reheater protection. A cross-connection between the two hot reheat leads is often furnished to ensure equal pressure at each reheat stop and intercept valve before entering the turbine.

***Nuclear-Fueled Power Plants.*** Only the stationary light water reactor, either of the PWR or BWR type, is discussed in this chapter as being typical of nuclear practice. In the typical boiling water reactor the reactor is the steam generator. Water is circulated through the reactor core, producing steam which is separated from recirculation water, dried in the top of the reactor vessel, and directed to the steam turbine.

The pressurized water reactor uses two closed systems—a primary system including the reactor and its cooling system, and a secondary system including a turbine-generator, a condensate system, and a feedwater system. The two systems communicate with each other at the steam-generator tube interface, where pressurized water of the primary system transfers fission-reaction heat to the steam-generator feedwater in the secondary system, producing steam to drive the turbine generators. PWR power plants typically utilize two, three, or four primary loops, each containing a steam generator which transfers the energy from primary coolant to the feedwater on the secondary side.

The nuclear steam supply system (NSSS) customarily consists of those components in contact with the reactor coolant, and specialized auxiliary machinery.

The main steam system in LWRs transports steam from the outlet of the reactor/steam generators to the turbine stop valves. The main steam system also provides steam for various auxiliary services. Additionally, it provides a means of controlled heat release from the NSSS during periods of station electrical load rejection.

In PWRs, each steam generator supplies steam to a line which connects to a common main steam manifold. Valves in each line permit isolation of individual steam generators. Safety valves on each line provide pressure-relief protection for the steam generators. The main steam manifold supplies steam to the high-pressure turbine throttle valves, the moisture separator reheaters, and auxiliary steam loads.

The main steam system includes a turbine bypass system that provides a direct

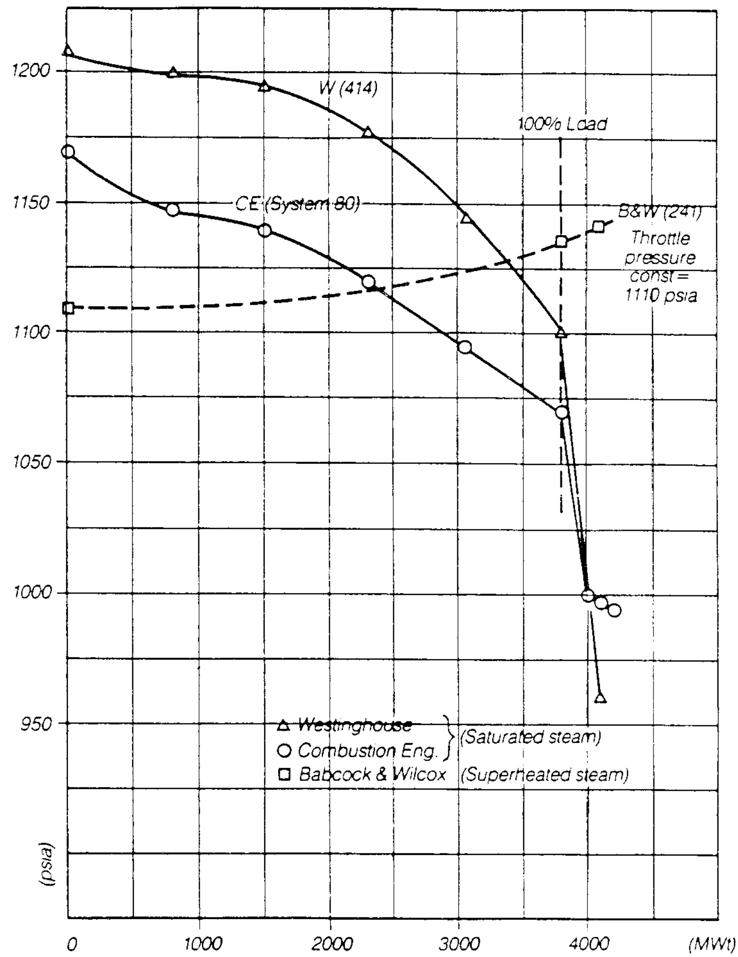


**FIGURE C3.5** Typical NSSS performance curves.

steam path to the condenser and to the atmosphere in order to prevent unnecessary reactor trips during load rejections up to 100 percent of full electrical load.

The turbine bypass system also permits testing NSSS at power levels up to 55 percent without having the turbine loaded. It is also used to maintain reactor coolant temperature during hot standby and shutdown operation and is used to conduct controlled cooldown of the plant to the point where the residual heat removal system can be placed in operation.

For the PWR and BWR, the main steam system design pressure is typically about 1200 psia (8280 kPa) (see Fig. C3.5) and 600°F (316°C). Actual design conditions depend upon the specific reactor design. Main steam line sizing is determined by performing an economic analysis of pressure drop, pipe cost, and so on, with a maximum velocity of 15,000 fpm (250 fps) (4570 m/min, 76 m/s); see "Preventing

Turbine Overspeed'' for more flow velocity data. Preliminary pipe sizing is based on a 3 percent pressure drop from the steam generator outlet to the turbine stop valves. Wall thickness is calculated using the equations presented in ''Design Pressure.''

For PWR and BWR main steam lines, turbine water induction problems are of a great concern as the main steam is saturated or contains some moisture. Prevention criteria are to be as specified in ASME Standard No. TDP-2,[7] ''Recommended Practices for the Prevention of Water Damage to Steam Turbines Used for Electric Power Generation,'' Nuclear Fueled Plants.

Each type of nuclear plant has isolation valves outside of the reactor containment. BWR plants also have isolation valves in main steam lines inboard of the reactor containment. Since the BWR main steam is radioactive, it must be shielded, and special considerations must be made in its design to minimize crud traps (small deposits of radioactive contaminants in flow passages). When welding pipes together, the use of backing rings is avoided.

In LWR nuclear plants, reheating is achieved in a combined moisture separator and reheater unit. High-pressure turbine exhaust steam passes through the moisture separator portion of the unit, where most of the moisture is removed mechanically. The steam is then reheated in one or two stages by passing over the bundles of tubes containing high-temperature heating steam. For single-stage reheat and for the second stage of two-stage reheat, the heating steam is supplied from the main steam system at a point ahead of the turbine stop valves. Pipelines carrying live steam to reheaters belong to the main steam system. For two-stage reheat, the first-

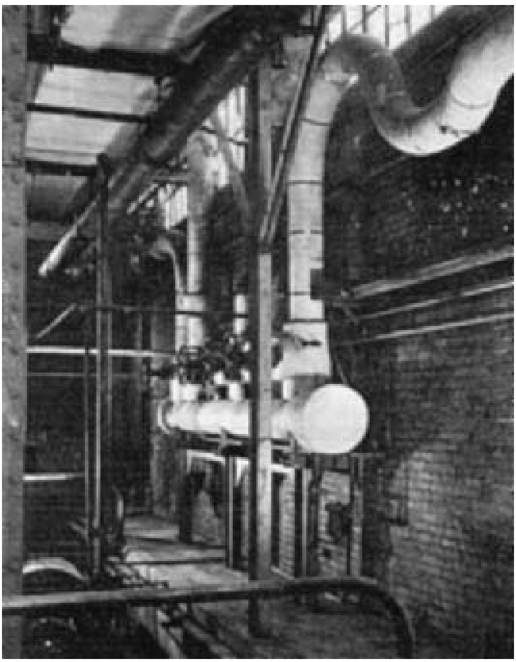

**FIGURE C3.6** Steam header in industrial plant. *(Courtesy of Valve World.)*

stage heating steam supply is from an intermediate stage of the high-pressure turbine. More advanced liquid metal cooled fast breeder reactor (LMFBR) nuclear power plants may employ the liquid metal for steam reheating.

***Industrial (Process) Power Plants.*** In industrial power plants, power generation may be considered as a by-product, depending upon the process steam requirements. Process steam is the main product of the cogenerating plant. Industrial turbines, with controlled exhaust and extraction pressures, are very efficient throttling devices (replacing valves), supplying steam at desired pressures to a process plant (e.g., oil refinery) while generating electricity. Variations in process heat and power demands usually do not coincide. In a case where electrical power generation requirements are specified, separate condensing steam turbines or gas turbines must be installed in the power plant to make up the difference between electric power demand and the process-heat dependent power generated by industrial steam turbines.

A typical steam header in a modern industrial plant is shown in Fig. C3.6, which is a photograph of an installation in the Wabash, Indiana, plant of the General Tire Company. As may be noted, the boiler leads and connections supplying steam to process, heating, pumps, and so on, are brought down to this header, which is located near the floor so that all valves are readily accessible.

## REFERENCE DOCUMENTS

Various codes and standards and other reference documents used for the design of steam systems piping are listed under ''References'' at the end of this chapter. The reader may find other documents suitable for use in the area of steam systems piping.

Extensive safety requirements for power-piping systems are contained in the ASME Boiler and Pressure Vessel Code, Section I, Power Boilers, and ASME Section III, Nuclear Power Plant Components, and the ASME B31.1 Power Piping Code. In designing the component parts of piping systems within the jurisdiction of these codes, reference should be made to specific provisions as representing standards for minimum safety requirements, but this is not intended to indicate necessarily the best practice known to the art. Requirements of the Code for Pressure Piping are not compulsory in any state until they have been adopted as law by that state. They are in common use, however, and frequently are referred to in contract specifications and similar documents. Sometimes, even though the state has not adopted the codes, the insurance carriers make it a requirement to comply with certain codes.

The selection of suitable dimensional standards for flanges, fittings, valves, pipe, and bolting for ordinary service conditions can be made from the appropriate publications of the American National Standards Institute (ANSI), American Standards for Testing and Materials (ASTM), and American Society of Mechanical Engineers (ASME) which are referenced in the governing codes, such as B31.1, ASME I, ASME III, and ASME B31.3. Appendix E10 provides a partial list of international codes and standards.

## DESIGN CONSIDERATIONS

### Design Conditions

Design conditions or loadings include design pressure, design temperature, and design mechanical loads.

The ASME Code, Section III, NCA-2142.1[1] defines *design pressure* as follows: "The specified internal and external Design Pressure shall not be less than the maximum difference in pressure between the inside and outside of the item, or between any two chambers of a combination unit, which exists under the most severe loadings for which the Level A Service Limits are applicable. The Design Pressure shall include allowances for pressure surges, control system error, and system configuration effects such as static pressure heads." Level A service limits encompass those normal loadings which a component may be subjected in the performance of its specified service function.

The B31.1 code, paragraph 101.2[2] defines *internal design pressure* as follows: "The internal design pressure shall not be less than the maximum sustained operating pressure (MSOP) within the piping system including the effects of static head." *External design pressure* is defined as follows: "Piping subject to external pressure shall be designed for the maximum differential pressure anticipated during operating, shutdown, or test conditions."

The ASME Code, Section III, NCA-2142.1[1] defines *design temperature* as follows: "The specified Design Temperature shall not be less than the expected maximum mean metal temperature through the thickness of the part considered for which Level A Service Limits are specified. Where a component is heated by trace heating, such as induction coils, jacketing, or by internal heat generation, the effect of such heat input shall be considered in establishing the Design Temperature. The Design Temperature shall consider control system error and system configuration effects."

The B31.1 code, paragraph 101.3[2] defines *design temperature* as follows: "The piping shall be designed for a metal temperature representing the maximum sustained condition expected. . . . For steam, feedwater, and hot water piping leading from fired equipment (such as boiler, reheater, superheater, economizer, etc.), the design temperature shall be based on the expected continuous operating condition plus the equipment manufacturer's guaranteed maximum temperature tolerance."

The ASME Code, Section III, NCA-2142.1[1] defines *design mechanical loads* as follows: "The specified Design Mechanical Loads shall be selected so that when combined with the effects of Design Pressure, they produce the highest primary stresses of any coincident combination of loadings for which Level A Service Limits are designated in the Design Specification."

The B31.1 code, paragraph 101.5[2] identifies *design mechanical loads* generally as dynamic effects as follows: Impact forces caused by all external and internal conditions shall be considered in the piping design. Water or steam hammer is a form of impact load which is caused by rapid opening or closing of a valve in the system. Exposed piping shall be designed to withstand wind loadings, using meteorological data to determine wind forces. The effect of earthquakes, where applicable, shall be considered in the design of piping, pipe supports and restraints, using data for the site as a guide in assessing the forces involved. Piping shall be arranged and supported with consideration of vibration. Suitable vibration dampers, sway braces, restraints, and anchors shall be used to control the movement of piping due to vibration.

***Design Pressure.*** The factors which determine the size of a steam pipe for a specific installation are as follows: (1) the initial steam pressure, and other conditions (temperature), (2) the minimum permissible discharge pressure, (3) the allowable velocity, (4) the quantity of steam, and (5) the length of line, including equivalent lengths for fittings. By knowing or assuming any one or all of these factors, the pipe size may be calculated by means of one of the pressure-drop formulas of Chap. B8 in this handbook.

The Unwin formula[8,9] has been widely used in the district-heating industry for many years. At elevated velocities, Unwin's formula gives pressure drops known to be higher than actual. This formula, in English units, is as follows:

$$\Delta P = \frac{0.0001306 W^2 L(1 + 3.6/d)}{Y d^5} \tag{C3.2}$$

where $P$ = pressure drop (psi)
$W$ = steam flow rate (lb/min)
$L$ = length of pipe (ft)
$d$ = pipe inside diameter (in)
$Y$ = steam density (lb/ft$^3$)

*Example C3.1*. Find the pressure drop in an NPS 8 straight pipe 100 ft long with a flow of 40,000 lb/hr of saturated steam and an initial pressure of 150 psia [steam density ($Y$) = 0.3318 lb/ft$^3$].

*Solution*. $P$ = 0.774 psi

The Unwin formula in Sl units is as follows:

$$\Delta P = \frac{0.6753 \times 10^6 W^2 L(1 + 91.4/d)}{Y d^5} \tag{C3.2M}$$

where $P$ = pressure drop (Pa)
$W$ = steam flow rate (kg/hr)
$L$ = length of pipe (m)
$d$ = pipe inside diameter (mm)
$Y$ = steam density (kg/m$^3$)

*Example C3.1M*. Find the pressure drop in a 203.2 mm diameter straight pipe 30.48 m long with a flow of 18,144 kg/hr of saturated steam and an initial pressure of 1034 kPa [steam density ($Y$) = 5.313 kg/m$^3$].

*Solution*. $P$ = 5337 Pa

Note that both of these examples represent the same physical pipe and flow conditions, and both result in the same pressure drop.

Pressure drops through fittings such as elbows, tees, and valves vary in proportion to the pressure drop through straight pipe. Because of this fact, it is possible to express the resistance of fittings as equivalent lengths of straight pipe and to compute the pressure drop for the whole line as if it consisted only of straight pipe as discussed in Chap. B8.

Where grid or network systems are involved, the computation of pipe size, pressure drops, and quantities of steam flowing presents a more complex problem. See Chap. B8 for a discussion of calculating these properties.

Minimum pipe-wall thickness for either heavy-wall or thin-wall pipe may be calculated from the following formulas from NC-3641 of Section III of the ASME Code[1] or equivalent, ASME B31.1 or B31.3 Code.[2,3] See Chap. B2 in this handbook for more information on wall thickness. The $y$ factors adjusts for elastic or plastic properties of the respective materials over the expected range of operating temperatures:

$$t_m = \frac{PD_o}{2(S + Py)} + A \tag{C3.3}$$

$$t_m = \frac{Pd + 2SA + 2yPA}{2(S + Py - P)} \tag{C3.4}$$

Equation (C3.3) is used when outside pipe diameter $D_o$ is considered for calculating required minimum wall thickness, $t_m$, whereas Eq. C3.4 is utilized with inside pipe diameter, $d$.

The allowable working pressure of pipe may be determined from the following equation:

$$P_a = \frac{2St}{D_o - 2yt} \tag{C3.5}$$

Large fossil power plants generally use a 2400 psig (16.6 MPa) drum-type steam generator or 3500 psig (24.2 MPa) once-through, supercritical steam generator. Small- and medium-sized power plants of less than 200 MW use lower standard pressures. For the large fossil power plants, the main steam design pressure is typically 2650 or 3860 psig (18.4 MPa or 26.7 MPa) (lowest superheater safety valve setting), and design temperature is 1015 or 1050°F (546 or 566°C) (maximum expected superheater outlet temperature). The steam at these conditions is superheated, so there is no moisture, but the lines must still have moisture-removal capabilities.

***Boiler External Piping (BEP).*** ASME B31.1[2] Paragraph 122.1.1 presents the minimum pressure and temperature and other special requirements for boiler external piping including the following:

> It is intended that the design pressure and temperature be selected sufficiently in excess of any expected operating conditions, not necessarily continuous, to permit satisfactory operation without operation of the overpressure protection devices. Also, since the operating temperatures of fired equipment can vary, the expected temperature at the connection to the fired equipment shall include the manufacturer's maximum temperature tolerance.

For steam piping the value of design pressure, $P$, to be used in Eqs. (C3.3), (C3.4), and (C3.5) according to ASME B31.1 Paragraph 122.1.2 shall be as follows:

> (A.1) For steam piping connected to the steam drum or to the superheater inlet header up to the first stop valve in each connection, the value of $P$ shall be not less than the lowest pressure at which any drum safety valve is set to blow, and the $S$ value shall not exceed that permitted for the corresponding saturated steam temperature.
>
> (A.2) For steam piping connected to the superheater outlet header up to the first stop valve in each connection, the design pressure, except as otherwise provided in (A.4) below shall be not less than the lowest pressure at which any safety valve on the superheater is set to blow, or not less than 85 percent of the lowest pressure at which any drum safety valve is set to blow, whichever is

greater, and the *S* value for the material used shall not exceed that permitted for the expected steam temperature.

(A.3) For steam piping between the first stop valve and the second valve, when one is required by Para. 122.1.7, the design pressure shall be not less than the expected maximum sustained operating pressure or 85 percent of the lowest pressure at which any drum safety valve is set to blow, whichever is greater, and the *S* value for the material used shall not exceed that permitted for the expected steam temperature.

(A.4) For boilers installed on the unit system (i.e., one boiler and one turbine or other prime mover) and provided with automatic combustion control equipment responsive to steam header pressure, the design pressure for the steam piping shall be not less than the design pressure at the throttle inlet plus 5 percent, or not less than 85 percent of the lowest pressure at which any drum safety valve is set to blow, or not less than the expected maximum sustained operating pressure at any point in the piping system, whichever is greater, and the *S* value for the material used shall not exceed that permitted for the expected steam temperature at the superheater outlet. For forced-flow steam generators with no fixed steam and waterline, the design pressure shall also be no less than the expected maximum sustained operating pressure.

(A.5) The design pressure shall not be taken as less than 100 psig [700 kPa (gauge)] for any condition of service or material.

For example, for a fossil power plant with the lowest pressure at which any drum safety valve is set to blow at 2400 psig (16.6 MPa), the design pressure shall not be less than maximum sustained operating pressure for all steam piping connected to the steam drum or to the superheater outlet header up to the first stop valve, or 2040 psig (85 percent of 2400) (14.2 MPa, 85 percent of 16.6 MPa), whichever is greater, for piping as noted in A.2, A.3, or A.4.

Initial steam conditions at the steam generator outlet depend upon the plant under consideration. For process plants, steam parameters are dictated by the process requirements. In nuclear power plants, steam parameters depend upon the characteristics of the nuclear reactor which is the source of heat for generating steam. The maximum permissible heat-flux for fuel, reactor vessel pressure, and the DNBR are limiting factors influencing turbine initial steam conditions of nearly saturated steam at about 1100 psia (7.6 MPa) (see Fig. C3.5) in a NSSS with a PWR. In BWR the steam moisture problem assumes a major role in optimizing steam pressure at about 1000 psia (6.9 MPa).

The most freedom in selecting initial steam conditions is found in fossil-fueled power plants, which may be designed as subcritical or supercritical units. The question then arises: What are the most economical initial steam conditions for the present technological state of the art? Generally, from a thermodynamic point of view, an increase in initial steam temperature or pressure increases the power plant cycle efficiency (see Fig. C3.7).

***Cold Reheat Systems.*** The cold reheat and hot reheat steam systems for fossil power plants, are designed to ASME B31.1. For PWR- and BWR-equipped plants the reheat lines are part of the turbine manufacturer's area of responsibility and are designed to turbine manufacturer's criteria which must equal or exceed the rules and requirements of ASME B31.1. Design temperatures and pressures are the maximum ones expected in the system (consult appropriate heat balance diagrams and system descriptions).

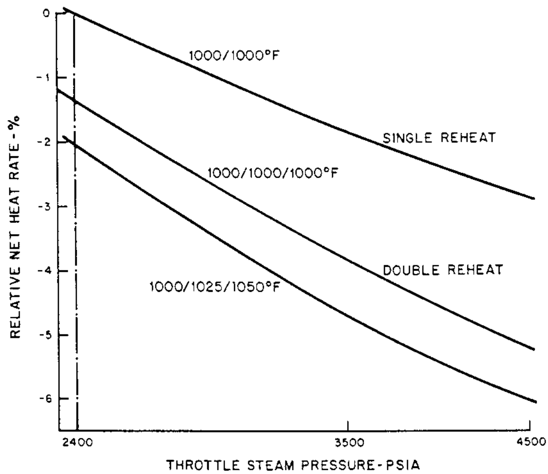


**FIGURE C3.7** Effect of steam conditions on heat rate.

***Extraction Steam Systems.*** The extraction steam system is designed to ASME B31.1. The extraction line design pressure and temperature depend upon the specific extraction point from the turbine as shown on the full-load heat balance diagram at valve wide open (VWO) and 5 percent overpressure. Extraction steam is typically superheated steam except for the lowest extraction points, which are in the wet steam region.

Extraction line sizing is determined by performing an economic analysis of pressure drop, pipe costs, and so on, with a maximum velocity limit of 15,000 fpm (250 fps) (4570 m/min, 76 m/s) for superheated steam. See "Determining Reasonable Flow Velocity" for more flow velocity data. Since extraction steam piping affects heat rate and output of a plant, this piping is sized so that pressure drop does not exceed about 5 percent of turbine stage pressure for the low-pressure lines and 3 percent for the higher-pressure lines. However, these pressure drops should be as low as practical, especially those related to the higher-pressure heaters. Also, keep them low to avoid erosion/corrosion problems.

***Design Temperature.*** Historical development of throttle parameters reveals that in the field of throttle temperatures certain stabilization was reached, while the throttle pressures are continuously rising and the condenser back-pressure is steady at approximately 1 psia (7 kPa). The effect of steam-turbine throttle conditions on overall power plant efficiency is shown in Fig. C3.8. The results shown in Fig. C3.8 indicate that the efficiency increases by 1.5 to 2.5 points as pressure increases from 2400 to 5000 psia (16.6 to 34.5 MPa) at constant throttle and reheat temperature of 1000 to 1400°F (538 to 760°C). It is also seen that the 3500 psia (24.1 MPa) plant efficiency increases by nearly four points as the throttle temperature is increased from 1000 to 1400°F (538 to 760°C).

Some utilities have installed units using supercritical steam pressure; that is, pressures higher than 3206 psia (22.1 MPa), at which point the specific volumes of steam and water are equal. Thus, the main steam pressure for larger units lies between, say, 1450 psia (10.0 MPa) and the supercritical region. Main steam tempera-

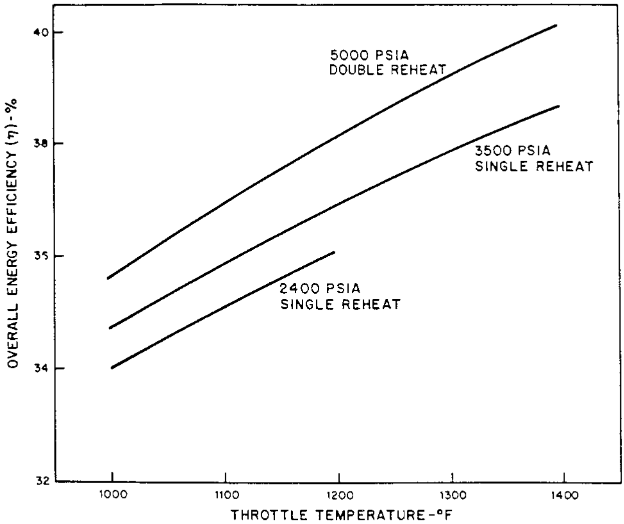


**FIGURE C3.8** Effect of steam turbine throttle pressure and temperature on overall efficiency of a power plant.

ture may then range up to and beyond 1050°F (566°C), necessitating the upgrading of main steam line material as the temperature increases, as noted in the following recommendations:

Up to 775°F (413°C)—Carbon steel.

Up to 950°F (510°C)—Use 1¼ Cr.

From more than 950 to 1050°F (510 to 566°C)—Use 2¼ Cr.

From 1000 to 1200°F (540 to 650°C)—Consider using 9–12 Cr such as P91 as discussed in "Design Features, Pipe Size and Materials" later in this section.

From 1050 to 1200°F (566 to 650°C)—Consider using austenitic stainless steel.

Beyond 1200°F (650°C)—Use austenitic stainless steel.

***Weight Effects.*** See Chap. B2 of this handbook for consideration of weight effects.

***Thermal Expansion and Contraction Loads.*** Thermal expansion of pipelines can be compensated for by use of pipe bends, offsets, or expansion loops, or changes in direction of the pipeline itself. Where pipe bends or offsets can be used or where the pipeline direction is changed to provide for expansion, the provisions of the appropriate codes must be followed as required to ensure that all expansion stresses are within the applicable code limits.

***Dynamic Effects.*** Consider transient effects for turbine trip and safety and relief valve discharge early in the design schedule so that pipe supports and structural steel are adequate to withstand dynamic loads. Typical turbine trip loads on pipe segments in a main steam system (Fig. C3.3) range up to 17,500 lb (78,000 N) after the main steam stop valve closes in approximately 100 milliseconds.[11] Corresponding

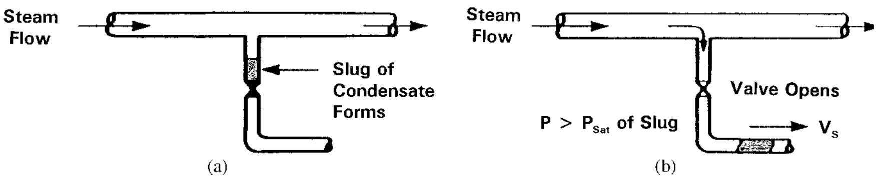


**FIGURE C3.9** Steam-propelled water slug: (*a*) slug forms in closed drain line; (*b*) rapidly moving slug loads pipe due to impact on elbows or restrictions.

reheat steam system (Fig. C3.4) pipe segment loads range up to 13,200 lb (58,700 N) after the reheat intercept valve simultaneously closes in approximately 150 milliseconds.

Another significant dynamic event to be considered in steam piping is steam-propelled water slug,[12] as shown schematically in Fig. C3.9. When a drain valve opens, an upstream slug of condensate could be propelled through the drain piping, causing large loads at each bend until the slug dissipates or exits the piping. Condensate in nearly horizontal steam lines may be swept into slugs, which could also result in large steam-propelled water slug forces, particularly when initiating steam flow into cold piping. The recommended method to prevent steam-propelled dynamic events is to eliminate condensate in steam lines by proper line sloping and drainage, and by carefully warming up cold steam lines.

Also, nozzle design loads for safety and relief valve discharges should be considered. See Chap. B8 in this handbook for more information about transient flow analysis.

## Design Features

***Pipe Size and Materials.*** Specific materials for components used in steam piping systems must satisfy the requirements of the applicable codes. Any standards or specifications quoted in this chapter are minimum requirements. Construction at least equal to that required by the applicable codes is mandatory. The following discussion for piping and fittings is generally acceptable for American National Standards Institute (ANSI) and may be used as a guide. However, other codes and standards, including Deutche Industrie Norm (DIN), Japanese Industrial Standard (JIS), and many others such as British, Canadian, Norwegian, and Swedish codes, must be considered for worldwide steam piping applications. See Apps. E2 and E3 for pipe and tube properties, and see App. E6 for ''International Piping Material Specifications.'' Appendix E5 lists ASTM/ASME material specifications acceptable for use for the ASME B31, Pressure Piping Code, and the ASME Boiler and Pressure Vessel Code, Sections I and III. Refer to App. E10 for international codes and standards.

Steel pipe, either seamless or welded, is generally used in steam systems, although the piping codes permit a variety of materials and several types of welded pipe. For welding and bending, a carbon seamless steel pipe is recommended. Seamless or electric-resistance welded steel pipe A53, Grade B and seamless steel pipe A106, Grade B are popular selections.

Adjusted ratings at temperatures above and below 750°F (399°C) for carbon and alloy steels and other alloys are given in the standards to govern their use under pressure or temperature other than the primary service ratings.

Selection of materials for temperatures above 750°F (399°C) from the various grades of alloys described in ASTM Specifications for high-temperature service is facilitated by reference to specific standards. The multiplicity of services in a large plant and the variety of dimensional standards and materials, possible joints, and different types of welding available make it desirable to provide proper standards, design details, and materials selection guidelines.

Main steam line sizing is determined by performing an economic analysis of pressure drop, pipe cost, and so on, with a maximum velocity limit of 15,000 fpm (250 fps) (4570 m/min, 76 m/s). See ''Determining Reasonable Flow Velocity'' for more flow velocity data. Preliminary pipe sizing is often based on approximately a 3 percent pressure drop from the superheater outlet to the turbine stop valves. Wall thickness is calculated using the equations presented in ''Design Pressure.'' Thus the main steam piping between the turbine and boiler may consist of one or more lines, with metallurgy varying from pressure-temperature rating requirements.

Because of the high cost of alloy piping, the selection of its size is usually the subject of an economic study where the increased pressure drop and its effect on turbine output are weighted against unit pipe cost including installation and hangers. For systems operating at or below 1050°F (566°C), chrome-moly alloy steel is often used in order to bring the pipe thickness down to an economically acceptable value for the steam pressures used.

Main steam piping at 1015°F (546°C) is typically 2¼ percent chrome, 1 percent molybdenum steel to ASTM A335, Grade P22 in U.S. fossil power plants (see ''Design Temperature''). However, there have recently been extensive investigations of using higher levels of chrome, ranging up to 12 percent. There is a gap between the low- and medium-alloyed ferritic steels on one side and the high-alloyed austenitic steels on the other, as shown on Fig. C3.10.[13] This gap in the creep rupture strength versus temperature diagram is occupied by the group of ferritic/martensitic 9–12 percent chromium steels. Compared to P22 (10 CrMo 9 10), the creep rupture strength of P91 is even twice as high.[14] Several of these 9–12 percent chromium steels have been developed and used worldwide,[13] including X 12 CrMo 9 1, which has been used as steel under hydrogen pressure for tubes and pipes mainly in chemical plants, and X 20 CrMo 12 1, which has been used with great success in power plant piping, tubing, and headers. A similar steel, X 10 CrMoVNb 9 1, developed in the United States, has been standardized in ASTM A213 for tubing (T91), in ASTM A335 for piping (P91), and in ASTM A336 for forging (F91). Parallel developments include the Japanese steel NF 61615.[15]

For the given initial steam conditions, power plant capacity, pipe material, and economic factors, the number of main steam leads should be optimized considering the steam generator manufacturer's header arrangement and the turbine valve arrangement. Selection of the operating steam pressure and temperature for a turbine-generator installation is based on an economic study or on experience derived from previous installations. Studies have shown that higher pressures and temperatures are associated with larger units or with higher expected fuel costs.

Main steam piping for the PWR and BWR is typically carbon steel to ASTM A106, Grade B or C.

In practical applications, steam generator and main steam line material limitations, plant reliability, and economic considerations influence the initial steam conditions. For a given pipe, an increase of initial steam temperature above certain values requires lower initial pressure. The maximum allowable stress values in tension for pipe materials for various metal temperatures may be obtained from the appropriate codes being used for design of the piping.

If better materials are used for main steam (MS) pipes, steam generators, and turbines, additional steam reheat may be beneficial, since this will allow higher throttle pressure at fixed throttle steam temperature. This additional steam reheat

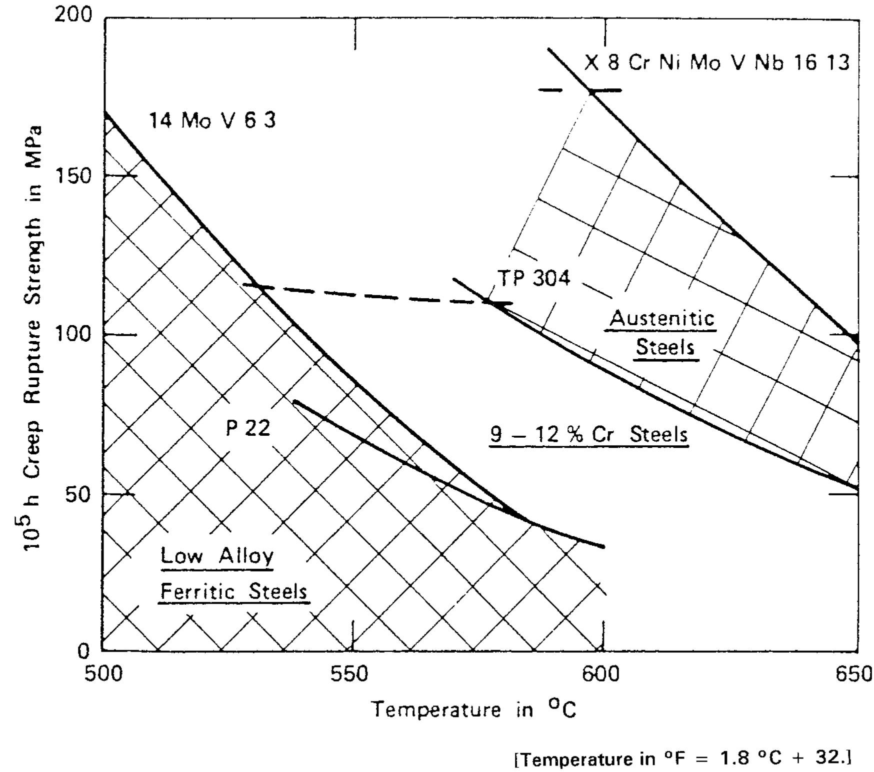


**FIGURE C3.10** Creep rupture strength of heat-resistant ferritic and austenitic steels. *(From Ref. 13.)*

not only guards a turbine against operation in a high moisture region, but also increases fuel savings due to the carnotization of a Clausius-Rankine Cycle. See "Understanding the Extraction Steam System."

The high-pressure main steam piping must be designed with attention to the need for adequate size (pipe diameter), thermal expansion, piping supports, and drain of condensate, with the last factor being especially important when warming up the system preparatory to a start.

Pressure drop in steam lines to turbines and heat losses to the surroundings have an important influence on power plant performance. Long runs of pipe, or shorter runs of expensive, thick-walled, alloy steel, high-pressure pipe create the need for an economic determination of optimum pipe size. The higher the steam velocity, the smaller the required size of pipe, but unfortunately the frictional pressure loss increases as the square of the velocity. In an economic analysis, the higher capital cost of larger pipe diameters, including insulation and support, must be weighed against the lower operating costs resulting from better plant performance. For more information on the economic optimization of line sizes, see Chapter B8 in this handbook. The amount of insulation to apply is a separate economic consideration and is also governed by the maximum allowable surface temperature for personnel protection.

***Piping Joints.*** To a large extent, pressure piping is welded. Welded joints have generally replaced most of the screwed or flanged joints in new construction of steam distribution system mains and customers' service lines. Joint construction must satisfy the requirements of the code applicable for construction.

Although welding has largely displaced flanged joints, some flanged connections are nevertheless required, particularly in making connections to flanged valves, expansion joints, or fittings where space limitations do not permit welding, or where easy removal of a fitting or valve is desired. Malleable, cast-iron, bronze, or brass fittings may be used for pressures not exceeding 250 psi (1725 kPa) and temperatures not in excess of 450°F (232°C). Cast or forged carbon steel fittings are required for pressures above 250 psi (1725 kPa). For temperatures above 775°F (413°C), carbon steel is not recommended. Welded fittings must comply with the American Standard for Factory-Made Wrought Steel Buttwelding Fittings (ASME B16.9[6]) or American Standard for Forged Steel Fittings, Socket-Welding and Threaded (ASME B16.11[16]) where applicable, and the material shall conform to ASTM Specification A216, A234, or A105. Special fittings or welded assemblies fabricated in either the shop or field are required to conform to the requirements of the appropriate codes.

***Gaskets.*** Gaskets may be made of metal or other material which will not burn, char, or change in character so as not to perform the service intended. Asbestos is prohibited. Several alternatives for asbestos, including spiral-wound gasket applications, have recently been developed by the gasket manufacturers. A summary of material specifications for selected lines for a 150 MW circulating fluidized bed power plant is given in Table C3.4.

***Threaded Joints.*** Limitations for threaded joints are contained in both the ASME Code, Section III[1] and ASME B31.1.[2] Paragraph 114 in B31.1 includes the following guidance, which is similar to that in the ASME code for Class 2 and 3 piping (threaded joints are not allowed in ASME Class 1 piping): All threads on piping components shall be taper pipe threads in accordance with the applicable Standards listed in Table 126.1.[2] Threads other than taper pipe threads may be used for piping components where tightness of the joint depends on a seal weld or a seating surface other than the threads, and where experience or test has demonstrated that such threads are suitable.

Threaded joints shall not be used where severe erosion, crevice corrosion, shock, or vibration is expected to occur, nor at temperatures over 925°F (495°C). Size limits for steam and hot water service [above 220°F (105°C)] shall be as listed in Table C3.5. Threaded access holes with plugs, which serve as openings for radiographic inspection of welds, are not subject to the limitations of Table C3.5. Threaded connections for insertion-type fluid temperature determination and sampling devices are not subject to the temperature limitations stated in Table C3.5. At temperatures greater than 925°F (495°C) or at pressures greater than 1500 psi (10,350 kPa), these threaded connections shall be seal-welded. The design and installation of insertion-type fluid temperature determination and sampling devices shall be adequate to withstand the effects of the fluid characteristics, fluid flow, and vibration.

Paragraph 104.1.2(C) of B31.1[2] states that "Where steel pipe is threaded and used for steam service at pressures above 250 psi (1750 kPa) or for water service above 100 psi (700 kPa) with water temperatures above 220°F (105°C), the pipe shall be seamless having the minimum ultimate tensile strength of 48,000 psi (330 MPa) and a weight at least equal to Schedule 80 of ANSI B36.10M.[4]

**TABLE C3.4** Material Specification for Selected Lines (150 MW Circulating Fluidized Bed Power Plant)

| | SYSTEM | | | | | |
|---|---|---|---|---|---|---|
| Item | Main steam | Hot reheat | Cold reheat first point extraction | Second point extraction | Condensate pump discharge; fourth, fifth, sixth point heater outlet | Third-sixth point extraction, raw water, service water, other condensate piping |
| Design pressure and temperature | 1985 psig, 1015°F | 590 psig, 1015°F | 615 psig, 710°F | 210 psig, 785°F | 330 psig, 285°F | 110 psig, 650°F or less |
| Pipe size, schedule | 16.1 in OD, 2.10 $t_{min.}$<br>11.7 in OD, 1.51 $t_{min.}$ | 22 in Sch 80 | 24 in Sch 60,<br>8 in XS | 8 in Std | Over 2 in, Std<br>Up to 2 in, Sch 80 | Over 2 in, Std<br>Up to 2 in, Sch 80 |
| Pipe material | A335 P22 | A335 P22 | A106 Gr B | A335 P22 | A106 Gr B | A53 Gr B |
| Pipe construction | Seamless | Seamless | Seamless | Seamless | Seamless | Seamless |
| Flange type | Not allowed | Welding neck | Welding neck | Welding neck | Welding neck | Slip-on or welding neck |
| Flange material | N/A | A182 F22 | A105 | A182 F11 | A-105 | A105 |
| ANSI standard | N/A | B16.5 | B16.5 | B16.5 | B16.5 | B16.5 or MSS-SP-44 |
| Class | N/A | 900 RF | 600 RF | 300 RF | 300 RF | 150 RF |
| Gaskets | Not allowed | Spiral-wound | Spiral-wound | Spiral-wound | Spiral-metal | * |
| Fittings over 2 in | | | | | | |
| ASTM spec. | A234 WP22 | A234 WP22 | A234 WBP | A234 WP11 | SA234 WPB | A234 WPB, or A216 WCB |
| ANSI std. | B16.9, B16.28 | B16.9, B16.28 | B16.9, B16.28 | B16.9, B16.28 | B16.9, B16.28 | B16.9, B16.28 |
| Type | Butt weld | Butt weld | Butt weld | Butt weld | Butt weld | Butt weld, or flanged |
| 2 in and smaller | | | | | | |
| ASTM spec. | A182 F22 | A182 F22 | A105 | A182 F11 | SA-105 | A-105 |
| ANSI | B16.11 | B16.11,<br>MSS-SP-79 | B16.11,<br>MSS-SP-79 | B16.11,<br>MSS-SP-79 | B16.11,<br>MSS-SP-79 | B16.11,<br>MSS-SP-79 |
| Rating | 9,000 Class | 3,000 Class | 3,000 Class | 3,000 Class | 3,000 Class | 3,000 Class |
| Type | Socket weld | Socket weld | Socket weld | Socket weld | Socket weld | Socket weld |

* Spiral-wound if over 180°F; red rubber J-M Style 107, or equal if below 180°F; Vellumoid if oil.

**TABLE C3.5** Threaded Joint Size Limits for Steam and Hot Water Service Above 220°F (105°C)[1,2]

| Maximum nominal pipe size | Maximum pressure | |
|---|---|---|
| | psi | kPa |
| 3 | 400 | 2,750 |
| 2 | 600 | 4,150 |
| 1 | 1200 | 8,300 |
| ¾ and smaller | 1500 | 10,350 |

***Note:*** For instrument, control, and sampling lines, refer to Para. 122.3.6.[2]

Paragraph 102.4.2 of B31.1[2] states that "The calculated minimum thickness of piping (or tubing) which is to be threaded shall be increased by an allowance equal to thread depth; dimension *h* of ASME B1.20.1 or equivalent shall apply. For machined surfaces or grooves, where the tolerance is not specified, the tolerance shall be assumed to be 1⁄64 in (0.40 mm) in addition to the specified depth of cut."

***Cast-Iron Flanges.*** It is advisable to express a word of caution regarding connecting cast-iron flanges and fittings to steel flanges and fittings. Class 25 and Class 125 cast-iron flanges have plain faces, while the Class 250 flanges have raised faces which come out practically to the inner edge of the bolt holes whereas those of steel flanges are narrower and stop some distance inside the bolt holes. The plain face or wide raised face on cast-iron fittings and flanges is a necessary precaution to prevent cracking the flange in drawing up the bolts. Numerous instances have been observed where cast-iron flanges have cracked when being bolted to raised-face steel flanges. In cases where it is necessary to bolt cast-iron and steel flanges together, the raised face of the steel flange should be machined down flush with the flange edge. For the same reason, when lap-joint pipe is made up with cast-iron flanges, the lapped end should be brought out to the inner edge of the bolt holes. These type of flange joints should only be used where permitted by the governing code.

For reasons similar to those just stated, it is inadvisable to use alloy steel bolts in cast-iron flanges. Commercial carbon-steel bolts (A307, Grade B) are amply strong for use in cast-iron flanges up to a temperature of 400°F (205°C), and there is no occasion to risk cracking such flanges through the use of high-strength bolts. Steel flanges are properly designed, both as regards dimensions and material, for use with a narrow raised face and alloy-steel bolts.

***Valves.*** Valves must be of a design at least compatible with the service conditions and constructed of the materials allowed by the appropriate codes for design pressure and design temperature.

ANSI pressure class determination should be based on the design and operating conditions of the system (i.e., temperature, pressure). It should be noted that for valves with elastomeric or plastic gaskets, packing, or seating elements, the valves may not meet the entire range of pressure/temperature conditions for their designated ASME pressure class. See Chap. A10 for selection and application of valves.

For power work, the following codes give pressure/temperature tables for valves:

ASME B16.34—"Valves-Flanges, Threaded, and Welding End"[17]
API602—"Compact Steel Gate Valves, Fifth Edition"[19]

When selecting a valve, consider the following:

- The system safety category
- The design and operating conditions of the system, or portion of the system in which the valve will be installed
- Valve function; that is, isolation, throttling, or modulating
- Radioactivity level, if any
- Whether pressure drop is critical
- Preferred end connection; that is, socket weld, butt weld, threaded, flanged, or mechanical
- Special features, including minimum operating time, stem leakoff required, exceptional tightness required for seat and/or seals, body taps
- Material requirements; that is, alloy, carbon steel, hard-facing, and so on

***Isolation and Control Valves.*** Selecting the proper valve for a particular purpose depends on the operating pressure and temperature and on the type of valve best-suited for the use to which it will be put. In general, it is customary to use *gate valves* in locations where pressure drop through the valve is a consideration, and where the valve will either be wide open or fully closed. Guard valves and shutoffs for boiler and turbine leads, and so on, are almost always of the gate type. *Globe valves* are commonly used in water, steam, and air lines for control or throttling purposes, as the globe type permits closer regulation of flow. Throttling often involves some steam cutting of the seat and disk, and these parts of globe valves are more easily repaired or replaced than in gates. Some appropriate uses for globe valves under these conditions are turbine and engine throttles, bypasses around traps or reducing valves, and hand-feed regulation on boilers.

*Check valves* are required in feed lines close to a boiler to prevent water or steam from blowing back from the boiler, if, for any reason, the feed line ruptures or its pressure fails. It is advisable to use check valves in individual pump or trap discharges before they join a common header and where different lines are joined together to discharge into a common header. A check valve cannot be counted on for closing a line off completely against pressure working back through, but it will stop a considerable flow. In pump discharges where the header remains under pressure after the pump is shut down, a gate valve should be provided in addition to the check valve. It is also desirable to provide a small relief valve on the pump suction to prevent pressure backing up through the pump and overpressurizing the suction side of the system pump.

With pressure-reducing or other control valves it is desirable to select a size that is loaded somewhere near capacity under normal operation, as such valves are then more stable in their operation. If there is considerable seasonal variation in the load on a reducing or control station, it is good practice to install a large and a small valve in parallel, and use the one best fitting the load at any particular time. It is frequently desirable to install a hand-operated bypass around a control valve so that service can be maintained while the special valve is being repaired. Such an arrangement with provision for shutoff to repair the special valve while the line

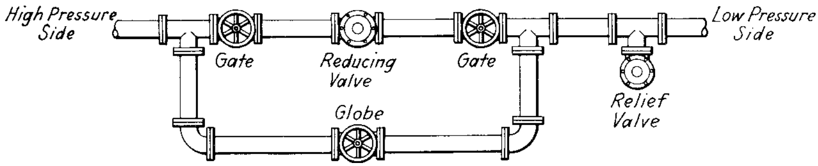


**FIGURE C3.11** Pressure-reducing station.

is in operation is shown in Fig. C3.11, which includes flanged valves. Advantage can be taken of improved valve design, wherein weld ends are used while still permitting the internals to be removed for repairs. Main steam lines usually do not use flanges. Filler pieces should be provided between a flanged reducing valve and the adjacent gates, as reducing valves usually are constructed so it is impossible to remove all the end-flange bolts when they are bolted directly to other valves or fittings because of close clearance at the valve bonnets or fitting necks. For convenience in removing these valves from the line, flanged connections are used frequently at such points, even though the rest of the line is made with screwed or welded joints. When reducing from a pressure that requires the use of a heavy standard for flanges and fittings to a pressure with which a lighter standard is used or to which low-pressure equipment is connected, relief valves should be provided on the low-pressure side. The use of the heavier standard should be continued through to the last valve ahead of the relief valve, as it is possible to have full pressure up to that point. A reducing valve seldom has a tight shutoff, and at times of negligible steam consumption, leakage through the valve may be enough to build up full line pressure on the low-pressure side. The ASME Code for Power Piping requires that the combined discharge capacity of the safety or relief valves shall be such that the pressure rating of the lower-pressure piping will not be exceeded in case the reducing valve sticks open.

***Safety and Relief Valves.*** Safety and relief valves are defined in Section I of the ASME Code[20] as follows:

> Safety Valve: An automatic pressure relieving device actuated by the static pressure upstream of the valve and characterized by full-opening pop action. It is used for gas or vapor service.
>
> Relief Valve: An automatic pressure relieving device actuated by the static pressure upstream of the valve which opens further with the increase in pressure over the opening pressure. It is used primarily for liquid service.
>
> Safety Relief Valve: an automatic pressure-actuated relieving device suitable for use either as a safety valve or relief valve, depending on application.

The construction and method of installing safety valves for power boilers are explained in Section I of the ASME Boiler and Pressure Vessel Code.[20] The boiler safety valve requirements are excerpted as follows:

> Each boiler shall have at least one safety valve or safety relief valve and if it has more than 500 ft$^2$ [46 m$^2$] of bare tube water-heating surface, or if an electric boiler has a power input more than 1100 kW, it shall have two or more safety valves or safety relief valves. For a boiler with combined bare tube and extended water-heating surface exceeding 500 ft$^2$ [46 m$^2$], two or more safety valves or

> safety relief valves are required only if the design steam generating capacity of the boiler exceeds 4000 lb/hr [1810 kg/hr]. . . .
>
> The safety valve or safety relief valve capacity for each boiler shall be such that the safety valve, or valves will discharge all the steam that can be generated by the boiler without allowing the pressure to rise more than 6 percent above the highest pressure at which any valve is set and in no case to more than 6 percent above the maximum allowable working pressure. . . .
>
> One or more safety valves on the boiler proper shall be set at or below the maximum allowable working pressure. If additional valves are used the highest pressure setting shall not exceed the maximum allowable working pressure by more than 3 percent. The complete range of pressure settings of all the saturated-steam safety valves on a boiler shall not exceed 10 percent of the highest pressure to which any valve is set. Pressure setting of safety relief valves on high-temperature water boilers may exceed this 10 percent range.

Additional requirements and requirements for alternate protection against overpressure are included in Section I of the ASME Boiler and Pressure Vessel Code,[20] Section PG-67.

Where more than one safety or relief valve is used on a boiler or other pressure vessel, it is desirable to set one or more of the valves to relieve at a lower pressure than the rest. This serves as a warning before too much steam is lost through all the valves opening at once and also tends to facilitate repairs by confining any cutting action to the one or more valves that open first. In some cases an extra safety valve, known as the power-control valve, is set to blow before the others and is mounted above a gate valve so that it can be removed for repairs while the boiler or steam line is in service. The capacity of this valve cannot be considered in meeting Code or other safety requirements, because it might be shut off. Where the hazard involved does not require the installation of a full-size relief valve, it is sometimes desirable to install a small-size pop valve as a telltale to give warning when the usual working pressure is exceeded. The operator can then attend to restoring it to normal conditions. A safety valve for use with a compressible fluid, such as steam or air, is distinguished from a relief valve in that a safety valve has an adjusting, or huddling, ring and chamber to control the amount the pressure blows down before the valve reseats.

***Bolting.*** In bolting cast-iron flanges or steel flanges to cast-iron flanges, valves, fittings, and so forth, bolts must be of carbon steel equivalent to ASTM A307, Grade B, without heat treatment other than stress relief. Otherwise, they may be too strong for the cast-iron flanges. Threads in accordance with the coarse-thread series of the standard for screw threads, ASME B1.1 and B18.2.1,[21,22] are recommended for carbon-steel bolts. Carbon-steel bolts conforming to A307 may be the standard regular or heavy hexagonal-head bolts and must be used with standard heavy semifinished hexagonal nuts, B18.2.2,[23] which conform to ASTM A194.

For high-temperature service or for insurance of a tight joint in the case of steel flanges, bolts or stud bolts should be of alloy steel, conforming to ASTM A193, typically Grade B7. Nuts must be of steel according to ASTM A194, Grade 2H.

Bolt and nut selection for use with flanges in power piping may often be made from the alloy-steel bolts listed in Tables 112 and 126.1 and Section 108.5 of ASME B31.1.[2] See Chap. A7 in this handbook for additional information on bolted joints.

Following is a summary of the principal bolting selections by the architect/

constructor for a typical 150 MW circulating fluidized bed power plant (see Table C3.4):

- *Bolt studs:* Eight threads per in (25.4 mm) [coarse thread for less than 1 in (25 mm)] with two nuts.
- No bolted joints are allowed for main steam.
- *Stud material:* A193, B16 for hot reheat, if present; A193, B7 for all other systems noted.
- *Nuts:* Hexagon semifinished Heavy Series per B18.2.2.[23]
- *Nut material:* A194, Grade 4 for hot reheat; A194, Grade 2H nut standard for all other systems noted.

***Protective Conduits for Underground Lines.*** Protective conduits for underground steam mains and services are necessary to protect the pipe and insulation from damage due to earth pressure and impact loadings; to allow free longitudinal expansion and contraction while held in proper alignment; and to prevent groundwater seepage or flooding by providing either drains or a completely waterproof structure.

Many types of conduits are used by the steam-piping industry today. They can be categorized into the following general classifications: prefabricated, boxtype, solid pour, granular fused, and tunnels.

*Prefabricated conduits* are popular with utilities located in large cities, where conditions exist such as congested subsurface, heavy surface traffic, tidewater, and rock areas. The ideal design for these conditions is one with the smallest cross-sectional area, consistent with thermal requirements, to fit into limited subsurface space. Two prefabricated designs are shown in Fig. C3.12.

In Fig. C3.12*a*, the insulated pipe is placed within a corrosion-resistant, helically corrugated metal jacket which is protected with a heavy asphaltic coating. In Fig. C3.12*b*, the insulated pipe is placed concentrically within the corrosion-resistant metal jacket, and the intervening space between jacket and insulation is poured full of high-melting-point asphalt, which is a protective medium for the insulation. Some of these prefabricated designs are also available with cast-iron casings with fittings designed for joints, bends, and so on.

*Boxtype conduits* have many variations. Depending on loading conditions, reinforcement may or may not be necessary. Drainage is provided by filling drainage pockets with crushed stone or gravel; these pockets may be installed on either or both sides of the conduit. They conduct water from the top of the conduit to the lower drain, which is connected to a sewer.

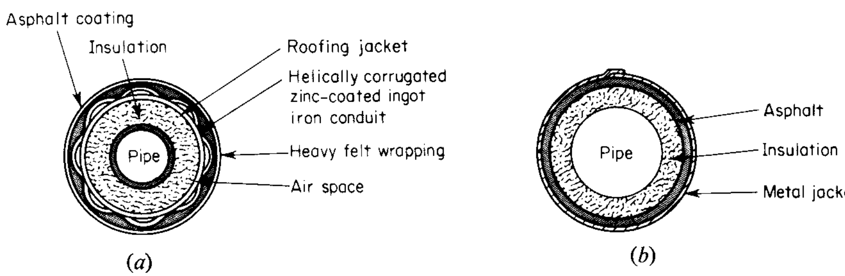


**FIGURE C3.12** Prefabricated conduits for underground steam pipes: (*a*) coated and wrapped corrugated conduit; (*b*) poured asphalt protects thermal insulation.

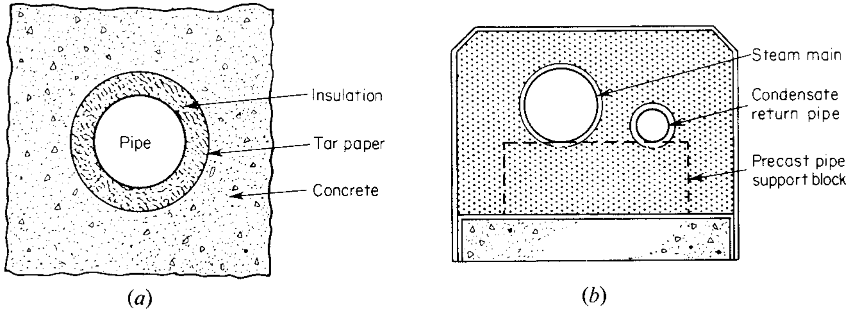


**FIGURE C3.13** Solid-pour protection for underground steam pipe: (*a*) insulated and wrapped pipe in structural concrete; (*b*) pipe is surrounded by insulating concrete.

*Solid-pour construction*, such as is shown in Fig. C3.13*a*, consists of poured structural concrete which is vibrated or tamped around a conventionally insulated steam line. If the insulation has a high compressive strength, such as is characteristic of diatomaceous earth or silica, it can support the pipe; otherwise, it should be supported independently. Reinforcing is sometimes used to prevent settlement. An eccentric space may be left around the pipe by using insulation larger than the outside diameter of the pipe. This permits the pipe to rise during the warming-up period without crushing the insulation.

The design of Fig. C3.13*b* utilizes an insulating concrete as a conduit. The concrete is poured around the steam pipe, which is supported on precast blocks of the same material. The insulating concrete may consist of any mixture of insulation or other cellular materials and portland cement or a mixture of a special foaming material mixed with portland cement; the aim is to create a cellular mass composed of minute air cells in the concrete so as to develop insulating qualities. The piping (which could be a number of pipes) is wrapped with corrugated paper before the insulation is poured, and a heavy-asphalt-coated waterproofing membrane, or equivalent waterproof sheeting, is installed to protect the top and side of the structure before backfilling.

*Granular-fused* types of conduits as shown in Fig. C3.14 consist of a granular

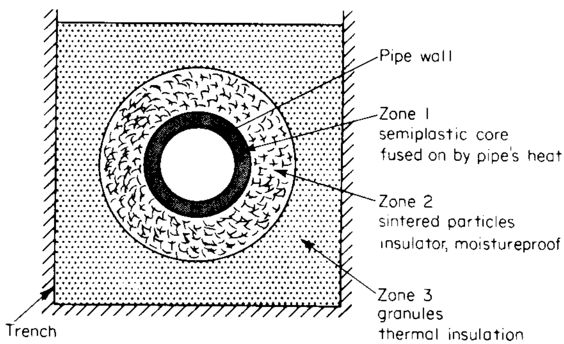


**FIGURE C3.14** Granular-fused conduit for underground steam pipe.

bitumen, selected for the required temperature range and poured and tamped around the pipe or pipes in a trench. Heat passing through the pipe forms three concentric zones: (1) a dense semiplastic core fused on by the pipe's own heat, (2) a sintered zone providing thermal insulation and moisture proofness, and (3) an outer layer of granules providing a final zone of thermal insulation and the load-bearing portion of the structure.

*Walk-through tunnels* are usually not provided for steam mains unless they are required for underground passage between buildings, as in institutions, and then, they are usually constructed to accommodate other utilities services such as electricity, water, and gas. Tunnels are very costly and are constructed only out of necessity.

Provision for condensate drainage, ventilation, and insulating methods to reduce heat loss are considered for all portions of underground steam mains.

The illustrations shown here are only a few of the many designs utilized in practice today. Choice and application depend upon the project design requirements and site conditions. In the economic evaluation, factors to be considered include life expectancy, operation and maintenance costs, impact of geotechnical conditions on foundations, external and internal drainage, corrosive action of soil conditions, loads to be imposed, field installation practices, and necessary insulating properties.

***Drainage of Condensate.*** To prevent a water slug type of water hammer and a possible rupture of the steam main, condensed steam or condensate within the steam main must be removed. Even superheated steam lines need drainage since condensation forms during the warming-up period, or while the line is hot but without flow. The points to be drained are the low points in the line, moisture separators, drip pockets, and valves, especially in vertical lines. Horizontal portions of the steam lines should be pitched downward approximately ⅛ in/ft (10 mm/m) in the direction of flow or ¼ in/ft (20 mm/m) for lines that contain a steam/water mixture or require draining periodically. Condensate flow against the steam should be avoided if possible.

Since the contour of the ground or subsurface structures will dictate the pitch of the steam main, provisions must be made during construction to grade the steam main carefully so as to pitch the pipe adequate for condensate drainage. Since it is unlikely that a continuous slope can be maintained in a long run of main, removal of condensate (drainage) must be provided for at all low points where a water pocket will exist. In any event, recommended lengths of steam main for draining off condensate should not exceed 300 to 400 ft (90 to 120 m).

To provide for condensate drainage from a steam main, a drain pocket is welded to the bottom of the pipe to be drained. The diameter of the pocket should be about one-third the diameter of the line, up to a maximum of NPS 6 (DN 150) for NPS 18 (DN 450) and larger mains. The pocket not only provides for condensate removal but also allows for sediment removal. Figure C3.15 shows this design. Figure C3.16 is a design in the form of a separator, which is more prevalent in underground steam mains.

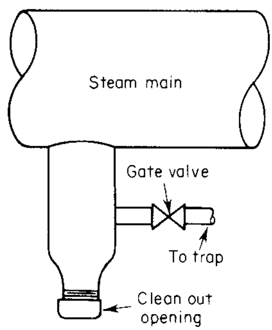


**FIGURE C3.15** Drain pocket for steam-trap connection to low-velocity steam main.

For systems which supply high flow rates to concentrated loads, it probably will be economical to return the condensate in order to recover the energy in the condensate and thereby reduce the costs of treating raw makeup water.

Sometimes, however, it is not economically feasible or practical to return the condensate from district-heating systems because of the high cost of installing and maintaining pumping equipment, manholes, and return piping.

In such cases, traps off the drain pocket can often be discharged through cooling coils to the city sewer system. City regulations may require the condensate discharge to be cooled before entering the sewer. In some cases, condensate discharge lines are piped to sump pits in manholes, where the water is removed by float-controlled electric pumps which discharge it into the sewer. See Figs. C3.17 and C3.18.

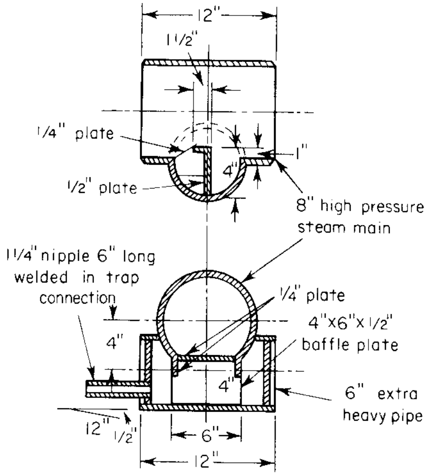


**FIGURE C3.16** Drain pocket for steam-trap connection to high-velocity steam main.

In process and power system piping, condensate removal is an important consideration to prevent water slug formation in steam lines. Paragraph 122.11 of ASME B31.1[2] contains the following provisions for drains, drips, and steam traps:

> Drip lines from piping or equipment operating at different pressures shall not be connected to discharge through the same trap. If two or more traps discharge into the same header which is, or may be, under pressure, a stop valve and a check valve shall be provided in the discharge line from each trap.

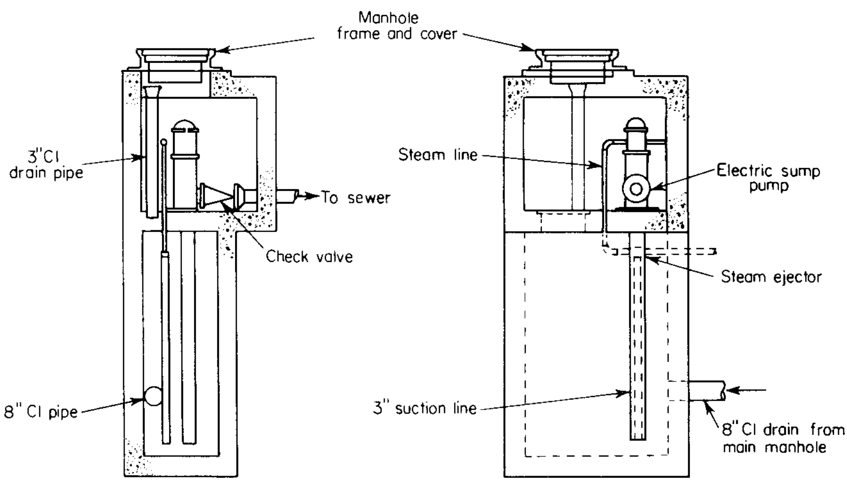


**FIGURE C3.17** Manhole for sump pump.

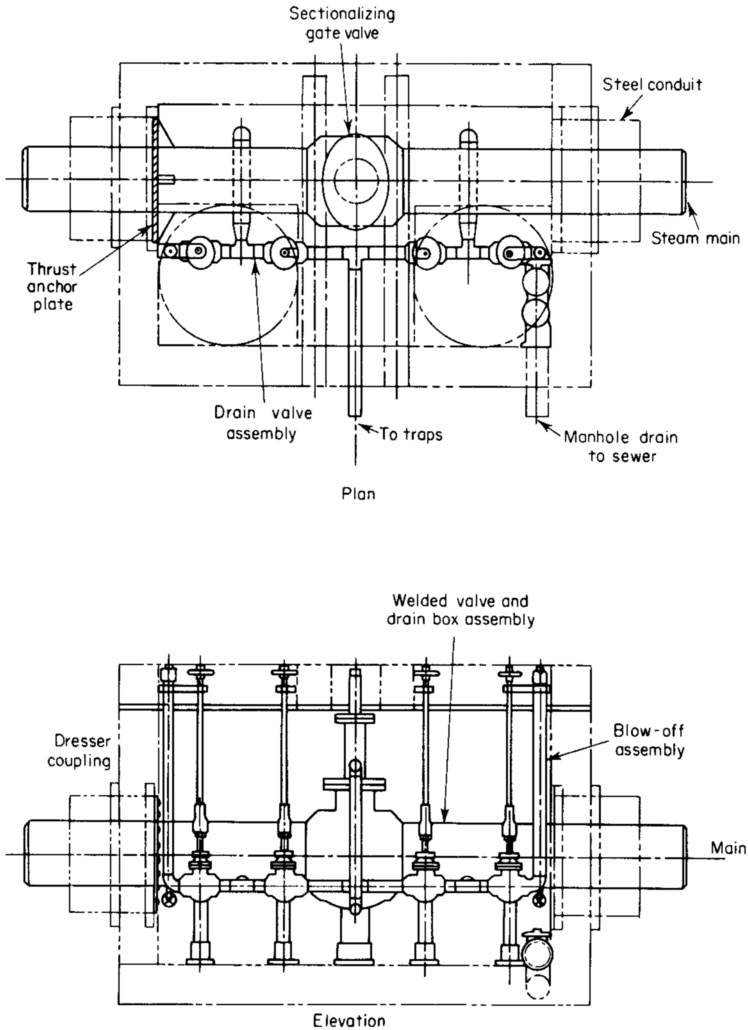


**FIGURE C3.18** Manhole for sectionalizing valve and drain valve assembly.

Trap discharge piping shall be designed for the same pressure as the inlet piping unless the discharge is vented to atmosphere, or is operated under low pressure and has no stop valves. In no case shall the design pressure of trap discharge piping be less than the maximum discharge pressure to which it may be subjected.

*Steam traps* are automatic devices used to trap or hold steam until it has condensed and to allow condensate and air to pass as soon as they accumulate. In general, a trap consists of a vessel in which the condensate accumulates, an orifice through which the condensate is discharged, a valve to close the orifice port, mechanisms to operate the valve, and inlet and outlet openings for the entrance and

**TABLE C3.6** Reasonable Design Velocities for Flow of Fluids in Pipes

| | | | Reasonable velocity | |
|---|---|---|---|---|
| Fluid | Pressure, psig | Use | Ft/minute | Ft/second |
| Water | 25–40 | City water | 120–300 | 2–5 |
| Water | 50–150 | General service | 300–600 | 5–10 |
| Water | 150 up | Boiler feed | 600–1,200 | 10–20 |
| Saturated steam | 0–15 | Heating | 4,000–6,000 | 67–100 |
| Saturated steam | 50 up | Miscellaneous | 6,000–10,000 | 100–167 |
| Superheated steam | 200 up | Large turbine and boiler leads | 10,000–20,000 | 167–334 |

***Source:*** Courtesy of Stone & Webster.

discharge of the condensate from the trap vessel. Steam traps are discussed in detail in Chap. A2.

***Determining Reasonable Flow Velocity.*** Before proceeding beyond a preliminary layout of piping systems, it is necessary to determine pipe sizes which allow reasonable velocities and friction losses. The maximum allowable velocity of the fluid in a pipeline is that which corresponds to the permissible pressure drop from the point of supply to the point of consumption or is that which does not result in excessive pipeline erosion. The economic optimization of line sizes is discussed in Chap. B8 in this handbook. The values of velocity listed in Tables C3.6 and C3.6M (Metric) are reasonable for use in such cases. The lower velocities should be used for small pipes, and the upper limits for large ones. These values represent good average practice and may be used as a guide in many cases where actual pressure drops are not computed. Additional steam line velocity information is given in Chap. B8 in this handbook.

Erosive action on valve seats and similar exposed parts also affects permissible velocity. This action is much more pronounced in the case of wet steam than with superheated steam, and velocities should be correspondingly lower when there is much moisture in the steam.

High velocities are sometimes used in dry steam lines where excess pressure exists, to absorb the higher pressure drop. The high velocity is not in itself objection-

**TABLE C3.6M (Metric)** Reasonable Design Velocities for Flow of Fluids in Pipes

| | | | Reasonable velocity | |
|---|---|---|---|---|
| Fluid | Pressure (kPa gauge) | Use | m per minute | m per second |
| Water | 172–276 | City water | 36–91 | 0.61–1.52 |
| Water | 345–1034 | General use | 91–183 | 1.52–3.05 |
| Water | 1034 up | Boiler feed | 183–366 | 3.05–6.10 |
| Saturated steam | 1–103 | Heating | 1220–1830 | 20.4–30.5 |
| Saturated steam | 345 up | Miscellaneous | 1830–3050 | 30.5–50.9 |
| Superheated steam | 1380 up | Large turbine and boiler leads | 3050–4570 | 50.9–76.2 |

***Source:*** Courtesy of Stone & Webster.

able, since with dry steam there is no appreciable erosion of the pipe walls and no undesirable effects save that high velocities are accompanied by considerable noise (which would be objectionable in heating systems in office buildings and dwellings but may not be in power plants).

In the selection of pipe sizes for exhaust lines from auxiliary turbines and similar services where high velocities may be used to advantage, consideration should be given to the limiting velocities which can be obtained. This limiting or sonic velocity has a definite value for each combination of steam density and pressure (see Chap. B8).

It is often cheaper to use a larger standard pipe size than to use a unique pipe size. Table C3.4 shows a special size for main steam piping and standard sizes recommended for the other systems. Where space is at a premium, it may be necessary to use a smaller size than would otherwise be good practice. The calculation of velocity or pressure drop is a valuable check, but in the last analysis judgment is the deciding factor, and the blind use of generalized criteria is impracticable.

Special attention should be given to sizing hot and cold reheat piping since pressure drop in reheat systems affects turbine heat rate. Cost of piping must be compared with these effects to arrive at the most economical piping arrangement. Normally, total pressure drop in reheat piping and the reheating section of the steam generator should be 7 to 9 percent of high-pressure turbine exhaust pressure. It is desirable to use a smaller-diameter hot reheat line and larger-diameter cold reheat line, taking a greater pressure drop in the more expensive (alloy) hot reheat line.

***Loss Due to Steam Leaks.*** Modern power plants with almost all welded joints experience leakage through only a few nonwelded joints at valves or other equipment connections. The hazard of high-pressure, high-temperature steam leaks, together with the cost of chemicals required in most makeup systems, makes early repairs mandatory.

Existing low-pressure plants using flanged or screwed joints experience considerable leakage and where maintenance is deferred are wasting both water and energy at a rate which depends on the size of the opening and steam pressure.

The value of steam which can be lost through a comparatively small leak becomes appreciable when considered over a period of time. The amounts of steam which will escape through various sizes of orifices at different pressures can be computed by an appropriate flow formula. Table C3.7, which was computed by Grashof's formula,[24] gives the pounds of steam and gallons of water wasted per month, while Table C3.7M (Metric) gives the kg of steam and $m^3$ of water wasted per month.

***Expansion, Flexibility, and Supporting.*** Chapters B4 and B5 in this handbook provide basic information on the thermal expansion, the needed flexibility, and the supporting of piping systems. In underground steam piping, expansion joint fittings are more commonly used for thermal expansion owing to the limitations of space and costly trenching. Expansion joints are often used for application in underground steam mains.

The basic design of a *slip-type joint* consists of a cast-iron or steel body with a stuffing box and a sliding sleeve. The advantage of slip-type joints is the longer traverses that can be obtained to absorb pipe expansion; the disadvantage is the necessity for maintaining packing. By the very nature of its construction, the slip joint is capable of absorbing only axial movement.

The *metallic bellows expansion joint* is composed of the following components: (1) the flexible element proper (corrugated metal tube), the rings enabling reinforced

**TABLE C3.7** Loss Due to Steam Leaks*

| Size of orifice, in | Pounds steam wasted per month | Water wasted, gal |
|---|---|---|
| 250-lb gauge | | |
| ½ | 1,780,000 | 213,600 |
| ⅜ | 1,001,000 | 120,100 |
| ¼ | 445,000 | 53,400 |
| ⅛ | 111,000 | 13,300 |
| 1⁄16 | 27,800 | 3,300 |
| 1⁄32 | 7,000 | 800 |
| 300-lb gauge | | |
| ½ | 2,125,000 | 255,000 |
| ⅜ | 1,195,000 | 143,400 |
| ¼ | 531,000 | 63,700 |
| ⅛ | 132,800 | 15,900 |
| 1⁄16 | 33,200 | 4,000 |
| 1⁄32 | 8,300 | 1,000 |
| 400-lb gauge | | |
| ½ | 2,804,000 | 336,500 |
| ⅜ | 1,577.000 | 189,200 |
| ¼ | 701.000 | 84,100 |
| ⅛ | 175,200 | 21,000 |
| 1⁄16 | 43,800 | 5,300 |
| 1⁄32 | 11,000 | 1,300 |
| 600-lb gauge | | |
| ½ | 4,157,000 | 498,800 |
| ⅜ | 2,338,000 | 280,600 |
| ¼ | 1,039,000 | 124,700 |
| ⅛ | 259,000 | 31,200 |
| 1⁄16 | 65,000 | 7,800 |
| 1⁄32 | 16,300 | 2,000 |

* Values in the table are based on the use of Grashof's formula.

elements to support higher pressures, and the end collars or other structures serving to increase pressure capacity of cylindrical ends and transfer spring force and hydrostatic end thrust to connected piping, and (2) the end nipples or flanges and, for other than anchored axial joints, the various hardware items used to cross-connect the ends, such as tie rods, hinges, and gimbals.

The advantages of the bellows type of joint which permit its use in buried steam piping are that it requires no maintenance and can absorb a combination of axial, lateral, and angular movement within a large range of pressures and temperatures. Its disadvantage is the limitation of traverse, generally 7 to 8 in (180 to 200 mm) of axial movement.

Adequate anchoring and guiding are essential for the proper functioning of all expansion joints. Main anchors for end thrusts must be designed for the sum of the pressure thrust, the force required to deflect the joint, and the force due to friction in the piping guides. Intermediate anchors are used between balanced or double-type joints where the pressure thrust is balanced and the anchor need be designed only to restrict the expansion movement of the pipe. Guides are essential

**TABLE C3.7M (Metric)** Loss Due to Steam Leaks*

| Size of orifice (mm) | Steam wasted per month (kg) | Water wasted ($m^3$) |
|---|---|---|
| 1,724 kPa gauge | | |
| 12.70 | 807,000 | 808.6 |
| 9.52 | 454,000 | 454.6 |
| 6.35 | 201,800 | 202.1 |
| 3.18 | 50,300 | 50.3 |
| 1.59 | 12,600 | 12.5 |
| 0.79 | 3,200 | 3.0 |
| 2,068 kP gauge | | |
| 12.70 | 936,900 | 965.3 |
| 9.52 | 542,000 | 542.8 |
| 6.35 | 240,900 | 241.1 |
| 3.18 | 60,200 | 60.2 |
| 1.59 | 15,100 | 15.1 |
| 0.79 | 3,800 | 3.8 |
| 2,758 kPa gauge | | |
| 12.70 | 1,271,900 | 1,273.8 |
| 9.52 | 715,300 | 716.2 |
| 6.35 | 318,000 | 318.4 |
| 3.18 | 79,500 | 79.5 |
| 1.59 | 19,900 | 20.1 |
| 0.79 | 5,000 | 4.9 |
| 4,137 kP gauge | | |
| 12.70 | 1,885,600 | 1,888.2 |
| 9.52 | 1,060,500 | 1,062.2 |
| 6.35 | 471,300 | 472.0 |
| 3.18 | 117,500 | 118.1 |
| 1.59 | 29,500 | 29.5 |
| 0.79 | 7,400 | 7.6 |

* Values in the table are based on the use of Grashof's formula.

to assure proper pipe alignment into the expansion joint so that undesirable torques are not imposed on the expansion element.

Cold springing of expansion joints used only for lateral deflection can provide several advantages. The most significant of these is the optimization of the position of the expansion joint. In addition, the joint is more stable at high pressures, since the maximum angular displacement of the corrugations is reduced. Joints with internal sleeves and external covers must have adequate clearance to permit the lateral deflection of the expansion element. If the deflection can be reduced by 50 percent, these clearances can also be reduced by 50 percent. Internal sleeves can then be of maximum diameter, and external sleeves are held to a minimum diameter.

All piping systems expand with an increase in temperature in the system. Long runs of high-temperature piping generally use bends, fittings, and offsets to assist in keeping the stresses within the code's allowable values, and reactions (forces and moments) on the turbine or boiler connections within the manufacturer's allowable values. Rigid hangers, constant and variable support hangers, guides, and anchors are also useful to control the thermal growth but must be carefully engineered.

For piping such as main steam and hot reheat systems operating in the creep range [900°F (482°C) and above], cold springing may prove useful in controlling the reactions on the equipment within the manufacturer's specified limits, although this technique is not universally recommended. (Cold springing is the process of cutting short the piping by a certain percentage of the expected thermal growth so that equipment loading in the cold, erected position is that percentage of the calculated hot reactions from the thermal analysis.) Cold springing does not reduce the stresses in the piping as allowed by the Code, but may be useful in meeting the vendors' limits on allowable forces and moments on their equipment, provided that the piping does not creep significantly at operating temperature. As the piping heats up and cycles through a number of thermal cycles in the creep range, the cold reactions will gradually be self-relieved to a certain extent. See ASME B31.1 Code,[2] Paragraphs 119.9 and 119.10 or ASME Section III,[1] Paragraphs NC-3673.3 and NC-3673.5 for a more complete discussion of this subject.

The piping designer needs to work closely with the vendors of the equipment in laying out the high-temperature piping and support systems, in conjunction with the flexibility analyst, so that these systems will meet all requirements over the life of the plant. Especially critical are the loads imposed on rotating equipment such as steam turbines, since clearances are critical and distortion due to piping loads can cause real problems to this equipment.

The true economics of the system design must include all the above factors plus the normal cost factors (piping, insulation, hangers, etc.).

***Fabrication, Assembly, and Erection.*** For underground steam piping, manholes are required for sectionalizing valves and bypass valve piping, trap piping and traps, some types of expansion joints, and convenience of location of other expansion joints and anchorage. Modern manholes are constructed of reinforced concrete, cast-iron, or steel. In field-pour concrete construction, waterproofness should be assured by pouring the walls and floor monolithically. Prefabricated enclosures are pretested for waterproofness before installation. Other provisions that should be considered in manhole design are (1) adequate working space for maintenance, (2) clearance for removal of equipment, (3) ventilation, and (4) drainage. If the manhole floor elevation is below the sewer or if a sewer is not accessible for drainage piping from the manhole, sump pump manholes with automatic pumps or water ejectors must be provided. An example of such an installation is shown in Fig. C3.17. In prefabricated steam-main construction, prefabricated manholes with the necessary valves and piping installed may be delivered to the job site as a unit. This type of manhole is shown in Fig. C3.18.

With prefabricated installation, expansion joints of the bellows design may be delivered to the job site and welded into the steel piping and conduit of the steam main. This provides for directly buried expansion joints in a fully-encapsulated system. Where convenient or necessary, expansion joints are installed in manholes with sectionalizing valves and thrust-type anchors. For ease of maintenance, traps are usually installed in their own manholes; that is, in a manhole separate from that which houses the sectionalizing valve.

Steam piping in power plants is fabricated and installed in accordance with the appropriate code. The following discussion highlights several of the requirements for ASME Class 2 piping,[1] but specific codes should be reviewed as applicable. Components, parts, and appurtenances shall be fabricated and installed in accordance with the rules in the code and shall be manufactured from materials which meet the requirements of the code.

Material for pressure-retaining parts shall carry identification markings which will remain distinguishable until the component is assembled or installed. If the original identification markings are cut off or the material is divided, either the marks shall be transferred to the parts cut, or a coded marking shall be used to ensure identification of each piece of material during subsequent fabrication or installation. In either case, an as-built sketch or a tabulation of materials shall be made identifying each piece of material with the Certified Material Test Report, where applicable, and the coded marking. For studs, bolts, nuts, and heat exchanger tubes, it is permissible to identify the Certified Material Test Reports for material in each component in lieu of identifying each piece of material with the Certified Material Test Report and the coded marking. Material supplied with a Certificate of Compliance and welding and brazing materials shall be identified and controlled so that they can be traced to each component or installation of a piping system, or else a control procedure shall be employed which ensures that the specified materials are used.

Material originally accepted on delivery in which defects exceeding the limits of acceptance are known or discovered during the process of fabrication or installation, is unacceptable. However, the material may be used, provided the condition is corrected in accordance with the requirements for the applicable product form.

Materials may be cut to shape and size by mechanical means such as machining, shearing, chipping, or grinding, or by thermal cutting. When thermal cutting is performed to prepare weld joints or edges to remove attachments or defective material, or for any other purpose, consideration shall be given to preheating the material, using preheat schedules such as suggested in the code.

Any process may be used to hot- or cold-form or bend pressure-retaining materials, including weld metal, provided the impact properties of the materials, when required, are not reduced below the minimum specified values, or they are effectively restored by heat treatment following the forming operation. *Hot-forming* is defined as forming with the material temperature higher than 100°F (38°C) below the lower critical temperature of the material.

When impact testing is required by the design specifications, a procedure qualification test shall be conducted using specimens taken from materials of the same specification, grade or class, heat treatment, and with similar impact properties as required for the material in the component. These specimens shall be subjected to the equivalent forming or bending process and heat treatment as the material in the component. Applicable tests shall be conducted to determine that the required impact properties of the code are met after straining.

The tolerance requirements for formed or bent piping include maintaining minimum wall thickness and ovality. Bending processes shall be selected and qualified to ensure a wall thickness for bent piping sufficient to satisfy the requirements of the design calculations at the resultant section thickness. Unless otherwise justified by the design calculations, the ovality of piping after bending shall not exceed 8 percent as noted in Eq. (C3.6):

$$100 \times \frac{D_{\max} - D_{\min}}{D_o} \leq 0.08 \qquad \text{(C3.6)}$$

where $D_o$ = the nominal pipe outside diameter, in (mm)
$D_{\min}$ = the minimum outside diameter after bending or forming, in (mm)
$D_{\max}$ = the maximum outside diameter after bending or forming, in (mm)

Parts that are to be joined by welding may be fitted, aligned, and retained in position during the welding operation by the use of bars, jacks, clamps, tack welds, or temporary attachments. When the inside surfaces of items are inaccessible for welding or fairing, alignment of sections shall meet the following requirements:

> For circumferential joints the inside diameters shall match each other within 1⁄16 in (1.5 mm). When the items are aligned concentrically, a uniform mismatch of 1⁄32 in (0.8 mm) all around the joint can result. However, other variables not associated with the diameter of the item often result in alignments that are offset rather than concentric. In these cases, the maximum misalignment at any one point around the joint shall not exceed 3⁄32 in (2.4 mm). Should tolerances on diameter, wall thickness, out-of-roundness, and so on, result in inside diameter variation which does not meet these limits, the inside diameters shall be counterbored, sized, or ground to produce a bore within these limits. Offset of outside surfaces shall be faired to at least a 3:1 taper over the width of the finished weld or, if necessary, by adding additional weld metal.
>
> For longitudinal joints the misalignment of inside surfaces shall not exceed 3⁄32 in (2.4 mm), and the offset of outside surfaces shall be faired to at least a 3:1 taper over the width of the finished weld or, if necessary, by adding additional weld metal.

No welding shall be undertaken until after the welding procedures which are to be used have been qualified. Only welders and welding operators who are qualified in accordance with code requirements shall be used. All welding procedure qualification tests shall be in accordance with the requirements of the code.

The method used to prepare the base metal shall leave the weld preparation with reasonably smooth surfaces. The surfaces for welding shall be free of scale, rust, oil, grease, and other deleterious material. The work shall be protected from deleterious contamination and from rain, snow, and wind during welding. Welding shall not be performed on wet surfaces. Rules for making welded joints in piping are described in the following paragraphs.

Backing rings which remain in place may be used for piping in accordance with the requirements of the code. The materials for backing rings shall be compatible with the base metal, but spacer pins shall not be incorporated into the weld.

In double-welded joints, before applying weld metal on the second side to be welded, the root of full penetration double-welded joints shall be prepared by suitable methods such as chipping, grinding, or thermal gouging, except for those processes of welding by which proper fusion and penetrations are otherwise obtained and demonstrated to be satisfactory by welding procedure qualifications.

Where single-welded joints are used, particular care shall be taken in aligning and separating the components to be jointed so that there will be complete penetration and fusion at the bottom of the joint for its full length.

When components of different diameters are welded together, there shall be a gradual transition between the two surfaces. The slope of the transition shall be such that the length-offset ratio shall not be less than 3:1, unless greater slopes are shown to be acceptable by analysis. The length of the transition may include the weld.

Thickness of weld reinforcements for piping is specified in Table C3.8. For double-welded butt joints the limitation on the reinforcement given in Column 1 of the table shall apply separately to both inside and outside surfaces of the joint. For single-welded butt joints, the reinforcement given in Column 2 shall apply to the inside surface and the reinforcement given in Column 1 shall apply to the outside surface. The reinforcements shall be determined from the higher of the abutting surfaces involved.

Fillet welds may vary from convex to concave. A fillet weld in any single continu-

**TABLE C3.8** Thickness of Weld Reinforcements for Piping[1]

| Material nominal thickness in (mm) | Column 1: Both double-welded butt joints and outside single-welded butt joints | Column 2: Inside single-welded butt joints |
|---|---|---|
| Up to ½ (3.2), incl. | 3⁄32 (2.4) | 3⁄32 (2.4) |
| Over 1⁄8–3⁄16 (3.2–4.8), incl. | 1⁄8 (3.2) | 3⁄32 (2.4) |
| Over 3⁄16–½ (4.8–12.7), incl. | 5⁄32 (4.0) | 1⁄8 (3.2) |
| Over ½–1 (12.7–25.4), incl. | 3⁄16 (4.8) | 5⁄32 (4.0) |
| Over 1–2 (25.4–51), incl. | ¼ (6.4) | 5⁄32 (4.0) |
| Over 2 (51) | The greater of ¼ in (6.4 mm) or 1⁄8 times the width of the weld in inches (mm) | 5⁄32 (4.0) |

ous weld may be less than the specified fillet weld dimension by not more than 1⁄16 in (1.5 mm), provided that the total undersize portion of the weld does not exceed 10 percent of the length of the weld. Individual undersize weld portions shall not exceed 2 in (52 mm) in length. In making socket welds, a gap of approximately 1⁄16 in (1.5 mm) at the end of the pipe shall be provided prior to welding. The gap need not be present nor be verified after welding.

Structural attachments shall conform reasonably to the curvature of the surface to which they are attached. Full penetration, fillet, or partial penetration continuous or intermittent welds are acceptable, depending on service conditions.

Attachments may be welded to the piping system after performance of the pressure test provided the welds do not require postweld heat treatment (PWHT); welds shall be restricted to fillet welds not exceeding 3⁄8 in (9.5 mm) throat thickness and to full penetration welds attaching materials not exceeding ½ in (12.7 mm) in thickness; welds shall not exceed a total length of 24 in (60 cm) for fillet welds or 12 in (30 cm) for full penetration welds; and welds shall be examined as required by the code.

The need for and temperature of preheat are dependent on factors such as the chemical analysis, degree of restraint of the parts being joined, elevated temperature, physical properties, and material thicknesses. Postweld heat treatment must provide the required heating and cooling rates, metal temperature, metal temperature uniformity, and temperature control.

The threads of all bolts or studs shall be engaged in accordance with the design. Any lubricant or compound used in threaded joints shall be suitable for the service conditions. All threading lubricants or compounds shall be removed from surfaces which are to be seal-welded. In bolting gasketed flanged joints, the contact faces of the flanges shall bear uniformly on the gasket in accordance with the design principles applicable to the type of gasket used. All flanged joints shall be made up with uniform bolt stress.

***Examination, Inspection, and Testing.*** The inside of all pipes, valves, fittings, traps, and other apparatus shall be smooth, clean, and free from all blisters, loose mill scale, sand, and dirt when erected. All main steam lines and significant branch lines shall be steam-blown before placing in service.

Steam piping in power plants must be examined, inspected, and tested in accordance with the appropriate code. The following discussion summarizes several general requirements for ASME Class 2 piping,[1] but specific codes should be reviewed as applicable.

The examinations required by the code shall be performed by personnel who have been qualified as required by code, and results of the examinations shall be evaluated in accordance with the specified acceptance standards.

Following any nondestructive examination in which examination materials are applied to the piece, the piece shall be thoroughly cleaned in accordance with applicable materials or procedure specifications.

Acceptance examinations of welds and weld metal cladding shall be performed at the following stipulated times during fabrication and installation.

1. Radiographic examination of welds will usually be performed after an intermediate or final PWHT, when required.
2. Magnetic particle or liquid penetrant examinations of welds shall be performed after any required PWHT, except that welds in P-No. 1 materials may be examined either before or after PWHT.
3. All dissimilar metal weld joints such as in austenitic or high nickel to ferritic material or using austenitic or high nickel alloy filler metal to join ferritic materials which penetrate the wall shall be examined after final PWHT.
4. The magnetic particle or liquid penetrant examination of weld surfaces that are to be covered with weld metal cladding shall be performed before the weld metal cladding is deposited. The magnetic particle or liquid penetrant examination of weld surfaces that are not accessible after a PWHT shall be performed prior to the operation which caused this inaccessibility. These examinations may be performed before PWHT.
5. Ultrasonic examination of electroslag welds in ferritic materials shall be performed after a grain refining heat treatment, when performed, or after final PWHT.

The codes provide specific acceptance standards for radiographic, ultrasonic, magnetic particle, liquid penetrant, and metallographic examination of all components. Unique examination requirements are usually specified for bellows expansion joints.

***Pressure Testing.*** Before installation, all valves, fittings, and so forth shall be capable of withstanding an appropriate hydrostatic shell test, and piping shall be capable of meeting the hydrostatic test requirements contained in the respective material specifications under which it was purchased. All pressure-retaining components, appurtenances, and completed systems shall be pressure tested. Bolts, studs, nuts, washers, and gaskets are exempted from the pressure test. Pressure testing required by the applicable code often must be performed in the presence of the inspector.

After installation, the complete system shall be hydrostatically tested as noted in Table C3.9. When conducting pressure tests at low metal temperatures, the possibility of brittle fracture shall be considered. At no time during the hydrostatic test shall test pressure exceed the maximum allowable test pressure of any nonisolated components, such as vessels, pumps, or valves. Nor shall any of the maximum allowable stress limits be exceeded for any components being tested.

Following the application of the hydrostatic test pressure for the required time, all joints, connections, and regions of high stress (such as regions around openings and thickness transition sections) shall be examined for leakage, except in the case of pumps and valves, which shall be examined while at test pressure. The piping

**TABLE C3.9** Selected Hydrostatic Test Pressure and Allowable Limits[1,2]

| Code class and category of component | Hydro test pressure as a multiple of the design pressure (DP) | Maximum allowable stress at test temperature, circumferential (hoop) stress, if not noted otherwise |
|---|---|---|
| ASME Class 1 | | |
| Vessels | 1.25 times DP | Note 1 |
| Piping | 1.25 times DP | Note 1 |
| Equipment | 1.5 times DP | Note 1 |
| Valves | Per NB-3531[1] | Per NB-3531[1] |
| Relief valves | 1.5 × set pressure | Per NB-3531[1] |
| ASME Class 2 and 3 | | |
| Vessels | 1.25 times DP | Note 2 |
| Piping | 1.25 × DP, Note 3 | Note 2 |
| Equipment | 1.5 times DP | Note 2 |
| Valves | Per NC/ND-3500[1] | Per NC/ND-3500[1] |
| ASME B31.1 | | |
| System | 1.5 times DP | 90% of yield strength, Note 4 |
| Piping | 1.5 times DP | 90% of yield strength, Note 4 |
| Equipment | Note 5 | Note 5 |
| Valves | Note 5 | Note 5 |

***Notes:*** 1. Primary membrane stress intensity less than 90% of tabulated yield strength at test temperature and other limits in NB-3226.[1]
2. Primary membrane stress intensity less than 90% of tabulated yield strength at test temperature and other limits in NC-3218.[1]
3. As an alternative, piping between the discharge side of a centrifugal pump and the first shutoff valve may be hydrostatically tested at the shutoff head of the pump (NC/ND-6221[2]).
4. In addition, the sum of longitudinal stresses due to test pressure and live and dead loads at the time of test, excluding occasional loads, shall not exceed 90% of the yield strength at test temperature (Para. 102.3.3[2]).
5. Hydrostatic test pressure shall not exceed the maximum allowable test pressure of any nonisolated components, such as vessels, pumps, or valves.

system, exclusive of possible localized instances at pump or valve packing, shall show no visual evidence of weeping or leaking. Leakage of temporary gaskets and seals, installed for the purpose of conducting the hydrostatic test, and which will be replaced later, may be permitted unless the leakage exceeds the capacity to maintain system test pressure for the required amount of time. Other leaks, such as from permanent seals, seats, and gasketed joints in components, may be permitted when specifically allowed by the design specifications.

The component or system in which the test is to be conducted shall be vented during the filling operation to minimize air pocketing. Water or an alternative liquid, as permitted by the design specification, shall be used for the hydrostatic test. It is recommended that the test be made at a temperature that will minimize the possibility of brittle fracture. The test pressure shall not be applied until the component, appurtenance, or system and the pressurizing fluid are at approximately the same temperature.

The pressure test may be performed progressively on erected portions of the system. Components and appurtenances shall be pressure-tested prior to installation in a system except that the system pressure test may be substituted for a component or appurtenance pressure test provided the following is true: The component can be repaired by welding, if required, as a result of the system pressure test; the component repair weld can be postweld heat-treated, if required, and nondestructively examined, as applicable; and the component is resubjected to the required system pressure test following the completion of repair and examination if the repair is required to be radiographed. Valves also require pressure testing prior to installation in a system.

All joints, including welded joints, shall be left uninsulated and exposed for examination during the test. Components designed to contain vapor or gas may be provided with additional temporary supports, if necessary, to support the weight of the test liquid. Expansion joints shall be provided with temporary restraints, if required, for the additional pressure load under test. Equipment that is not to be subjected to the pressure test shall be either disconnected from the component or system or isolated during the test by a blind flange or similar means. Valves may be used if the valves with their closures are suitable for the proposed test pressure. Flanged joints at which blanks are inserted to isolate other equipment during the test need not be retested. If a pressure test is to be maintained for a period of time and the test medium in the system is subject to thermal expansion, precautions shall be taken to avoid excessive pressure. The test equipment shall be examined before pressure is applied to ensure that it is tight and that all low-pressure filling lines and other items that should not be subjected to the test have been disconnected or isolated.

The hydrostatic test requirements for bellows expansion joints require that the completed expansion joint shall be subjected to a hydrostatic test in accordance with the applicable code. This test may be performed with the bellows fixed in the straight position, at its neutral length, or in some cases, this test shall be performed with the bellows fixed at the maximum design rotation angle or offset movement.

Pressure test gauges used in pressure testing shall be indicating pressure gauges and shall be connected directly to the component. If the indicating gauge is not readily visible to the operator controlling the pressure applied, an additional indicating gauge shall be provided where it will be visible to the operator throughout the duration of the test. For systems with a large volumetric content, it is recommended that a recording gauge be used in addition to the indicating gauge. All test gauges shall be calibrated against a standard dead-weight tester or a calibrated master gauge. The test gauges shall be calibrated before each test or series of tests.

When hydrostatic testing is impractical, it shall be required that the piping be tested with steam at a pressure at least equal to the pressure at which the piping is to be operated. These tests shall be made on sections, or on the whole of the piping system, but the connections between the sections must be similarly tested.

## Special Design Criteria

***Overpressure Protection.*** Overpressure protection of the components in a steam system shall be provided as an integrated system using pressure-relief devices and associated pressure-sensing elements. Detail requirements for overpressure protection for ASME Class 1, 2, and 3 systems are provided in Articles NB-7000, NC-7000, and ND-7000,[1] respectively.

Each main steam lead generally has safety valves at the superheater outlet and often a motor-operated block valve which is used for isolation. A main steam stop valve at the boiler outlet facilitates boiler hydro testing prior to completion of turbine erection; whether to provide it is a matter of economics. In the case of two leads, the cross-connection between the two main steam leads shall permit full-closed testing of one turbine stop valve at a time while the turbine is operating.

The cross-connection line may have connections to the steam-generator feed pump turbines, auxiliary steam loads, turbine-generator gland seal system, and the steam-generator restart system, depending on system design requirements.

***Protecting District Heating and Steam Distribution Systems.*** Construction of underground steam mains must adhere to the requirements as provided for in Section 122.14 of ASME B31.1[2] when the piping is within the jurisdiction of ASME B31.1. Where pressure-reducing valves are used, one or more relief devices or safety valves are provided on the low-pressure side of the system. The relief or safety devices are located as close as practicable to the reducing valve. The combined relieving capacity provided must be such that the design pressure of the low-pressure system will not be exceeded if the reducing valve fails to open.

In district heating and steam distribution systems where the steam pressure does not exceed 400 psi (2760 kPa) and where the use of relief valves as previously described is not feasible, alternative designs may be substituted for the relief devices. For example, this could occur if there is no acceptable discharge location for the vent piping. Two or more steam-pressure-reducing valves capable of independent operation may be installed in series, each set at or below the safe working pressure of the equipment and piping system served. In this case, no relief device is required. A trip-stop steam valve set to close at or below the design pressure of the low-pressure system may be used in place of a second reducing valve or a relief valve.

***Design of Restrained Underground Piping.*** Appendix VII in B31.1[2] provides non-mandatory procedures which may be conservatively applied to the design and analysis of buried piping systems. Buried piping is supported, confined, and restrained continuously by the passive effects of the backfill and the trench bedding. The effects of continuous restraint cannot be easily evaluated by the usual methods applied to exposed piping, since these methods cannot easily accommodate the effects of bearing and friction at the pipe/soil interface. All components in the buried system must be given consideration, including the building penetrations, branches, bends, elbows, flanges, valves, grade penetrations, and tank attachments. Welds should be made in accordance with the B31.1 code, and appropriate corrosion protection procedures for buried piping are to be followed.

***Preventing Turbine Overspeed.*** Turbine water induction problems from main steam lines would normally only occur during start-up or shortly after shutdown, so prevention criteria are limited to low-point drain design per ASME Standard No. TDP-1.[26] Steam storage in the turbine, reheater, and reheat piping presents a problem in turbine speed control because surplus energy can continue to be generated in the IP and the LP sections for some time after the main governor has reacted to a load decrease. This condition could result in dangerous overspeeding. To prevent this, governor-operated intercept valves are installed ahead of the point of reintroduction of the reheated steam to the turbine. The IVs are adjusted to begin closing when the normal speed is exceeded but before turbine speed has increased to the point (approximately 10 percent over normal RPM) where an emergency overspeed trip would shut down the whole unit.

## Understanding Reheat Steam Systems

***Advantages of Reheat Cycles.*** The example of power plant cycle efficiency improvement by increased initial steam conditions in "Design Temperature" suggests that maintaining the initial temperature constant over a range of pressures (as in the Carnot Cycle) might also prove fruitful. This would require reheating the main steam to its initial temperature again after each infinitesimal amount of expansion in the turbine. Such a cycle might be said to have an infinite number of reheats.

There is also another important advantage derived from using steam reheat in a power cycle. Reheating is a feature associated with high steam pressures where there is insufficiently high initial steam temperature to yield an expansion that will end with an acceptable moisture content at the condenser pressure. This limit of moisture at the last stage of a turbine is roughly 12 to 14 percent. Moisture in the steam flow causes erosion of turbine buckets, shortening their useful life, and also results in an efficiency loss in the turbine. The purpose of reheat is to take full advantage of higher initial steam parameters discussed in ''Design Temperature'' as they become commercially available.

***Reheat Performance Efficiency.*** The number of reheats to be used must be evaluated. The high-pressure, high-temperature steam generator, together with the extra cost of a turbine, piping, and controls, makes plants with reheat more expensive than those without this feature. Economically, single reheating has proved desirable on most machines above 100 MW. In actual practice the gain from single reheating is about 4 to 7 percent in thermal efficiency over an equivalent nonreheat cycle. The gain in performance which can be realized from a second reheating is smaller than that from the first. For large units operating at 3500 psig (24.2 MPa), the gain in performance due to a second reheat is in the neighborhood of 2 percent of the net heat rate (see Fig. C3.8). Performance gains for additional reheats diminish rapidly, and the justification for greater expenditures for equipment depends strongly upon the price of fuel.

For every set of initial steam conditions there exists an optimum reheat pressure (or pressures in a double-reheat cycle) which yields the maximum efficiency (minimum heat rate) of the power plant. Studies made of the performance and economic application of single-reheat cycles indicate that for a seven (or less) feedwater heater cycle the cold reheat pressure is adjusted by the turbine manufacturer to provide either the optimum or desired final feedwater temperature. If eight (or more) feedwater heaters are used, it proves economical to design the cycle with the first point heater located above the reheat point (HARP extraction). Figure C3.19[27] indicates typical optimum reheat pressures corresponding to 2400 psig/1000°F/1000°F (16.6 MPa/538°C/538°C) steam conditions for units taking extraction for the top heater at the reheat point (curve marked ''Top heater at reheat point'') or above the reheat point where the optimum reheat pressure is found at the low point of a curve corresponding to required final feedwater temperature.

It is seen from this figure that minor deviations from the optimum reheat pressure have a small effect on performance; however, the performance penalties should always be estimated. Size of the turbine, throttle temperature, and reheat temperature have a relatively minor effect on the indicated optimum conditions, and Fig. C3.19 may therefore be considered as reasonably accurate for any single-reheat unit at indicated throttle pressure. However, if more accurate information is needed, it should be obtained from the turbine manufacturer. It should be noted that the relative performance indicated applies only at rated output.

The selection of optimum reheat pressures for a double-reheat turbine is an involved problem. It is affected not only by thermodynamic optimization but also by design limitations (e.g., second-reheat intercept-valve pressure must be fixed at about 300 to 400 psia (2070 to 2760 kPa) at rated conditions due to the very large volumetric steam flows which would result at lower pressures). Figures C3.20 and

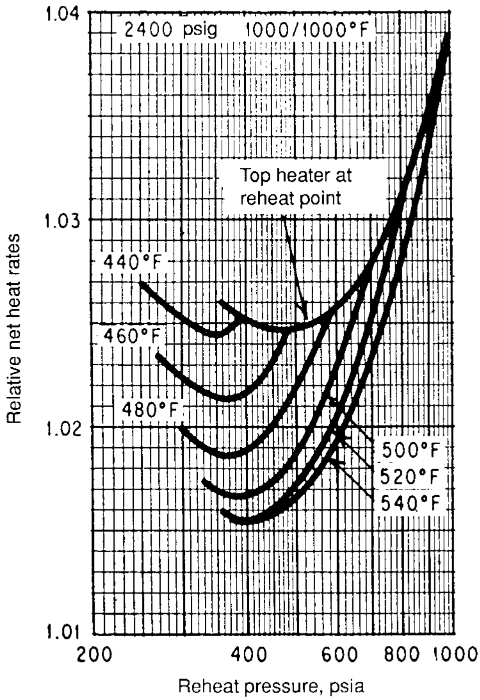


**FIGURE C3.19** Relative performance for typical single-reheat units with various combinations of first-reheat intercept-valve pressure and final feedwater temperature.[27] *(Courtesy of General Electric Co.)*

C3.21[27] show the heat rate penalty which results from the use of reheat pressures other than optimum when the throttle pressure is 3500 to 4500 psig (24.2 to 31.1 MPa), respectively, and the second cold reheat pressure is fixed at 300 psia (2070 kPa).

Tables C3.10[28] and C3.10M (Metric) compare the basic cycle parameters for the 738-MW, 3500-psig (24.2-MPa) unit and the 900-MW, 4500-psig (31.1-MPa) design. The relatively higher first reheat pressure of the 4500-psig (31.1-MPa) cycle makes possible an additional attractive thermodynamic gain when an eighth feedwater heater, extracting steam from the first reheat section, is added to the cycle. The use of a heater above the reheat point in the 3500-psig (24.2-MPa) cycle will result in an improvement in the heat rate. In this case the required final feedwater temperature would set the limit on the HARP extraction pressure.

## Understanding the Extraction Steam System

Regenerative feedwater heating improves the Rankine cycle efficiency. It is achieved by withdrawing a portion of steam flowing through the turbine at a number of points along the flow path (extraction, bleeding). The steam so extracted is condensed in

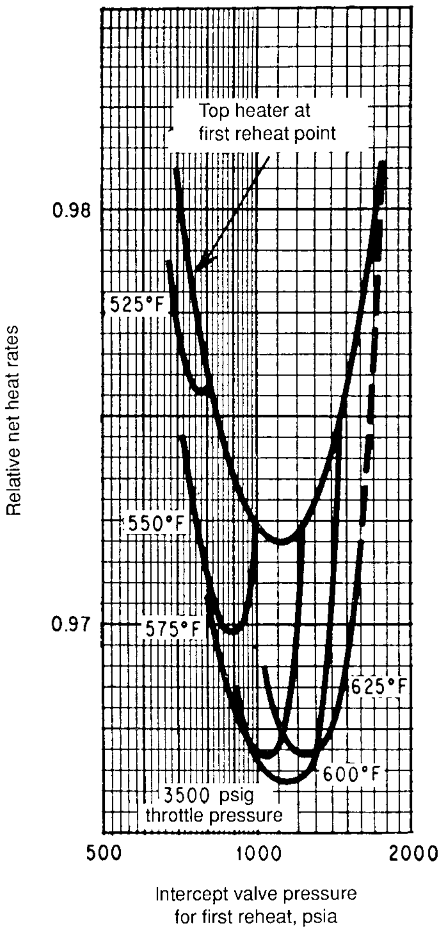


**FIGURE C3.20** Relative performance for typical double-reheat units with various combinations of first-reheat intercept-valve pressure and final feedwater temperature.[27] *(Courtesy of General Electric Co.)*

a series of heat exchangers (feedwater heaters) through which flows the feedwater from the condenser on its return to the steam generator. Bleeding is also advantageous for the turbine in that it reduces the mass flow rate of steam in the lower-pressure stages, which is a desirable feature from the turbine designer's point of view. In most cases it reduces the exhaust losses which occur between the last turbine stage and the condenser.

One of the most frequent cycle changes which is considered in designing a power plant is whether to increase the number of feedwater heating extraction points. This problem does not have a simple answer applicable in every case. Theoretical prediction of a change in station heat rate, as published by many authors, is based

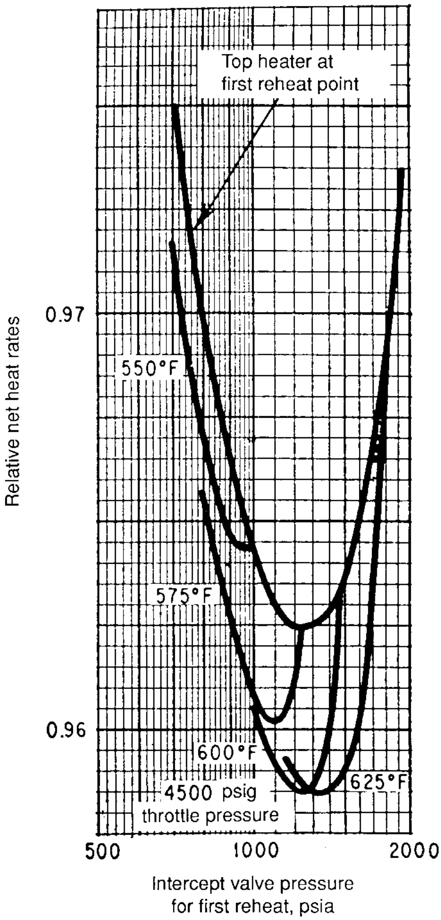


**FIGURE C3.21** Relative performance for typical double-reheat units with various combinations of first-reheat intercept-valve pressure and final feedwater temperature.[27] *(Courtesy of General Electric Co.)*

**TABLE C3.10** Basic Cycle Parameters[28]

| | 738 MW, 3500 psig | 900 MW, 4500 psig |
|---|---|---|
| Maximum capability, MW | 826 | 1000 |
| Maximum throttle pressures, psia | 3500 | 4500 |
| First reheat intercept valve pressure, psia | 980 | 1360 |
| Second reheat intercept valve pressure, psia | 340 | 390 |
| Final feedwater temperature, °F | 548 | 590 |
| Number of feedwater heaters | 7 | 8 |

**TABLE C3.10M (Metric)** Basic Cycle Parameters[28]

| | 738 MW, 24.13 MPa gauge (3500 psig) | 900 MW, 31.03 MPa gauge (4500 psig) |
|---|---|---|
| Maximum capability (MW) | 826 | 1000 |
| Maximum throttle pressure (MPa) | 24.13 | 31.03 |
| First reheat intercept valve pressure (MPa) | 6.76 | 9.38 |
| Second reheat intercept valve pressure (MPa) | 2.34 | 2.69 |
| Final feedwater temperature (°C) | 287 | 310 |
| Number of feedwater heaters | 7 | 8 |

on the assumption that the enthalpy (or temperature) rise is divided equally among the heaters. This is difficult to achieve in practice since the actual feedwater enthalpy rises in the heaters are determined by the extraction location available in the turbine. As a result, the heat-rate gains due to additional heaters typically may be only about one half the theoretical value. Normally for optimum application, the heater above the reheat point has a feedwater temperature rise of 50 to 100°F (10 to 38°C). The number of feedwater heaters to be used is also an economic consideration which must be determined on the basis of power plant cycle optimization. Recently, seven or eight heaters have been used in both fossil-fueled and nuclear power plants.

It must be emphasized that the stage pressures at extraction points change with turbine load and are roughly proportional to the mass flow rate of steam to the following stage.

In a multistring cycle when a heater is removed from service, the extraction cross-tie arrangement will be one of the deciding factors in the turbine load limitation. Extraction cross ties are the means by which extraction pipes at the same pressure level can be connected.

## *EXPERIENCE FEEDBACK*

Use appropriate materials and welding processes and procedures for all piping. Be aware of erosion and corrosion problems in high-moisture-content steam lines (>5 percent moisture steam extraction lines) and in drain lines (feedwater heater, moisture separator, and reheat drain lines).

### Preventing Turbine Water Induction

Turbine water induction prevention criteria for hot reheat lines in fossil-fired power plants are similar to those for main steam, that is, low-point drain design per ASME TDP-1.[26] Turbine water induction prevention criteria for cold reheat lines are more of a concern due to reheater atemperator (regulating the hot reheat steam temperature) spray or feedwater heaters which extract steam from the cold reheat line. The special prevention criteria for cold reheat lines are specified in the ASME Standard No. TDP-1. For nuclear power plants with light water cooled reactors (PWR, BWR), cold reheat lines carry steam containing moisture. Therefore, the special prevention criteria specified in ASME Standard No. TDP-2[7] must be applied.

## *REPAIRS, REPLACEMENTS, AND MODIFICATIONS*

Use of flanges in high-pressure and high-temperature steam may cause leakage problems. Defects in weld metal detected by examinations or by tests shall be eliminated and repaired when necessary or the indication reduced to an acceptable limit. Weld metal surface defects may be removed by grinding or machining and need not be repaired by welding, provided the remaining thickness of the section is not reduced below that required. The depression, after defect elimination, is blended uniformly into the surrounding surface; and the area is examined by a magnetic particle or liquid penetrant method in accordance with the code after blending. Defects detected by visual or volumetric method and located on an interior surface need only be reexamined by the method which initially detected the defect when the interior surface is inaccessible for surface examination. Excavations in weld metal, when repaired by welding, shall meet the requirements of the code. The repaired area shall be heat treated, if required.

It is important, for the convenience of the operating and maintenance personnel and to minimize the possibility of making mistakes, that piping be arranged so as to make the purpose of each line and valve as obvious as possible. Good arrangement is especially important to personnel making repairs, replacements, and modifications to piping systems and components. Valves in general, and bypass valves in particular, should be installed so that their purpose is evident at a glance. It is desirable that bypass valves be grouped together where all can be seen at once rather than placed in scattered locations close to separate pieces of equipment. The possibility of an error in operation or maintenance procedures can be further reduced by stenciling the purpose of each line on the pipe close to its valve. A simple illustration of this point is shown in Fig. C3.22, a photograph of the piping for a small deaerator.

A good piping arrangement should utilize the foregoing criteria. It must also

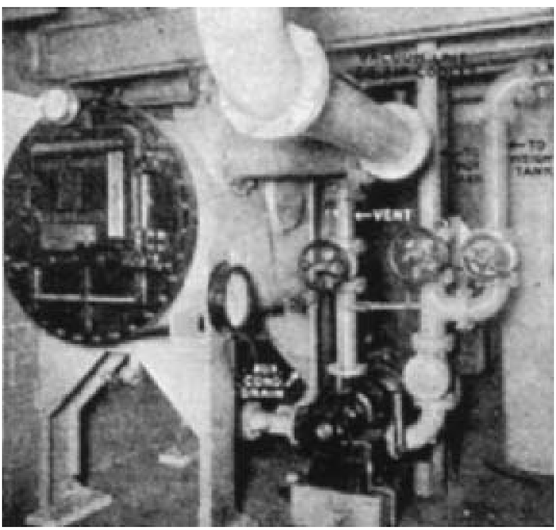


**FIGURE C3.22** How to make the purpose of valves obvious.

be coordinated with the plant structure; heating, ventilating, and air-conditioning (HVAC) systems; electrical layout; equipment selection; and other local features that impact access, visibility, and related factors. The concepts in Chap. B1 and B3, the flow characteristics in Chap. B8, and the computer methods in App. E9 of this handbook provide guidance for the design phase of steam piping systems. Operational and life-extension phases must also be considered in steam systems.

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