

Heat Recovery Systems

API RECOMMENDED PRACTICE 669
FIRST EDITION, JUNE 2022



American
Petroleum
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Introduction

Users of this recommended practice (RP) need to be aware that further or differing requirements can be needed for individual applications. This RP is not intended to inhibit a vendor from offering, or the purchaser from accepting, alternative equipment or engineering solutions for the individual application. This can be particularly applicable where there is innovative or developing technology. Where an alternative is offered, the vendor should identify any variations from this RP and provide details.

This RP requires the purchaser to specify certain details and features.

In this RP, System International (SI) units are used and where practical, U.S. customary (USC) units are included in parentheses for information.

Heat Recovery Systems

1 Scope

1.1 This recommended practice (RP) provides guidelines for heat transfer equipment used in waste heat recovery systems in the petroleum, petrochemical, and natural gas industries. Details of related equipment designs are included only where these are necessary to ensure proper design and operation and safe interaction with the heat recovery system. It indicates areas that need attention and offers information and descriptions of various types of heat transfer equipment available to aid in the selection of the appropriate heat recovery system.

1.2 This RP does not include detailed requirements for:

- process to process heat recovery systems,
- gas turbine exhaust heat recovery,
- CO boilers, or
- convection sections used within fired heaters.

1.3 The waste heat recovery systems included in this RP are typical of those currently in use and should not limit the use of alternate systems.

1.4 Instrumentation and controls for waste heat recovery systems are sometimes mentioned but is not meant to completely define all requirements. Other standards and documents, such as API Recommended Practice 556, *Instrumentation, Control, and Protective Systems for Fired Heaters*, should be used to fully define the needed instruments and controls.

2 Normative References

There are no normative references in this document.

3 Terms, Definitions, Acronyms, and Abbreviations

3.1 Definitions

For the purposes of this document, the following terms and definitions apply.

3.1.1

approach temperature

The difference between the saturation temperature of the steam at the selected pressure and the temperature of the water leaving the economizer.

3.1.2

desuperheater attemperator

A device located internal or external to the HRSG that controls the exit temperature of the steam from the superheater. The device typically injects water with very low solids content into the steam to control the steam temperature by reducing it. Also called an attemperator.

3.1.3

downcomer

A heated or unheated pipe carrying water from the steam drum to an evaporator/generator section of an HRSG.

**3.1.4
evaporator
generator**

The portion of the HRSG in which water is boiling to form steam. Typically, a mixture of water and steam exists at the exit of this portion. In some configurations, the steam and water are separated prior to leaving this portion of the HRSG. Also referred to as a steam generator section.

**3.1.5
economizer**

The portion of the HRSG where incoming feedwater temperature is raised to less than saturation temperature by recovery of the heat from the heating medium.

**3.1.6
ferrule**

A high temperature resistant metallic or ceramic shaped tube insert provided at the tube and tubesheet area to insulate components from high temperatures and to limit heat transfer.

**3.1.7
firetube HRSG**

A shell-and-tube heat exchanger in which steam is generated on the shell side by heat transferred from hot fluid flowing through the tubes.

**3.1.8
heat recovery steam generator (HRSG)**

A system in which steam is generated and may be superheated or water heated by the transfer from heat medium usually gaseous products of combustion or other hot process fluids.

**3.1.9
heat recovery system**

A system that transfers energy from one medium to a second medium for reuse.

**3.1.10
pinch temperature**

The difference between the heating medium temperature leaving the steam generator section and the steam's saturation temperature at the selected pressure.

**3.1.11
process fluid**

The heating medium used to supply the heat to the utility fluid.

**3.1.12
riser**

A heated or unheated pipe carrying water and steam from an evaporator/generator section of an HRSG to the steam drum.

**3.1.13
shell-and-tube watertube HRSG**

A shell-and-tube heat exchanger in which steam is generated in the tubes by heat transferred from a hot fluid on the shell side.

**3.1.14
steam drum**

A pressure vessel whose primary purpose is to separate water and steam.

**3.1.15
superheater**

The portion of the HRSG in which saturated steam is heated to higher temperatures.

3.1.16**waste heat recovery unit (WHRU)**

A non-fired heat exchanger in which waste heat is transferred to a cold fluid which is not water being generated to steam.

3.1.17**watertube**

A multiple tube circuit heat exchanger in which the utility fluid flows inside the tubes and the heating medium flows over the tubes. Most commonly the tube contains water in a water/steam system.

3.1.18**watertube low-pressure casing HRSG**

A multiple tube circuit heat exchanger within a gas-containing casing in which steam is generated inside the tubes by heat transferred from a hot gas flowing over the tubes.

3.2 Acronyms and Abbreviations

For the purposes of this document, the following acronyms and abbreviations apply.

| | |
|------|------------------------------------|
| BFW | boiler feedwater |
| CR | circulation ratio |
| FAC | flow accelerated corrosion |
| FCCU | fluid catalytic cracking unit |
| HRSG | heat recovery steam generator |
| HTF | heat transfer fluid |
| MAWP | maximum allowable working pressure |
| NPSH | net positive suction head |
| RCCU | residual catalytic cracking unit |
| SRU | sulfur recovery unit |
| TO | thermal oxidizer |
| TRSG | thermal reactor steam generator |
| WHB | waste heat boiler |
| WHRU | waste heat recovery unit |

4 General Information**4.1 Utility Systems**

4.1.1 See Annex A for information on "Steam Systems".

4.1.2 See Annex B for information on "Hot Oil and Other Heat Transfer Fluid Systems".

4.1.3 See Annex C for information on "Hot Water/Glycol Systems".

4.2 Mechanical Descriptions

4.2.1 See Annex D for information on "Watertube WHRU Configurations".

4.2.2 See Annex E for information on "Firetube WHRU Configurations".

4.3 Specific Heat Recovery Applications

- 4.3.1** See Annex F for information on “FCCU/RCCU Process Unit Heat Recovery Systems”.
- 4.3.2** See Annex G for information on “Process Heater Convection Sections”.
- 4.3.3** See Annex H for information on “Gas Turbine Exhaust Heat Recovery Systems”.
- 4.3.4** See Annex I for information on “Thermal Oxidizers Heat Recovery Systems”.
- 4.3.5** See Annex J for information on “Process/Flue Gas Heat Recovery Systems”.
- 4.3.6** See Annex K for information on “Process Liquid Heat Recovery Systems”.
- 4.3.7** See Annex L for information on “Sulfur Recovery Unit Heat Recovery Systems”.
- 4.3.8** See Annex M for information on “Heat Flux and Circulation Ratio”.

Annex A (informative)

Steam Systems

A.1 General

A.1.1 Steam systems are used for recovering waste energy from a process or flue gas stream and returning that energy by vaporizing water to steam for low, medium, or high-pressure steam systems within the plant. There are also opportunities for preheating BFW as well as superheating steam if the configuration benefits.

A.1.2 Steam systems may be closed, in which the condensate leaving the steam consumers is recovered and returned to boiler/steam generator, or they may be once through systems, in which the steam produced is often used in direct contact with the process medium and is non-recoverable.

A.1.3 There are many applications for steam generating systems including sulfur recovery units (sulfur condensers and thermal reactor steam generators), gas turbine exhaust systems, incinerator waste heat exchangers, process steam generators.

A.2 Boiler Feed Water/Condensate

A.2.1 Water conservation and quality control are important for closed systems and the boiler feedwater (BFW)/condensate is ideally clean water with minimal contaminant. Typically, de-aeration and chemical dosing are applied to the makeup water. Blowdown from the boiler drum(s) is used to maintain the quality of the boiler feedwater and prevent the buildup of impurities. The relatively high cost associated with the supply and treatment of makeup water means that it is important to minimize losses from the system and recirculate condensate wherever possible.

A.2.2 The steam exiting the HRSG should be of specified purity (contaminants) and quality (water content). Failure to control the steam purity is likely to accelerate damage to downstream equipment. For steam that is used in direct contact applications and is not recovered, the water quality and dryness of the steam may be less stringent but must still be adequate to prevent scaling and damage to the boiler/steam generator and downstream equipment. Contaminants can be controlled as with any steam generating or boiler system. Refer to API Recommended Practice 538 for additional detail on feedwater preparation, chemical treatment, and steam purity.

A.2.3 In most closed system applications, it is preferable for the quality of the steam (the dryness) to be high. Water droplets entrained within the steam can have a detrimental impact on the downstream equipment, possibly causing fouling or scaling. Water droplets carried along at high speed in steam piping systems can cause erosion of pipes, fittings and valves and can cause considerable damage if allowed to impact steam turbine blades. If there is a large amount of water carried over into the steam mains/headers and insufficient means of removing it, pooling could occur within these lines which may result in water hammer or slugging.

A.2.4 To achieve a high steam quality at its source, demister pads or other types of steam/water separation devices may be installed at the outlet of the steam generator, or these may be used internal or external to a steam drum. As the steam flows through the distribution mains/headers, some cooling and condensing will occur (unless superheated) and the quality will be reduced. Steam separators and steam traps may be installed within the mains/headers to ensure that the steam quality reaching the consumers remains high. Refer to API Recommended Practice 538 for additional details on steam purity.

A.2.5 Superheating can be used to ensure the dryness of the exported steam. A high steam quality into the superheater section is desirable to avoid fouling of the superheater heat transfer surface. Reducing the steam pressure can be another means of increasing the dryness of the steam depending upon where the steam system

operating conditions are on the Mollier diagram; e.g. reducing the pressure of high-pressure saturated steam to around 4000 kPa(a) (580 psia) or higher will reduce the steam quality. In once through applications low quality steam may be acceptable and the measures described above may be unnecessary but the potential issues of erosion and condensate buildup in the distribution lines must still be addressed.

A.3 Steam System Components

A.3.1 Typical Steam System Components

Figure A.1 shows a sketch of typical steam system components. Not all components are used in every design nor are all the needed components shown.

A.3.2 Steam Drum

A.3.2.1 The steam drum provides several functions within the steam system. Primarily it provides phase separation for the steam/boiler water stream returning from the evaporator but also serves to provide adequate boiler water holdup to ensure constant flow to the evaporator. A drum will typically be provided with level control and safeguarding to ensure that the makeup boiler water flow is balanced with the steam export rate, blowdown rate, and the liquid level within the drum remains between safe limits for operation. Maintaining the vapor space above the liquid level is also important for achieving the required steam quality leaving the steam drum.

A.3.2.2 The boiler water quality within the drum should be routinely monitored below the liquid level through sampling. Blowdown lines (continuous and/or intermittent) are employed to control the buildup of dissolved solids. The rate of continuous blowdown or frequency of intermittent blowdown will depend on the quality of the makeup boiler water.

A.3.2.3 In a natural circulation system, the drum must always be mounted above the evaporator. In a forced circulation system, the drum is typically above the evaporator but can be below the evaporator when the system is properly designed. In all cases it must be elevated above the boiler water circulation pumps to prevent cavitation at the pump inlet by ensuring sufficient net positive suction head (NPSH). Designing to ensure that all heat transfer surfaces are filled/covered with water even if the pumps stop makes the system intrinsically safe. This may mean stopping the heating medium flow into the system.

A.3.3 Liquid-vapor Separation Devices

The sizing of the vapor space above the liquid level of the steam drum will determine the amount of water entrainment within the export steam – some entrainment is inevitable. Where the process demand is for very dry steam it will be necessary to install separation devices to minimize liquid entrainment. The type and size of the separation device can be tailored to achieve a specific steam quality for the given export flow. Separation equipment will create pressure drop and this could be significant when selecting the separation device.

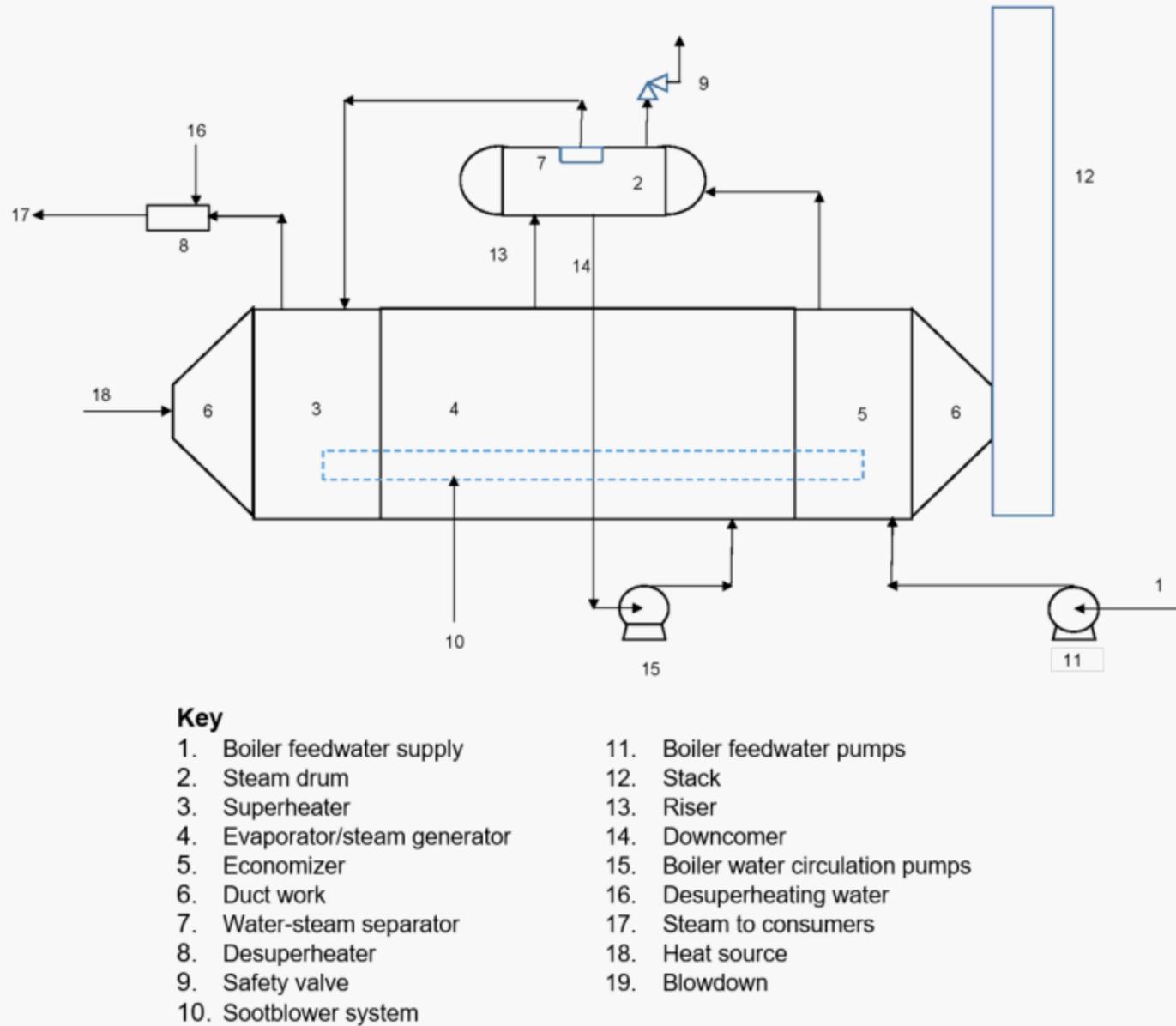


Figure A.1—Typical Steam System Components as Part of a Watertube Steam Generator

A.3.4 Downcomers

A.3.4.1 In a natural circulation system, the downcomers carry boiler water at the saturation temperature from the steam drum to the evaporator. The elevation of the steam drum provides the static head that drives the boiler water through the steam generation circuit. The number and size of downcomers must be determined according to the thermal and hydraulic requirements of the system.

A.3.4.2 In a forced circulation system, the downcomer will normally be a single line to the boiler water circulation pump(s). The static head gained by elevating the steam drum must exceed the required NPSH of the pump combined with the frictional losses in the downcomer to ensure vapor generation does not occur at the pump inlet.

A.3.5 Risers

The riser is the portion of the circuit which carries the steam/boiler water from the evaporator to the steam drum. A riser can be heated, or unheated, depending upon its contact with the heating medium.

A.3.6 Evaporator

A.3.6.1 The heat applied to the evaporator/steam generator causes the saturated boiler water to vaporize. In a natural circulation system water flow through the evaporator will normally be vertical. Sloped systems can also be used. The flow generated by the thermosiphon effect is a function of the amount of the heat recovered/

steam generated in the evaporator and the fluid hydraulics within the circuit. The total flow rate through the circuit relative to the flow rate of steam produced in the evaporator is known as the circulation ratio (CR). The designer must select the CR and heat flux so that phase separation does not occur within the evaporator which can lead to overheating and failure of the tubes. In a bank of evaporator tubes with a large reduction in heating medium temperature across the bank, consideration must be given to the variation in heat fluxes and steam generation rates within the various parts of the evaporator.

A.3.6.2 In forced circulation systems the evaporator tubes are more commonly horizontal. These circuits operate with higher velocities to stabilize the boiling and characteristically have fewer parallel heating paths and typically have multiple passes of each path through the heated zone. Where there is a change in elevation between inlet and outlet of the heating coils then the direction of flow should be upwards through the coil bundle. Since hydraulic system losses are overcome by the boiler water circulation pump the designer has greater flexibility in terms of circulation ratios and mass velocities. The designer should select a mass velocity high enough to ensure even flow distribution to each parallel pass.

A.3.7 Economizer and Superheater

A.3.7.1 Economizers are used to maximize heat recovery by preheating the BFW before it enters the steam drum. Economizers will be located at the back end of the heat recovery system usually at the point where there is no longer sufficient heat in the heating medium to generate steam. When using exhaust gas from a combustion source as the heating medium consideration must be given to the dew point of exhaust gases, particularly those containing sulfur species. Cold end corrosion is a common problem for low-temperature economizer surfaces. The BFW is usually co-current with the heating medium to keep the tube wall temperature above the acid gas dew point temperature. Depending on the layout there is more risk of damage due to flow instability or deposition occurring, therefore, economizers are normally not designed to vaporize any water.

A.3.7.2 The saturated steam leaving the steam drum, even very dry steam, can be cooled as it is transferred to the consumers causing condensation and the formation of water droplets to occur. This is undesirable in high velocity steam piping and mechanical consumers such as steam turbines where the impact of water droplets can cause erosion and damage. To prevent this saturated steam is often passed through a superheater and heated to a temperature above saturation that is sufficient to ensure the steam remains dry when it reaches the consumers. Superheating also increases the power that can be derived from the steam in steam driven equipment. In some systems, steam from external sources are added prior to entering the superheater.

A.3.7.3 The superheater is usually located in the hottest part of the heat recovery system, upstream of the evaporator. In some instances when the superheater may have to run-dry (i.e. no flow condition) and/or the inlet temperature of the heating medium is very high, some evaporator surface may be installed upstream of the superheater to provide some shielding. In this case, the heating medium temperature must be below the maximum allowable tube metal temperature.

A.3.8 Circulation Pump and Flow Regulation

A.3.8.1 In a forced circulation system, a pump is used to circulate the boiler water through the evaporator. The pump should be sized to provide sufficient flow to meet the required circulation ratio and generate sufficient head to overcome the pressure drop of the evaporator circuit. The NPSH required by the pump must not exceed the NPSH available. In critical service the pumps are often spared as loss of circulation flow will likely result in shut-down of the steam generation system.

A.3.8.2 The circulation flowrate will usually be controlled by either variable speed drive or a flow control device in the circulation line. Flow measurement and control should ensure that the design circulation ratio is achieved, and low-flow safeguards should cut off the heat input to the heating coils if the minimum safe circulation flowrate is not achieved.

A.3.9 Desuperheating

A.3.9.1 It is sometimes necessary, e.g. for turbines, to produce superheated steam to protect the equipment or provide temperature control. However, in thermal process applications it may be preferable to use steam closer to saturation temperature as this is usually more efficient for heat transfer and allows for lower heat exchange surfaces with lower design temperatures. Therefore, a proportion of the superheated steam produced may be de-superheated for use in such applications.

A.3.9.2 Desuperheating can be achieved by indirect heat exchange but more commonly it will be achieved by direct injection of water into the steam flow, either by means of a venturi or spray-type nozzle. The water should be high quality e.g. BFW or condensate. The impact of possible trace chemicals in injection water should be considered in these systems.

A.3.9.3 Desuperheating may also be used as a method of controlling the steam outlet temperature where there is no independent means of controlling the heat input into the superheater of the steam generator. It can reduce the dryness of the steam so intermediate desuperheating, located between banks of superheater surface, can be used to counter this.

A.3.10 Pressure Safety Valve (PSV)

Pressure safety valves (PSV) are an integral part of any steam system and are designed to prevent the system pressure from exceeding the MAWP of the system. A PSV is a spring-loaded device that is designed to open and vent steam from the system at a predetermined factory-set pressure. Typically, a steam system will have multiple PSVs.

NOTE Refer to API Standard 521 and other relevant jurisdictional codes for more information including sizing.

A.3.11 Steam Traps

Steam traps are used as a means of removing condensate and non-condensable gases from the steam distribution mains/headers to ensure the steam reaching the consumers remains dry and reduces the risk of water hammer developing within these lines. Steam traps allow the constant removal of condensate whilst preventing the escape of steam. The mechanism can either be mechanical, thermostatic, or thermodynamic. Properly installed float or thermostatic steam traps are also used to bleed air and other non-condensable gases from the steam system during start-up. They may also be used for the collection of condensate from steam consumers to route the condensate into the collection system. In some cases, they are used in lieu of condensate pots; in these cases, caution should be applied to eliminate water hammer.

A.3.12 Condensate Recovery

A.3.12.1 Condensate is produced when heat is released by condensing steam and then the condensate is collected by the use of either condensate pots or steam traps. The high cost of treating the water and the sensible heat remaining within the condensate means recovery of condensate for reuse is usually provided. Condensate will be collected in a receiver and then transported into the boiler feed vessel by pumping in a pressurized system. In the case of an atmospheric feed tank the pressure at the steam trap may be sufficient to return the condensate to the feed tank but some flashing at atmospheric pressure will occur and that steam produced will be lost from the system.

A.3.12.2 In plants with multiple steam consumers requiring different steam pressures the condensate from the high-pressure system may be flashed to produce medium or low-pressure steam that can be exported to the lower pressure steam mains/headers. Steam turbines also lower the steam pressure through operation, and this can be exported directly to the lower pressure steam main for further use after removing the condensate.

A.3.12.3 Care should be exercised when mixing water/condensate streams of different temperatures to avoid creating a water hammer.

A.3.13 Duct Work

Duct work may be provided between sections on the heating medium side to transport the heating medium. This ductwork design should provide uniform flow distribution of the heating medium, minimize heat loss to the atmosphere by using refractory linings or external insulation, and allowing for proper expansion of the ductwork in all directions often by utilizing expansion joints.

A.3.14 Trim, Instrumentation, and Controls

A well-designed system needs to provide trim connections to the fluids on both sides of the steam system to allow for installation of instrumentation and controls to monitor the operation of the system and to provide for operation and safety controls.

A.3.15 Online Cleaning/Sootblowing Systems

If the heating medium contains materials that can foul the heat transfer surfaces, an online cleaning system may be required. Sootblowers are typically used for this purpose, but other systems exist. The design of an online cleaning system needs to consider:

- the amount of fouling expected;
- the type of foulant that needs to be removed;
- the frequency of cleaning required;
- the impact of the cleaning medium, including on the heat transfer surface and supports.

Some other types of systems used include sonic horns, vibration (hammering), and water washing.

A.3.16 Pressure Design Codes

A.3.16.1 Steam systems are designed and fabricated in accordance with the Pressure Design Code. However, there may be jurisdictions where the use of a specific Boiler Code will be required in place of the Pressure Vessel Code. For example, ASME *BPVC Section VIII, Division 1* and ASME B31.3 or ASME *BPVC Section I* and ASME B31.1 could be required based on the jurisdiction's requirements.

A.3.16.2 Piping components in most configurations are designed and fabricated in accordance with the relevant Piping Code.

A.3.17 Blowdown Systems

This type of system includes valves, analyzers, and controls to regulate the boiler water chemistry.

A.4 Advantages and Disadvantages of Utility Steam

A.4.1 Advantages

Utility steam has the following advantages.

- a) Raw water is readily available and inexpensive compared with other heat transfer mediums.
- b) Steam is a nonflammable, nontoxic medium with low environmental impact (when used in closed systems).
- c) Steam carries a large amount of energy per unit mass.
- d) Transferring heat using latent heat is a very efficient process allowing for the reduction in size and capital cost of heat transfer equipment.
- e) Steam maintains a constant temperature as heat is transferred, resulting in even heat transfer.
- f) Versatility – A single steam generator installation can be used to derive steam at multiple pressure/temperature levels to suit a variety of consumers.
- g) Efficiency – With an economizer, a large proportion of the heat source available can be recovered and an efficient condensate recovery system can mean very little heat is wasted.

A.4.2 Disadvantages

Utility steam has the following disadvantages.

- a) A steam system has a relatively high capital cost compared with other systems. Depending on the source water quality, the system may need extensive water treatment facilities to improve the water quality to that needed for boiler feedwater.
- b) System design is complex and can result in operational issues if not performed correctly (e.g. inadequate condensate removal, water hammer, and equipment damage).
- c) Inspection and maintenance demands are more severe.
- d) Winterization concerns must be addressed for cold weather climates.
- e) Water may be scarce at some locations.
- f) Steam systems are corrosive and must be designed and treated properly.

Annex B (informative)

Hot Oil and Other Heat Transfer Fluids

B.1 General

Hot oil and other liquid phase heat transfer fluid (HTF) systems are commonly used to provide heat to process fluid services as an alternative to steam heating systems or the use of direct fired process heaters. As compared to steam heated applications, a properly designed HTF system can provide for lower capital and operational costs due to lower operating pressures and the elimination of large diameter vapor piping, pressure control equipment and devices, condensate handling systems, and BFW treatment equipment and chemicals. Compared to direct fired heaters, HTF systems can provide more uniform heating to temperature sensitive process streams, eliminating local hot spots and providing improved accuracy of temperature control. It also allows for a single heating source (fired heater or heat recovery exchanger for the heat transfer fluid) to be located remotely from the process unit and allows for heat to be provided to multiple process services, at different temperature levels, as well as being able to provide both process heating and cooling needs.

B.2 Commercial Heat Transfer Fluids

B.2.1 General

There are many heat transfer fluids that are available as commercial products. These include a number of mineral oils as well as synthetic aromatic or silicone-based fluids. There is no single best HTF for all applications, and different fluids are designed for use for specific applications and technologies and for distinct temperature ranges from $-115\text{ }^{\circ}\text{C}$ to $+400\text{ }^{\circ}\text{C}$ ($-175\text{ }^{\circ}\text{F}$ to $+750\text{ }^{\circ}\text{F}$). Heat transfer fluids are designed with optimized properties to provide efficient heat transfer at their intended operating temperatures including high specific heat, density, and thermal conductivities while having low viscosity. They have low vapor pressures and high boiling ranges, which permit operation at low system pressures and at their maximum recommended operating temperatures.

B.2.2 Decomposition of Heat Transfer Fluids

All commercial HTFs are subject to time and temperature dependent decomposition. The HTF supplier provides maximum bulk fluid and film temperatures for the fluid which should be strictly observed. Decomposition of the fluid results in the formation of components which have both lower molecular weights and boiling points (low-boilers) and higher molecular weights and boiling points (high-boilers) than the original fluid. Low boilers can reduce the flash and fire point of the HTF and should be periodically vented from the system. High boilers are generally soluble in the bulk fluid to a certain degree, above which the fluid would need to be replaced. Online purification systems can also be used to limit the amount of accumulation of high-boilers in the fluid charge. When operated below the maximum recommended bulk and film temperatures, and not exposed to moisture, air, or process fluid contamination, the HTFs should provide years of fluid service life. The fluid should be sampled regularly, however, for quality in accordance with the recommendations of the HTF supplier. The frequency of sampling will depend upon how close the actual operating bulk and film temperatures are to the maximum recommended values, or if operating upsets or process fluid contamination is thought to have occurred.

B.2.3 Use of Process Fluids as Heat Transfer Fluids

On occasion a process stream can be used as a heat transfer fluid. This is sometimes done to avoid using a commercial heat transfer fluid.

B.2.4 Molten Salts

Various molten salts are available as thermal fluids to suit the specified duty. Their temperatures are raised to the required levels in the thermal fluid heater. The temperature range selected is an important criterion when designing a thermal fluid system.

B.2.5 Molten Salt Heat Recovery Systems

B.2.5.1 The HTF heater is a critical component of the system which can be fuel fired or electrically heated.

B.2.5.2 At temperatures up to 600 °C (1100 °F), molten inorganic salts are suitable thermal fluids for supplying heat to various heat recovery systems. A typical system is shown in Figure B.1. A eutectic mixture of salts with a melting point of 142 °C (288 °F) is often used. The salt remains in the liquid phase under the operating conditions.

B.2.5.3 Salt allows media temperatures of over 500 °C (900 °F) to be achieved without the need for elevated pressures to prevent evaporation/vaporization.

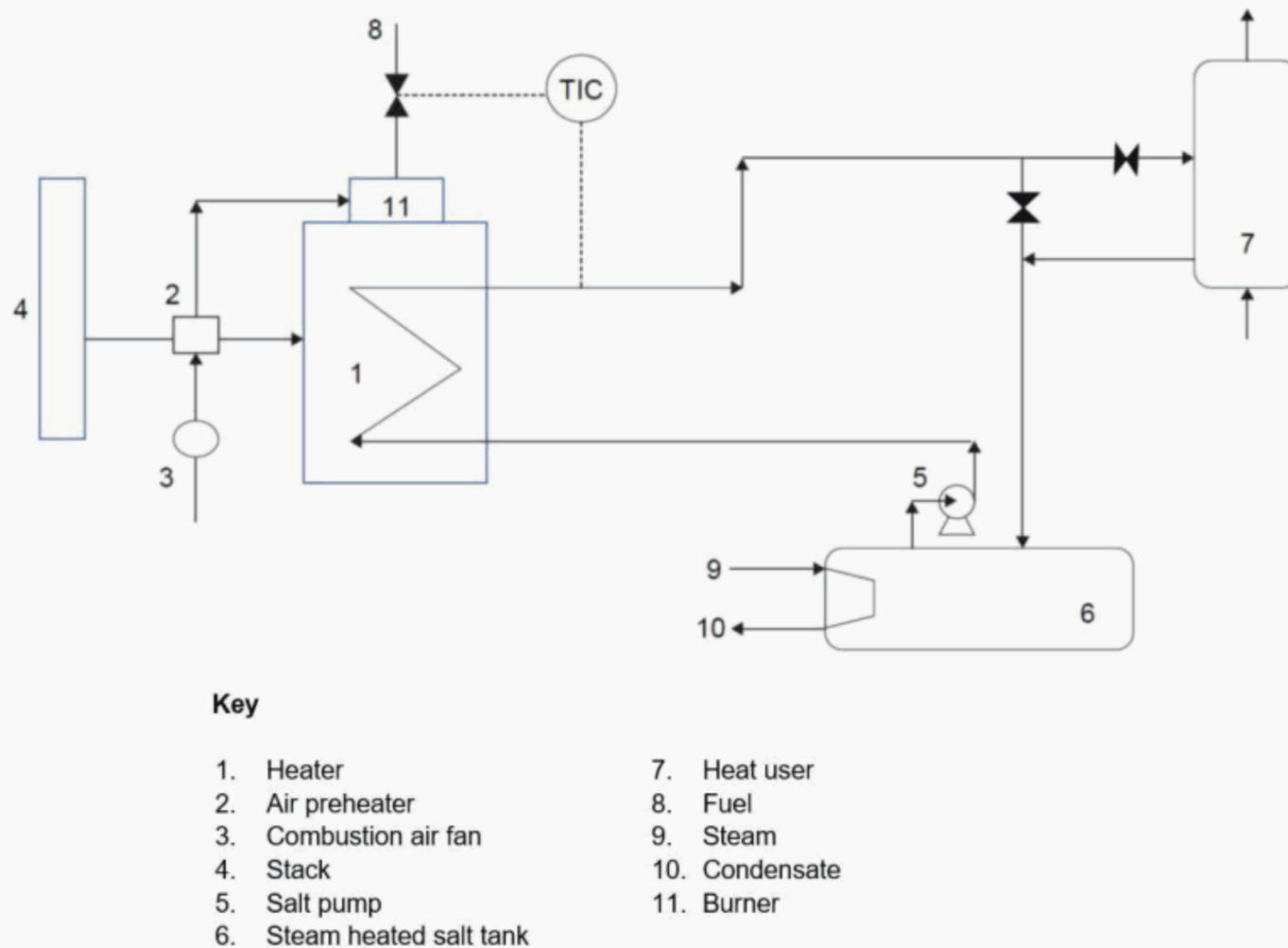
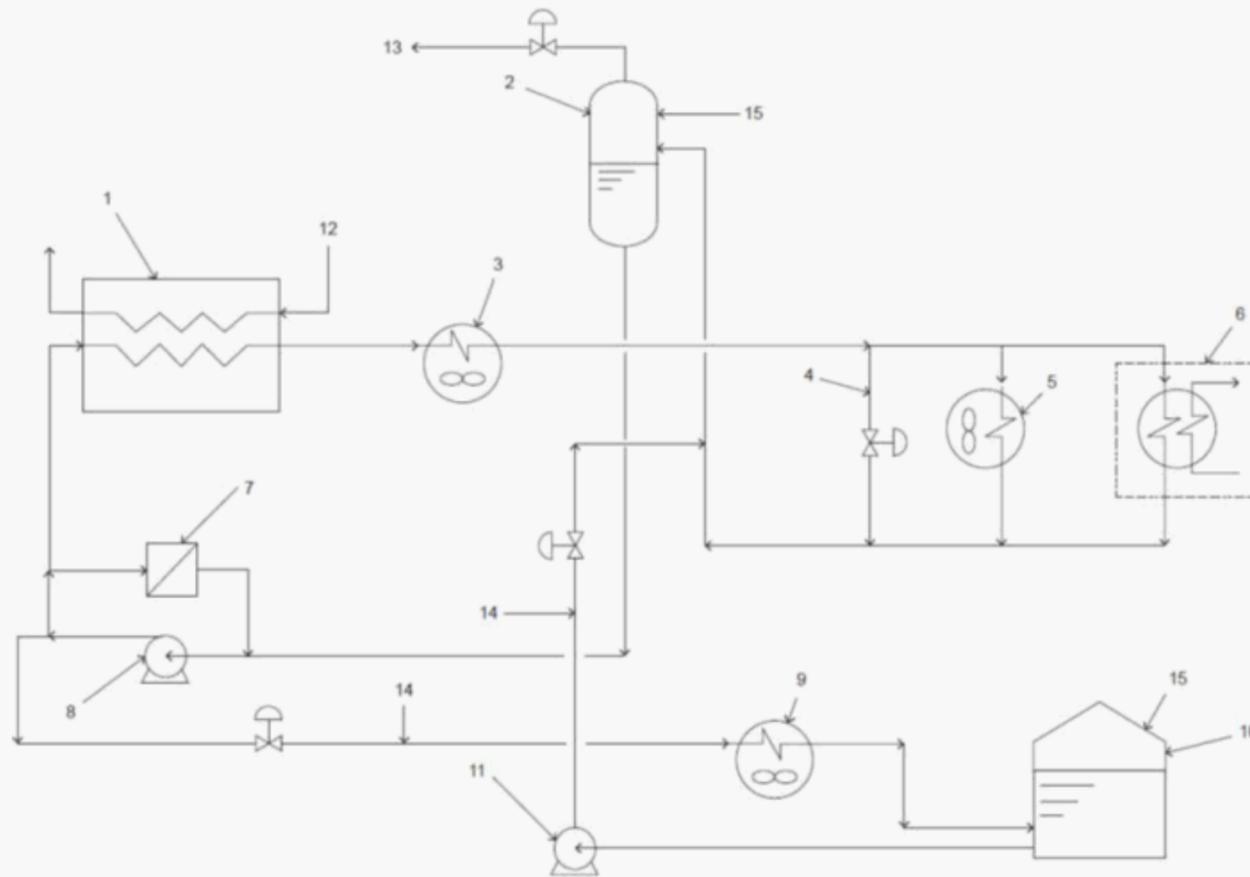


Figure B.1—Typical Molten Salt System

B.3 Heat Transfer Fluid System Components

B.3.1 General

Figure B.2 shows a sketch of a typical HTF system with a waste heat recovery unit. A properly designed HTF system will normally consist of the following components: HTF heater, expansion tank, circulation pumps and piping, strainers and filters, and the process heat exchangers (heat consumers). Depending upon the range of process heating demands and the heating source used, trim heaters, trim coolers, or dump coolers may be required. Storage tanks with makeup pumps, piping, and pump-out coolers may be provided when an inventory of HTF is to be provided on site. The following are guidance for these components.



Key

- | | |
|-----------------------------|----------------------------|
| 1. Waste heat recovery unit | 9. Pump-out cooler |
| 2. Expansion vessel | 10. Storage tank |
| 3. Trim cooler | 11. Make-up pump |
| 4. Full bypass line | 12. Heater or waste heat |
| 5. Dump/trim cooler | 13. To flare |
| 6. Heat consumers | 14. Normally no flow (NNF) |
| 7. Side stream filter | 15. Nitrogen |
| 8. Circulation pump | |

Figure B.2—Typical Heat Transfer Fluid System with Waste Heat Recovery Unit

B.3.2 Heat Transfer Fluid Heater

B.3.2.1 The HTF heater is a critical component of the system. Fuel fired or electrically heated furnaces are commonly employed to provide heat input to the system, but systems using waste heat from gas turbine exhaust or other high-temperature process waste heat streams can also be used and provides for an economic way to distribute and use this otherwise waste heat throughout the process unit. The design of the HTF heater is critical, as this is the location of the maximum fluid film temperature as well as the maximum fluid bulk temperature. The HTF velocity over the heat transfer surface should be maximized as this will result in lower skin temperatures, and dead flow zones which result in high residence times should be avoided. For this reason, liquid tube type heaters, with the HTF contained within coils or tubes, are often preferred. The effect of nonuniform heating, as may occur for example in fired heaters, must be considered when determining the maximum local fluid film temperature.

B.3.2.2 When a fired or electric heater is used to provide heat input to the HTF, the normal turn-down operability of these heater types allows for the heat input to the system to be matched to process heating demands which may be variable. A bypass line parallel to the consumers is provided to control the flow and supply pressure in the system due to consumer trips or demand changes and allows for the minimum flow of HTF through the heater to prevent exceeding the maximum recommended film temperatures. For widely variable systems, an additional bypass line with a trim cooler can be provided for increased operational flexibility. Heaters should be configured with both a high HTF temperature cut-off as well as a HTF low-flow cut-off.

B.3.2.3 When waste heat is recovered for use in the HTF system, from a gas turbine exhaust stream for example, it may not be possible to control the HTF system heat input to match the demand needs of the process. These systems may require supplemental heating devices or equipment such as auxiliary firing duct heaters or trim heaters to provide the necessary heat input, either for the design condition or when the gas turbine is running at lower load or is unavailable. Conversely, an additional trim cooler may be required to dissipate the excess heat when low process heating demand exists at times when the gas turbine output cannot be reduced.

B.3.3 Expansion Tank

B.3.3.1 The HTF system expansion tank provides sufficient surge volume for fluid expansion from its cold start-up condition to its operating temperature condition, which can result in an increase of fluid volume of 25 % or more depending on the specific fluid and operating conditions. The design inventory may also consider the volume of certain system consumers, to allow these to be taken out of service for maintenance while the remainder of the system remains in operation without having to partially drain the system. The tank also serves as the main vent location for the system from which the low boilers can be removed. The expansion tank is typically installed at the highest point in the system and is connected to the suction side of the circulation pumps, for which it provides net positive suction head. Level controls should be provided on the expansion tank. Low level control should provide for protection from the loss of pump suction or pump stoppage. Low level control should also cut off heater input in the event of a loss of fluid volume from the system. High level controls should be provided to allow for the vapor space to be maintained to prevent over-filling, and to indicate an increase in system volume due to a contamination of the HTF from process fluid in-leakage.

B.3.3.2 Due to potential HTF contamination and degradation resulting from oxidation or excessive moisture, the expansion tank should not be left open to the atmosphere. An inert gas (e.g. nitrogen) blanket should be maintained over the vapor space in the expansion tank to minimize fluid oxidation, although cold fluid seal traps can also be used where an inert gas blanket is not practical to provide. The pressure in the expansion tank is typically maintained at a minimum of 100 kPa(g) (15 psig) above the vapor pressure of the HTF.

B.3.4 Circulation Pumps

The HTF circulation pumps must have sufficient capacity and pressure head to serve the needs of the system and must consider the wide range of operating temperatures and viscosities that will be encountered. For large systems, centrifugal API pumps with a stand-by spare are commonly used. Because of the high-temperature service, fluid cooled bearings and seals are recommended, and mechanical seals are commonly employed. The spare pump is normally maintained in a hot stand-by condition due to the high operating temperature of the system. Seal-less pumps, e.g. magnetically-driven and canned motor pumps), also can be used. Proper control system design requires that heat input to the heater be stopped in case of pump failure.

B.3.5 Piping

B.3.5.1 Since the degradation of the HTF increases exponentially with operating temperature when near the maximum allowable bulk temperature, the HTF supply header piping to the consumers should be as short as possible. However, piping design in HTF systems requires attention to the piping flexibility that is necessary to provide for adequate thermal expansion and contraction due to the temperature changes that the system will undergo. The operating temperature of the system is normally above the heat transfer fluid's flash and fire points. Leaking HTF can represent a significant fire hazard when the piping insulation becomes fluid saturated at the high operating temperatures and the fluid undergoes oxidation due to porous insulating materials. For this reason, the use of close-celled high-temperature insulation systems (e.g. cellular glass) that resist fluid saturation are often used. The number of flanged and other mechanical joints should be minimized to prevent leakage, and threaded fittings should be avoided. Valves and stem seals should be chosen specifically for the high operating temperature and the desire to minimize fluid leakage. High temperature packing may be used, but metallic bellows stem seals are often specified for leak minimization. Suitable vents and drains should be

provided in the system to allow for the system to be properly dried prior to commissioning. Closed drain systems for the HTF are normally provided.

B.3.5.2 Strainer baskets should be installed in the pump suction for debris removal during start-up and can be removed for normal operation. Some systems, however, also include side-stream filters for the continual filtration of the system of particulates in the 10- to 20-micron size range as this will prolong the life of the system components and will reduce fouling in the heater and consumers.

B.3.6 Storage Tank

When it is desirable to provide for an on-site inventory of fresh HTF, a storage tank may be provided. The system would include a tank, makeup pump, and a pump-out cooler to cool the heat transfer fluid prior to drainage to the storage tank. The pump-out cooler can be omitted if there is a suitably sized trim cooler included in the HTF system that may be used. The storage tank should be maintained under an inert atmosphere (e.g. nitrogen blanket) to prevent oxidation of the fluid and moisture ingress. The minimum fluid level in the storage tank should be adequate to provide the necessary net positive suction head for the makeup pump. The storage tank may require internal or external heating coils, if required to maintain the HTF inventory above its pumpability temperature for sites with low ambient temperatures.

B.3.7 Heat Exchangers

B.3.7.1 Conventional shell and tube and hairpin type heat exchangers can be used in hot oil and other types of heat recovery systems. Refer to API Standard 660 and API Standard 663 for further details on these types of equipment. Other types of heat exchangers can also be used in these types of systems.

B.3.7.2 The process heat exchangers (consumers) should be designed for efficient heat transfer. Exchangers in HTF service are often subject to thermal shock and rapid temperature changes at start-up. In addition, there may be large temperature gradients between the HTF and the process fluid that may occur during both start-up and normal operation. The mechanical details and selection should be chosen with these conditions in mind to minimize the potential for fluid leakage, either external or inter-stream.

B.3.7.3 When process fluid temperature control is necessary, the HTF flow to each heat exchanger is normally regulated to meet the heat input demand of the process fluid while the supply temperature of the HTF system remains fixed through the control of the HTF heater. Where there may be several consumers requiring heat input at different temperature levels, a zoned temperature system can be provided to serve these exchangers. Each temperature zone would include its own recirculation pump upstream of the exchangers and a bypass line from the outlet of the consumers back the recirculation pump suction. This allows for the blending of the cold HTF from the exit of the consumers with fresh hot HTF to provide temperature control and a lower HTF supply temperature to this zone than that provided for the main system.

Annex C (informative)

Hot Water/Glycol Systems

C.1 General

Hot water utility systems are similar in design and operation to hot oil systems, with a few notable differences. For instance, hot water systems are normally designed to operate at more moderate operating temperatures than hot oil systems. They are often used to provide heat for general utility services as opposed to process heating demands. They are also used where cold utility water temperatures cannot be tolerated by the process fluid, or at temperatures above which the utility water should be exposed, often called tempered water in these services. In colder systems, glycol is added to the water to lower the freezing point of the mixture.

C.2 Hot Water Systems

C.2.1 Hot water systems are typically closed loop recirculating units that use demineralized water, often treated with anti-corrosion chemicals. When used in locations with low ambient temperatures, the addition of anti-freezing chemicals (e.g. ethylene glycol) may also be considered. The water must be of sufficient quality to allow it to be used at the elevated system bulk and skin temperatures while avoiding significant fouling. Water sampling facilities, with sample coolers, should be provided to allow regular water quality sampling and to allow for the detection of hydrocarbons or other process fluids that may ingress into the system through leaks in the heat exchangers.

C.2.2 Although hot water systems are not subject to the time and temperature dependent decomposition that is prevalent in hot oil systems, the temperature and operating pressure in the system must be controlled nevertheless to prevent unintended vaporization or 'steaming' of the water within the heater which can lead to the deposition and concentration of any impurities in the system or in water hammer issues. To safeguard the system from the risk of steaming, the operating pressure of the system should normally be maintained such that the maximum hot water skin (surface) temperature at all operating conditions is a minimum of 30 °C (50 °F) below the saturation temperature at that pressure. When corrosion treatment chemicals are added to the system, any maximum temperature limits prescribed by the treatment system providers should also be adhered to.

C.3 Hot Water System Components

C.3.1 General

The components of a hot water system are similar to that of hot oil systems. Hot water systems normally use the facility's existing demineralized water system for makeup, and do not require the on-site storage tanks, makeup pumps and run-down coolers that are common for hot oil systems. When no demineralized water system is present, the initial and future system fills may be provided by trucks from a contracted supplier. In addition, a hot water system does not require a purification system as a hot oil system might, although a chemical dosing system may be required based on the water quality and selected system metallurgy for corrosion control. Other differences in components of hot water and hot oil systems are noted below.

C.3.2 Heater

The system heater provided for hot water systems is usually either an electric heater or a waste heat recovery coil from, for example, a gas turbine system. Direct fired heaters are normally not provided due to the more moderate operating temperatures of the water system. In waste heat recovery units, the hot water coils can be located

downstream of other coils, improving the overall efficiency of the system, although auxiliary heating coils in the waste heat recovery unit (WHRU) exhaust duct may be required to meet the hot water system demand during operation of the turbine at reduced loads. High water temperature and low water flow instrument protection is normally provided to safeguard the system against steaming or water hammer.

C.3.3 Expansion Tank

The design of the expansion tank for a hot water system is similar to that of a hot oil system. Nitrogen is normally used for pressurization of the hot water system with high-pressure nitrogen supplied to the expansion vessel. Avoid dissolved oxygen in the system which can cause corrosion over time. A secure and reliable nitrogen supply system should be provided, as loss of pressure in the expansion tank will cause flashing of hot water in the circuit leading to disruption of the plant operation and potential risk of water hammer and consequential loss of containment due to damage to the system. This may require the use of redundant nitrogen compressors or the use of high-pressure nitrogen cylinders. In addition, the expansion tank design should reduce the risk of water carry over from the expansion vessel into the flare system. One scenario that should be considered is a tube rupture inside heat transfer equipment operating at process side pressures above that of the hot water circuit. The process fluid leaks into the hot water side, and then the sudden displacement of fluid from the hot water circuit into the expansion tank can result in water carry over to the flare system.

C.3.4 Trim Cooler

For hot water systems with a WHRU in the exhaust of a gas turbine, a trim cooler should be provided to reject the maximum excess heat possible in the system under all operating conditions including the heat consumer(s) trip scenarios. To eliminate the risk of steaming in the water heating coils due to loss of heat demand in the process and subsequent rise in hot water return temperature, an additional dump cooler is typically used to reject excess heat in the hot water return circuit. The dump cooler is typically sized based on the maximum duty to be dissipated when the hot water is cooled down to the required WHRU inlet temperature from the maximum supply temperature. Then the trim cooler duty should be based on the balance excess heat duty that is required to be dissipated from the system. In the designs with several heat consumers, the trim cooler alone may be sized for the maximum duty absorbed by the coil without a dump cooler when the temperature rise of the hot water is not significant on loss of heat demand from the largest consumer.

C.3.5 Piping System

The hot water piping system design may need to consider measures to minimize the possibility of flow accelerated corrosion (FAC) which is a common phenomenon in demineralized water systems. This typically includes limiting velocities to less than 2.4 m/s (8 ft/s). Higher values are sometimes used when using carbon steel metallurgy, minimizing turbulent areas in the piping system by component design and elimination of FAC as far as practical. The use of higher alloy steels with low levels of chromium ($\geq 1\%$) in the FAC prone areas may also be considered.

C.3.6 Glycol

To handle systems with temperatures where water by itself could freeze, glycol is added to the water to reduce the freezing point. Ethylene glycol and propylene glycol are commonly used. The freezing point is reduced per Table C.1 and Table C.2.

NOTE 1 Combining propylene and ethylene glycol into a single mixture is not recommended.

NOTE 2 Chemical treatment of the system may be required.

NOTE 3 Due to slush creation a glycol and water solution should not be used close to freezing points [i.e. maintain a margin of at least 5 °C to 10 °C (10 °F to 20 °F)].

NOTE 4 Environmental and health guidelines are to be considered when handling any glycol. Propylene glycol is considerably less toxic than ethylene glycol.

Table C.1—Freezing Points of Ethylene Glycol Based Water Solutions

| | | | | | | | |
|-----------------------|----|----|----|-----|-----|-----|-----|
| % By Volume | 0 | 10 | 20 | 30 | 40 | 50 | 60 |
| Temperature °F | 32 | 26 | 18 | 7 | -10 | -34 | -63 |
| Temperature °C | 0 | -3 | -8 | -14 | -23 | -37 | -53 |

Table C.2—Freezing Points of Propylene Glycol Based Water Solutions

| | | | | | | | |
|-----------------------|----|----|----|-----|-----|-----|-----|
| % By Volume | 0 | 10 | 20 | 30 | 40 | 50 | 60 |
| Temperature °F | 32 | 26 | 18 | 7 | -8 | -29 | -55 |
| Temperature °C | 0 | -3 | -8 | -14 | -22 | -34 | -48 |

Annex D (informative)

Watertube WHRU Configurations

D.1 General

The watertube heater generates steam or transfers heat to other utility systems or fluids inside several tube circuits which are heated by a hot gas stream flowing through an enclosure of insulated steel casing plates. The gas usually flows across the tubes in a single pass. In certain cases, baffles or directional vanes may be used to direct the gas across the tubes creating additional gas passes. Steam generating tubes are connected to drums or headers. The tubes may be arranged in one continuous circuit or may be manifolded at their inlet and outlet ends to form several parallel flow paths. Steam drums may be either integral to the steam generating tube circuit or mounted remotely from the tubes. Additional tube circuits may be used for preheating feedwater or superheating steam. The flow of the heating gas can be directed along the length of the tubes (longitudinal flow) or flowing across a bank of tubes (cross flow). Longitudinal flow is more common in small or dirty gas services. Cross flow is more common in larger units or those that involve multiple services like superheaters, generators, and economizers.

NOTE Many of the functionality and mechanical design requirements for watertube steam generators are covered in API Recommended Practice 538.

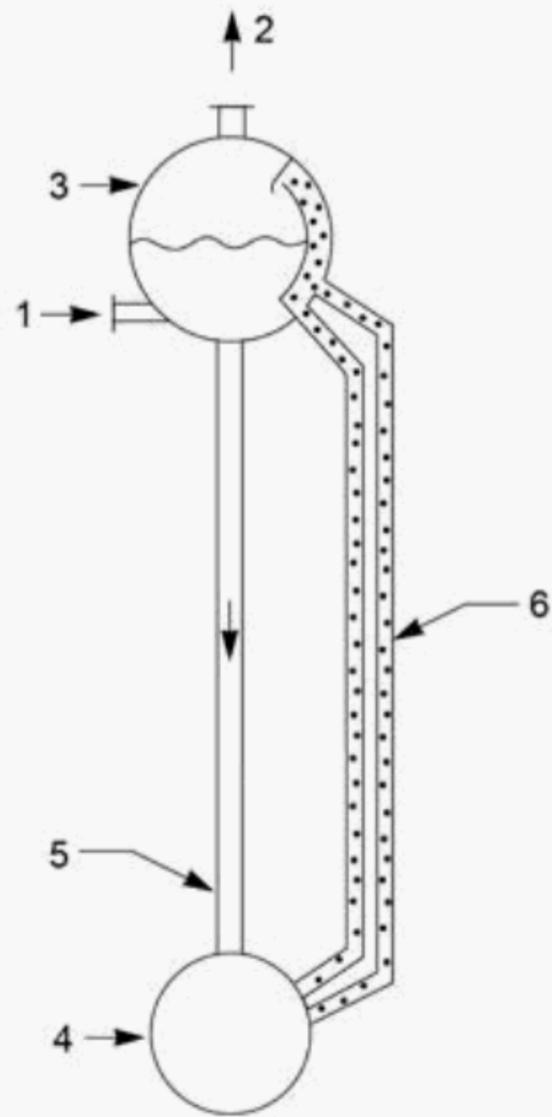
D.2 Waste Heat Sources

There are numerous possible sources of heat to waste heat watertube configurations. They may come indirectly from combustion equipment. Examples of this are:

- a) thermal oxidizers,
- b) fired process heaters,
- c) gas turbines, and
- d) fluid or residual catalytic cracking units.

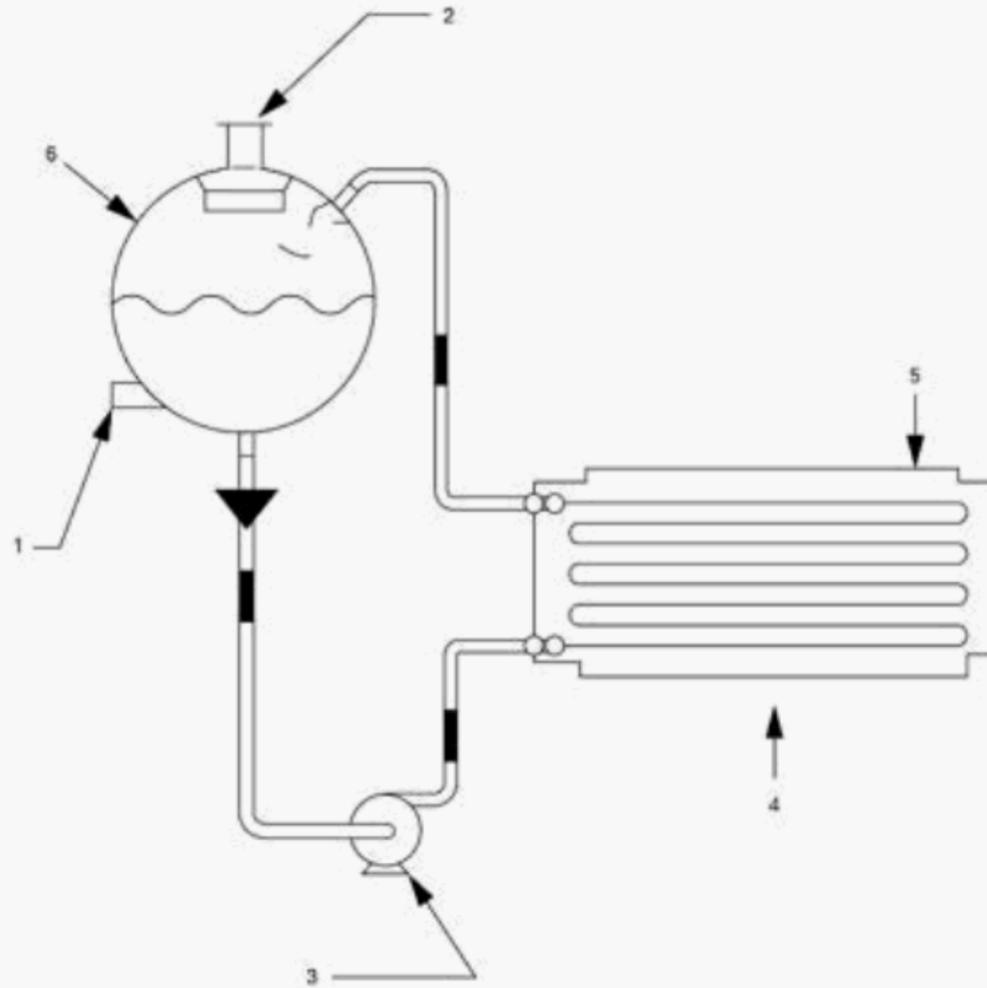
D.3 Utility Fluid Circulation Configurations

There are two means that the water or other boiling heat mediums circulates between the heating surfaces and steam/separation drum. The water may circulate naturally by means of a thermosiphon effect (natural circulation) or by the use of pumps (forced circulation). Figure D.1 shows a typical natural circulation configuration system for a steam generating system. If a non-vaporizing heat transfer medium is used, there is no separation drum, so only forced circulation systems are used. Figure D.2 shows a typical forced circulation system. This one shows a steam generating system.

**Key**

- | | |
|---------------|---------------------------|
| 1. Water in | 4. Mud drum |
| 2. Steam out | 5. Downcomer |
| 3. Steam drum | 6. Steam generating tubes |

Figure D.1—Typical Natural Circulation Watertube

**Key**

- | | |
|---------------------|-------------------|
| 1. Water in | 4. Heating medium |
| 2. Steam out | 5. Heat exchanger |
| 3. Circulation pump | 6. Steam drum |

Figure D.2—Typical Forced Circulation System

Annex E (informative)

Firetube WHRU Configurations

E.1 Introduction

With the hot gases on the inside of the tubes and a cold HTF on the shell side, there are several considerations to design and operate a reliable WHRU. General design and fabrication of firetube equipment would typically be as per API Standard 660 but idiosyncrasies to the design will be covered in this section.

E.2 Advantages of Firetube vs Watertube Configurations

E.2.1 Tubes containing fouling-prone, hot process streams such as olefins plant cracking furnace effluent, coal gasifier overhead, fluid catalytic cracking flue gas, or condensing media such as sulfur plant process gas from the thermal reactor are easier to clean in firetube WHRUs.

E.2.2 Firetube WHRUs have lower process fluid volume and residence time for services where time at temperature must be limited.

E.2.3 High-pressure process fluids contained on the tube side may minimize WHRU weight in a firetube WHRU. This is particularly beneficial when more expensive, alloy materials are used. For example, ammonia converter effluent can be at 34,500 kPa(g) (5,000 psig) and requires alloy or clad materials. For this example, a firetube WHRU may be preferred.

E.2.4 Firetube WHRUs are less susceptible to damaging flow induced tube vibration or acoustic vibration when cooling large volumetric flow rate gas streams.

E.2.5 Elevated temperature gas which requires insulating refractory to avoid overheating pressure-containing components is often best handled in firetube equipment. This is particularly true for pressurized gas streams, which cannot be handled in rectangular duct enclosures. Refractory lining in firetube WHRUs is generally required only in the inlet channel compartment. In comparison, shell-and-tube type watertube WHRUs require more extensive refractory linings, which must be engineered to accommodate bundle insertion and removal.

E.2.6 Firetube WHRUs normally require less plot space due to their compact design. Horizontal firetube HRSGs with an external steam drum may have the drum mounted on the shell. The drum is supported by the interconnecting risers and downcomers, thereby eliminating costs associated with independent support.

E.3 Disadvantages of Firetube vs Watertube Configurations

E.3.1 Firetube WHRUs are not well suited for handling large volumes of near atmospheric pressure gases. Streams such as gas turbine exhaust require large cross-sectional flow area as provided by watertube coils installed in rectangular duct enclosures.

E.3.2 The use of extended surface (fins) against a low-pressure process gas can be an effective means of reducing size. This option is often used in watertube WHRUs, but is generally considered impractical for firetube designs, as the water side heat transfer rate is not governing the equipment's size.

E.3.3 For cases involving high-pressure steam, typically 10,400 kPa(g) (1500 psig) and above, firetube WHRUs require heavier wall shell cylinders and tubes. This is particularly true for high capacity systems. For this reason, firetube WHRUs in high-pressure steam systems weigh more than their watertube counterparts.

E.3.4 The hot tubesheet design of firetube WHRUs may be complex, particularly its attachment to the shell and the tubes. The severity of service relates to the coexistence of multiple conditions, such as the following.

- a) High inlet gas temperature.
- b) High pressure on the steam side.
- c) Loading imposed by the tubes due to axial differential thermal growth relative to the shell.
- d) Potential erosive effects of particulate bearing gases.
- e) Potential for corrosive attack from the process and steam sides.

E.3.5 The tubesheet is commonly made of Cr-Mo ferritic steels which require special attention during fabrication and testing. Many firetube WHRUs require a thermal and stress analysis to prove the construction acceptable for all anticipated operating conditions.

E.3.6 Watertube units can easily be outfitted for higher efficiencies than a stand-alone firetube unit, which may require additional equipment items (e.g. BFW preheaters or steam superheaters) to reach the same efficiency.

E.4 Tube Thermal Expansion

The tubes in a fixed tubesheet exchanger will have a higher mean metal design temperature than the shell. Hot tubes and a cooler shell will restrict thermal growth of the tubes and place them in compression. The allowable stress of tubes in compression is typically between 3 to 5 times less than the allowable stress for tubes in longitudinal tension. As a result, they will be prone to buckling and this may be alleviated by making the following changes.

- a) A shell expansion joint or a thin flexible knuckled tubesheet can be incorporated to minimize buckling loads on tubes.
- b) The use of U-tubes or floating heads allow the tubes to expand independently from the shell, thus eliminating differential thermal expansion concerns and the need for expansion joints. These types of designs are difficult to incorporate into firetube applications with elevated temperatures.
- c) Consider changing to a watertube design.
- d) Heat-up and cool-down procedures should consider the differential thermal expansion of the thicker shell material and the thinner tube wall materials during the transient operations.

E.5 Steam Separation

E.5.1 In making a decision on steam separation equipment style, the following issues should be considered.

- a) Comparison of horizontal and vertical HRSG in relation to performance, operational, weight, and life cycle.
- b) Comparison of natural circulation and once through HRSG on arrangement, performance, cycling behavior, and plant life.
- c) Fast start and transient operations.

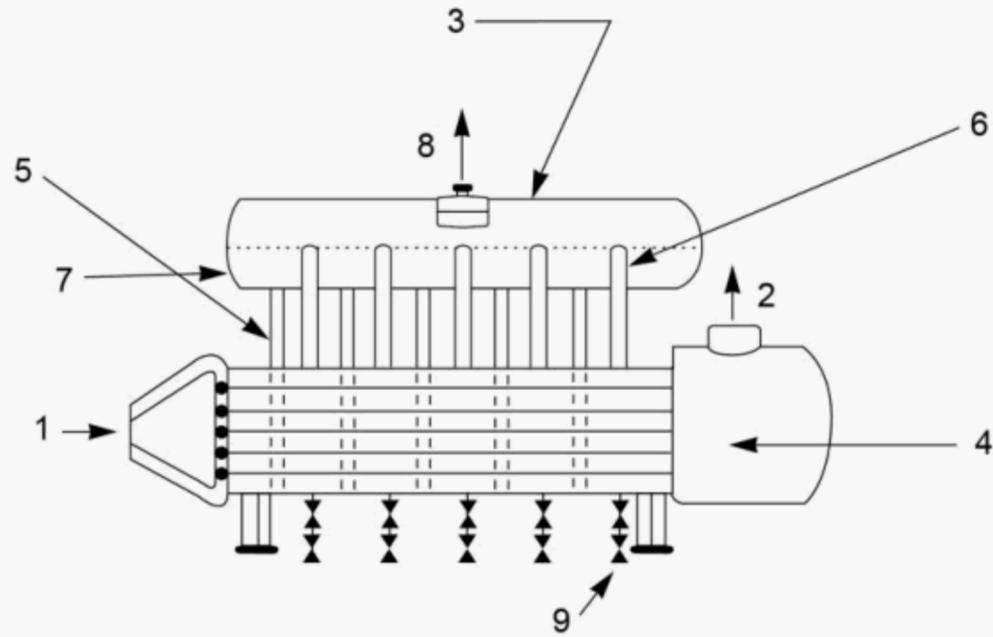
E.5.2 The water side must have sufficient surface area at the top of the liquid level to facilitate separation of steam and water droplets. Separate steam drums may be used for high-temperature/high-heat flux applications or where several WHRUs can be serviced by a single drum. There are several common configurations, some

of which are described in E.5.2 item a) through E.5.2 item c), although this list is not meant to limit the use of alternate designs or arrangements.

E.5.3

- a) Horizontal Firetube with External Drum HRSG—See Figure E.1 for a typical system. Sufficient riser pipes are required to allow the heated water/steam mixture to rise through the heat exchanger tube bundle and exit the shell into the steam drum above. Upon entering the steam drum, the liquid phase drops into the water (below the liquid level) while the steam can rise and exit the vessel. The water below the liquid level maintains a head pressure on the tube bundle and recirculates back into the shell side of the heat exchanger below. The steam drum ensures that the tubes are always fully submerged. The BFW (could be boiler feedwater or makeup condensate) inlet nozzle is only sized for the flow rate to replace the volume lost to steam and blowdowns.
- b) Vertical Firetube with External Drum HRSG—In Figure E.2, the arrangement includes a horizontal steam drum located above the heat exchanger. The interconnecting piping is limited to a single riser with upward flow and a single downcomer with downward flow. The steam is separated from the liquid in the horizontal steam drum. The number of interconnecting pipes can be reduced because the vertical nature of the exchanger provides greater differences in head pressure. The thermosiphon effect is greater in a vertical configuration. The flow is very much the same as with the configuration shown in Figure E.1 as heated water rises through the tube bundle and exits the shell into the drum above. As this steam/water mixture enters the drum, the liquid phase drops into the water level below while the steam vapor can rise and exit the top of the steam drum. The water maintains a head pressure on the tube bundle and recirculates back into the shell side of the heat exchanger below via the downcomer. The elevated steam drum ensures that the tubes are always fully submerged. The BFW (could be boiler feedwater or makeup condensate) inlet nozzle is sized for the flow rate to replace the volume lost to steam and blowdowns. Vapor blanketing at the underside of the top tubesheet is a major concern with vertical units. This must be addressed through suitable system and heat exchanger design features. If not addressed, vapor blanketing could result in dry-out of the tube surfaces leading to overheating or corrosion. See E.6.8 and E.6.10.2 for examples of ways to avoid vapor blanketing.
- c) Firetube Kettle Type HRSG—Figure E.3 shows a typical kettle style HRSG unit. In this case, the external steam drum is replaced with a larger shell section in the heat exchanger. The steam can escape directly from the water level, which is typically just above the tube bundle. This style may have a lower capital cost than systems with external drums, but they are more prone to operating problems during process upsets including changes in heat flux, due to their limited feedwater hold up volume. This can result in a fluctuating water level and lead to bundle dry-out. Kettles also have reduced steam volume available and are more prone to water carry-over or entrainment with the steam flow.

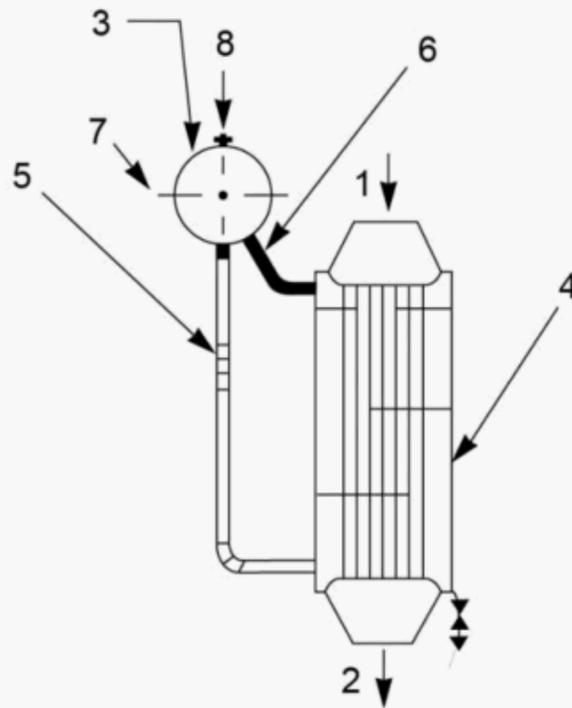
E.5.4 In any of these previous cases (Figure E.1, Figure E.2, or Figure E.3), it may be possible incorporate a steam separator device (e.g. mist elimination pad or chevron separator) into the design of the steam drum or kettle volume to minimize liquid entrainment while allowing the vapor to pass through and exit the vessel. An external steam separator is also possible if an internal separator is not practical.



Key

- | | |
|-------------------|------------------|
| 1. Hot fluid in | 7. BFW in |
| 2. Hot fluid out | 8. Steam out |
| 3. Steam drum | 9. Blowdown |
| 4. Heat exchanger | 10. Liquid level |
| 5. Downcomers | 11. Supports |
| 6. Risers | |

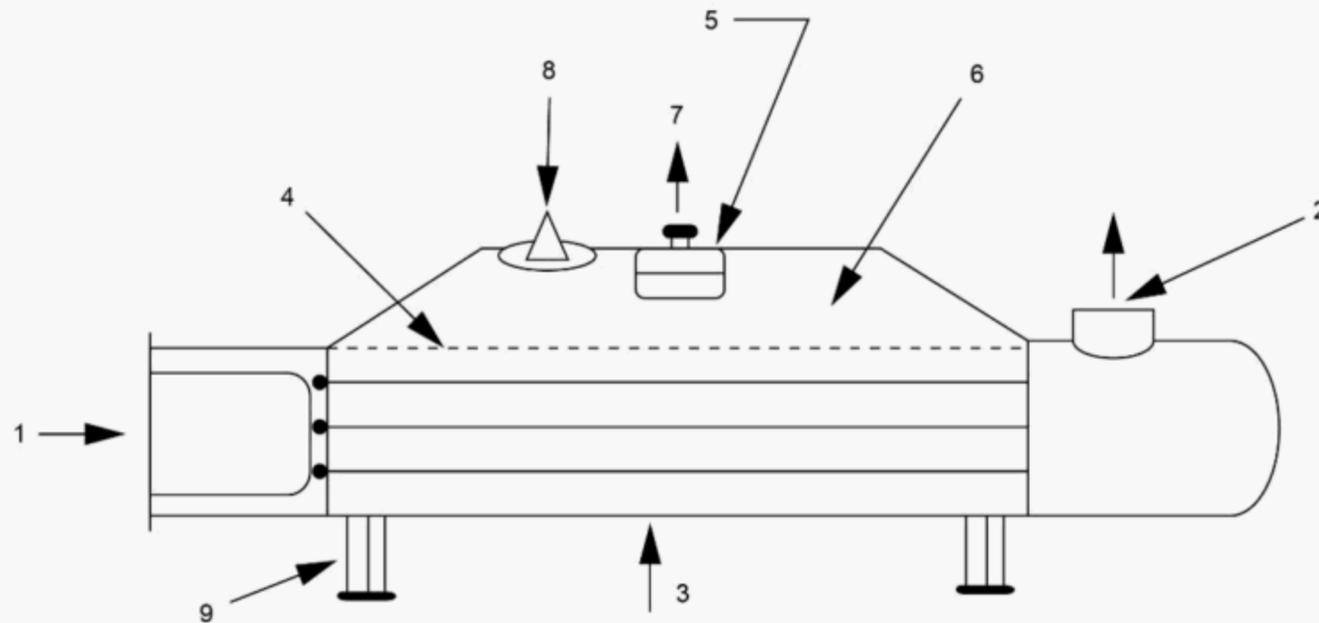
Figure E.1—Typical Horizontal Firetube with External Drum HRSG



Key

- | | |
|-------------------|--------------|
| 1. Hot fluid in | 6. Riser |
| 2. Hot fluid out | 7. BFW in |
| 3. Steam drum | 7. Steam out |
| 4. Heat exchanger | 9. Blowdown |
| 5. Downcomer | |

Figure E.2—Typical Vertical Firetube with External Drum HRSG



Key

- | | |
|--------------------|-----------------|
| 1. Hot fluid in | 6. Steam space |
| 2. Hot fluid out | 7. Steam out |
| 3. BFW in | 8. Safety valve |
| 4. Liquid level | 9. Supports |
| 5. Steam separator | |

Figure E.3—Typical Firetube Kettle Type HRSG

E.6 Firetube Mechanical Description

E.6.1 Inlet Channel

E.6.1.1 Inlet channels of high-temperature units are internally refractory lined to insulate the pressure components. Several refractory systems are available including dual and monolithic layers, cast and gunned, and either with or without internal liners. Various types of refractory anchoring systems are also used. Metallic needles dispersed into the castable may be considered as a means to further reinforce the refractory.

E.6.1.2 The selection of refractory materials and their application method must be compatible with the hot side service conditions. The design must account for concerns such as the following.

- Insulating capability, including effect of hydrogen content on the refractory thermal conductivity. The presence of hydrogen will increase the thermal conductivity of the refractory.
- Chemical interaction between the refractory and the process fluid.
- Gas dew point relative to cold face temperature.
- Erosion resistance against particulate bearing streams.
- Potential for coking under ferrules.
- Thermal shock.

E.6.1.3 Several inlet channel construction options exist. The gas connections may be inline axial or installed radially on a straight channel section. Inline is preferred for designs with low-pressure drop to ensure complete distribution of gas to all tubes. The channel should be designed to minimize flow turbulence and erosion of the refractory liner if present. Access into the channel compartment is generally through a manway in large diameter units, or through a full access cover in small units.

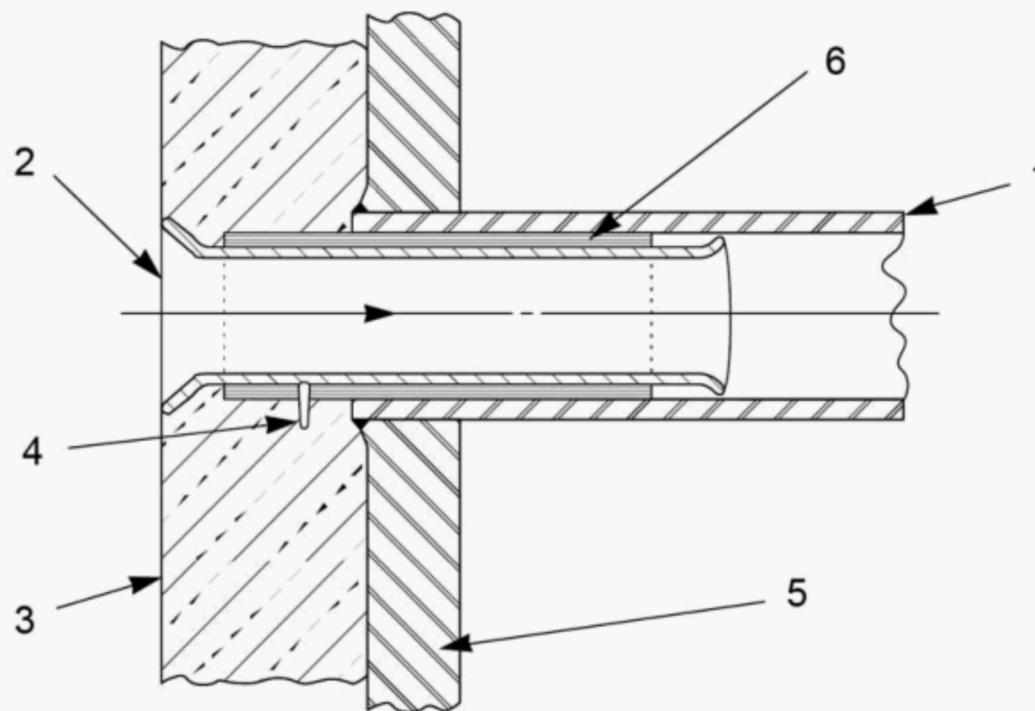
E.6.2 Tubesheets

E.6.2.1 The single most distinguishing feature of high-temperature firetube HRSGs is the thin tubesheet construction. Conventional shell-and-tube exchangers operating at moderate temperatures incorporate tubesheets which traditionally were designed according to the requirements of TEMA. In 2004 ASME *BPVC Section VIII, Division 1, Part UHX* replaced the TEMA method for tubesheet design. Typical tubesheet thicknesses in such units range from 50 mm (2 in.) to 150 mm (6 in.) but can be more. Use of ASME *BPVC Section VIII, Division 1, Part UHX* tubesheets in high-temperature service firetube WHRUs is not recommended because the tubesheet metal temperature gradient would be excessive and high stresses would result. Elevated temperatures of tubesheets in some process environments could lead to increased corrosion rates.

E.6.2.2 The thin tubesheet design is based on the use of the tubes as stays to provide the necessary support for the tubesheets. Tubesheet thicknesses typically range from 16 mm ($\frac{5}{8}$ in.) to 38 mm ($1\frac{1}{2}$ in.). Flat portions of the tubesheets without tubes must be supported by supplementary stays. The HRSG may have to be constructed to ASME *BPVC Section I*, or ASME *BPVC Section VIII, Division 2*, to permit use of a thin tubesheet.

E.6.2.3 Sufficient cooling of the tubesheet depends on efficient heat transfer at the tubesheet backface by shell side vaporization of water and by high local circulation rate for non-vaporizing applications. This offsets the heat input from the gas through the front face and, more importantly, the area created by all the tube hole perforations. The steady state tubesheet temperature is dependent on the tube pitch to diameter ratio and the tubesheet thickness. In vertical units, design provisions should be included to avoid steam blanketing at the hot tubesheet.

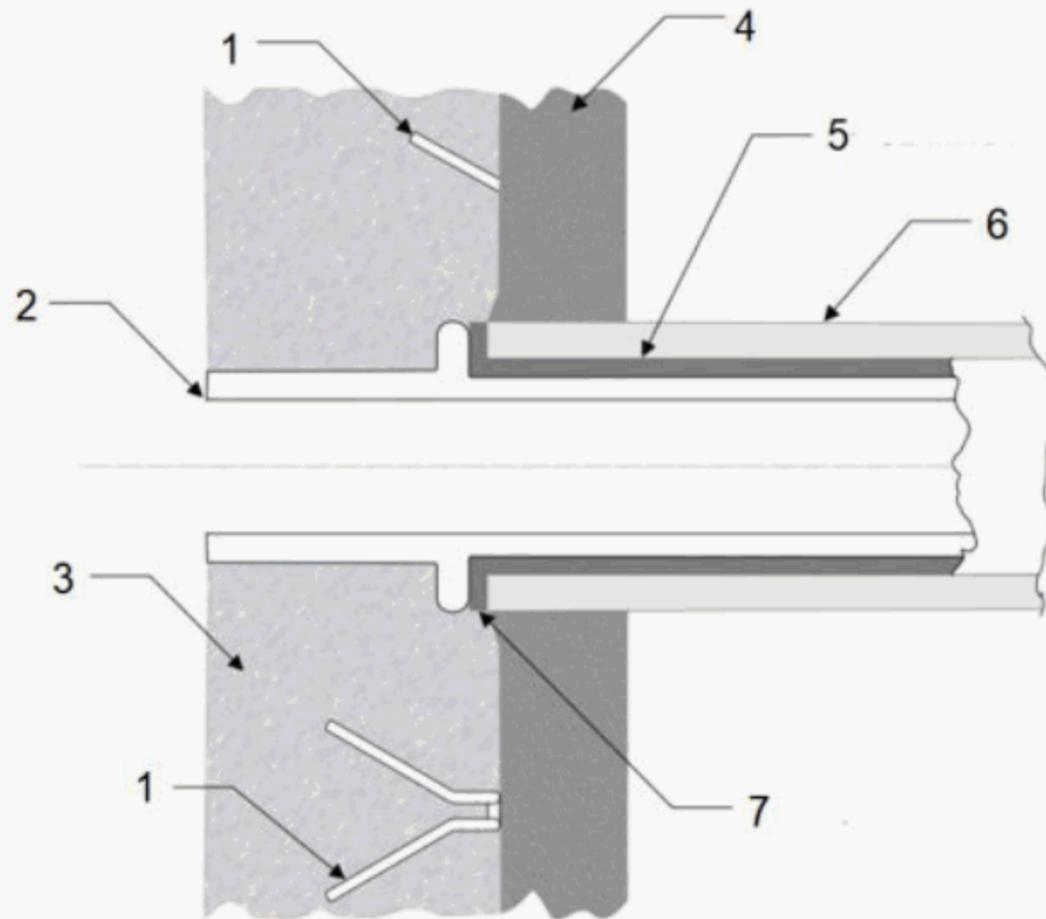
E.6.2.4 Tubesheet temperature can be further minimized by limiting heat flow to the tubesheet with the use of insulated ferrules inserted in each tube inlet. The ferrules typically project 75 mm (3 in.) to 100 mm (4 in.) from the tubesheet face. The space between the ferrules is packed with refractory, which secures the ferrules and insulates the tubesheet face. Ferrules are either a high-temperature resistant metallic or ceramic material. Figures E.4, E.5, E.6, and E.7 show typical details of metallic and ceramic ferrules. Other configurations may be used.



Key

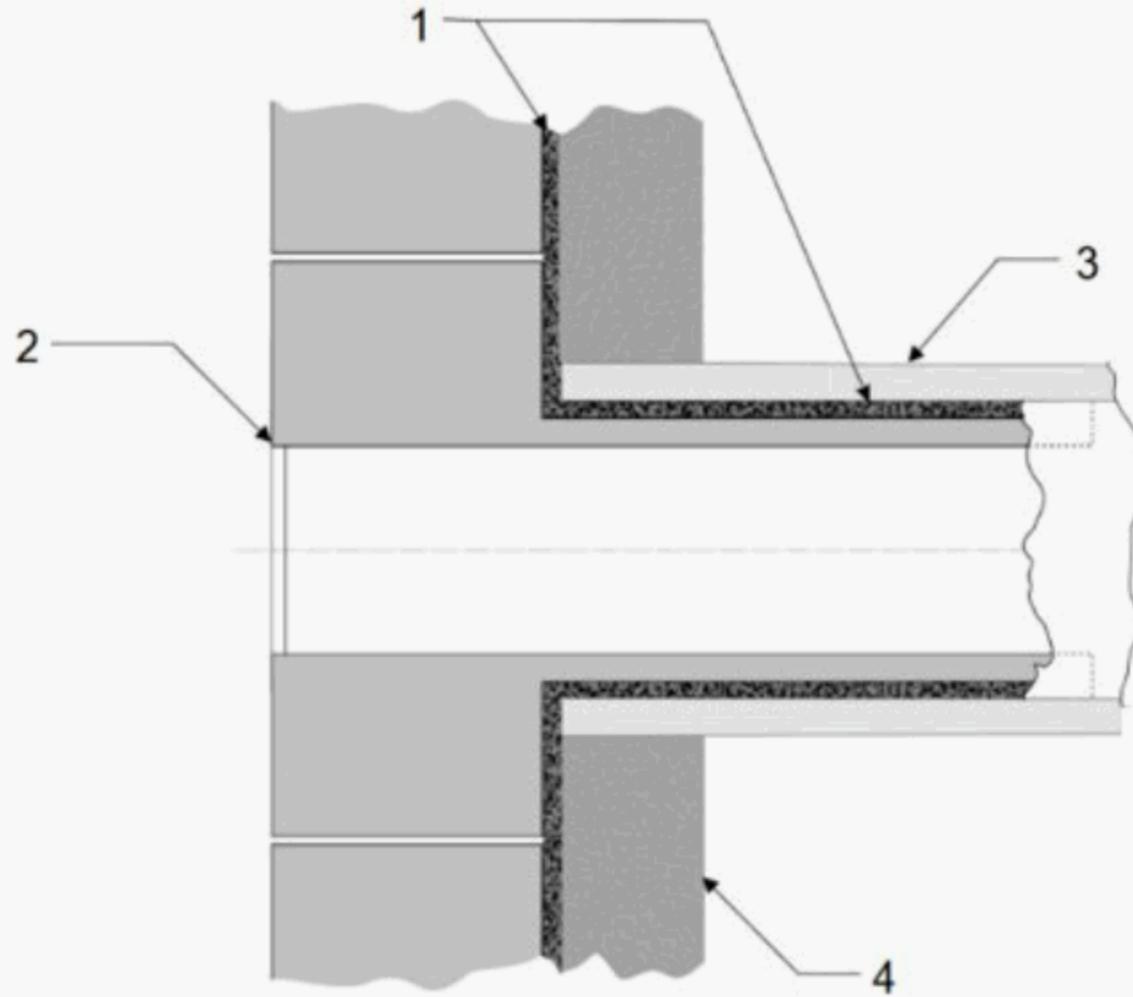
- | | |
|------------------|------------------------|
| 1. Tube | 4. Locating pin |
| 2. Metal ferrule | 5. Tubesheet |
| 3. Refractory | 6. Insulating material |

Figure E.4—Typical Insulated Metal Ferrule

**Key**

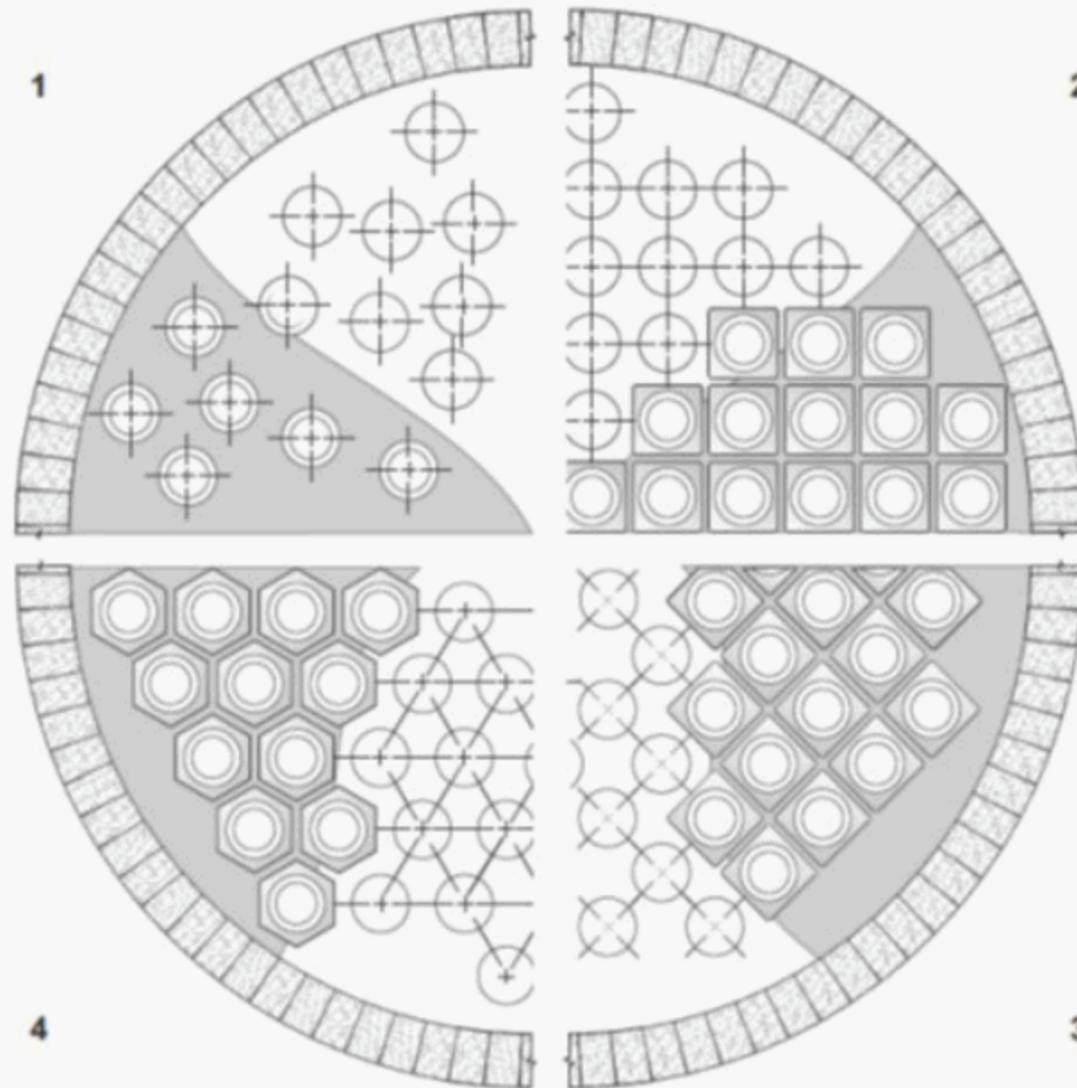
- | | |
|---------------|--|
| 1. Tube | 4. Tubesheet |
| 2. Ferrule | 5. Ceramic fiber paper insulating material |
| 3. Refractory | |

Figure E.5—Typical Straight Ceramic Ferrule

**Key**

- | | |
|-----------------------------|--------------|
| 1. Ceramic fiber insulation | 3. Tube |
| 2. Headed ferrule | 4. Tubesheet |

Figure E.6—Typical Headed Ceramic Ferrule



Key

- | | |
|--|---------------------------------|
| 1. Irregular Pitch Arrangement (unusual) | 3. Rotated Square Arrangement |
| 2. Inline Square Arrangement | 4. Triangular Pitch Arrangement |

Figure E.7—Tube Arrangements with Straight, Hex-head, and Square-head Ferrules

E.6.2.5 All types and styles of ferrules are individually wrapped with an insulating ceramic fiber paper sleeve for a lightly snug fit in the tube bore. Over compression of the insulation will reduce its effectiveness.

NOTE Each ferrule is typically installed with the ceramic fiber paper in place. The ceramic fiber paper provided with the ferrules is often mistaken for packing material and discarded. Even though the ceramic fiber paper is thin, it will also provide some thermal insulating value. Ferrules (metallic or ceramic) will not provide much thermal insulating value.

E.6.2.6 If it is difficult to insert the wrapped stem into the tube and the leading edge of the ceramic fiber paper wrap is damaged by bunching and tearing as shown in Figure E.8, then the ferrule fit is too tight, and the ferrule must be redesigned by the ferrule supplier. The insulating fiber paper length or thickness should not be reduced from that supplied to allow for ease of installation since this would provide insufficient thermal protection for the tubes and tubesheet.

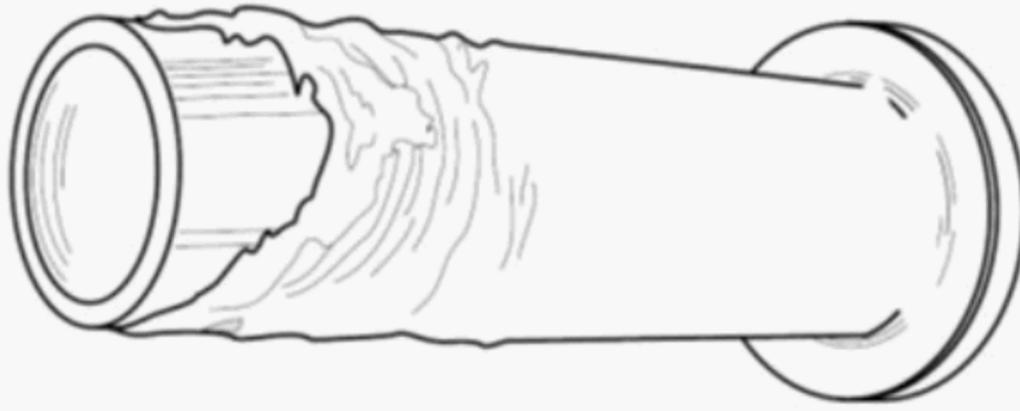


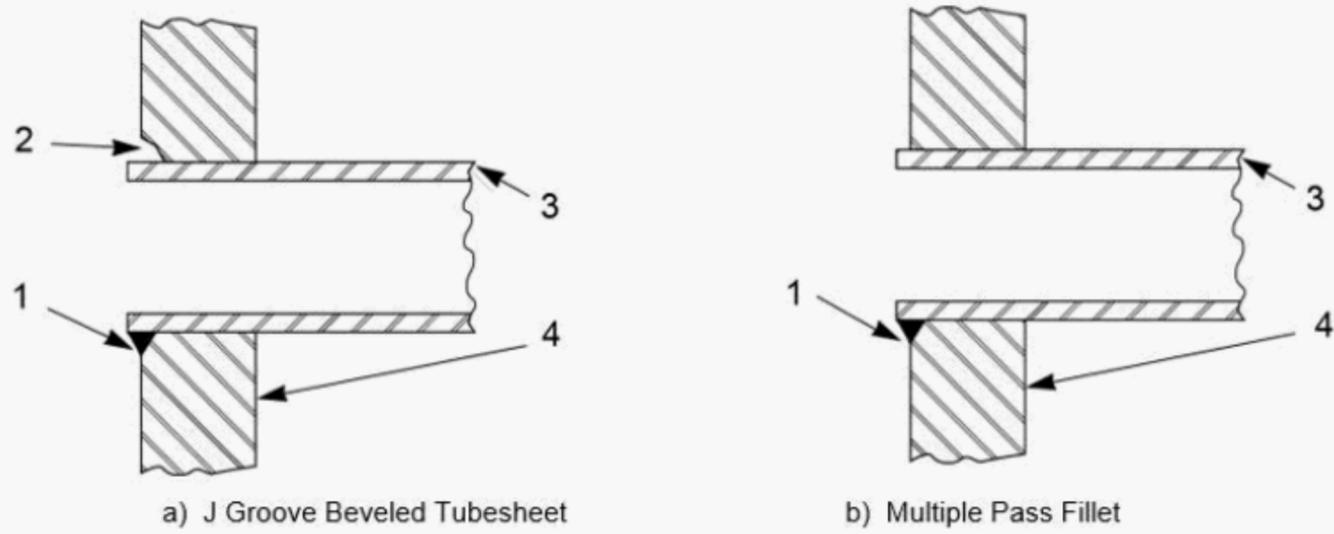
Figure E.8—Typical Damage when Ferrule Fit is Too Tight

E.6.3 Tube-to-Tubesheet Joints

E.6.3.1 The tube-to-tubesheet joints must provide a positive seal between the process fluid and the HTF (tube side and shell side) under all operating conditions at their resulting pressure and thermal loads. The joints must also withstand transient and cyclic conditions. The tube-hole tolerance should be as per TEMA R, special close fit.

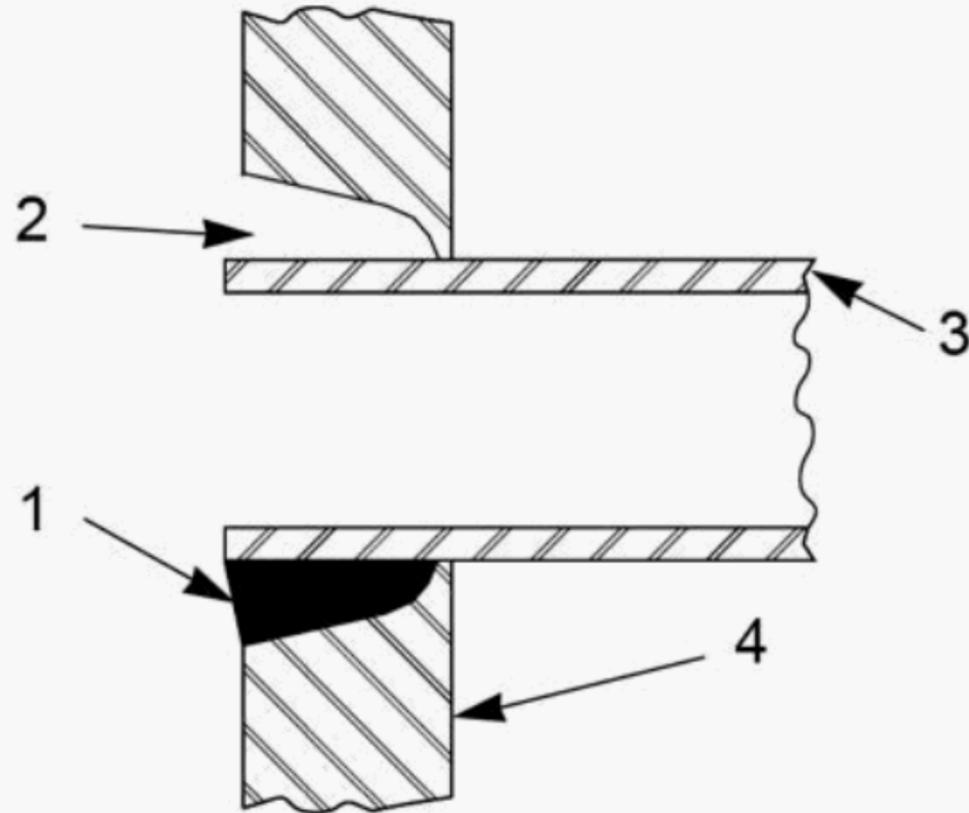
E.6.3.2 Tube-to-tubesheet joints in severe service applications are typically strength welded using one of the following configurations:

- a) Conventional strength weld, Front (tube side) face weld—The tubesheet may be J-groove beveled (Figure E.9.a) or just a simple angle beveled fillet (Figure E.9.b). The tube usually projects from the flat face. Additionally, each tube is hydraulically- or roller-expanded through the thickness of the tubesheet except near the weld and the back face of the tubesheet. Such joints may be used in elevated gas temperature applications. For generating steam, this is applicable for steam pressures to approximately 6900 kPa(g) (1000 psig).
- b) Full-depth strength weld—A deep J-groove with minimum thickness backside land is welded out with multiple passes as per Figure E.10. The land is consumed and fused so that the tube and tubesheet become integral through the full tubesheet thickness. Full-depth welded joints are often specified for high-temperature gases including those generating steam at pressures above 6900 kPa(g) (1000 psig).
- c) Back (shell side) face weld, Internal bore weld—This shell side type of joint is often called an internal bore weld, in that the welding is performed by reaching through the tubesheet tube hole (see Figure E.11). It has been applied to a wide range of firetube HRSG operating conditions, including high-pressure steam systems. A particular characteristic of this joint is that its integrity can be verified by radiographic examination. A mockup test is suggested for this type of weld to facilitate macro-examination and to confirm complete fusion has been achieved, with the proposed configuration and weld procedure. A tensile pull test may also be considered.

**Key**

1. Weld
2. Weld groove

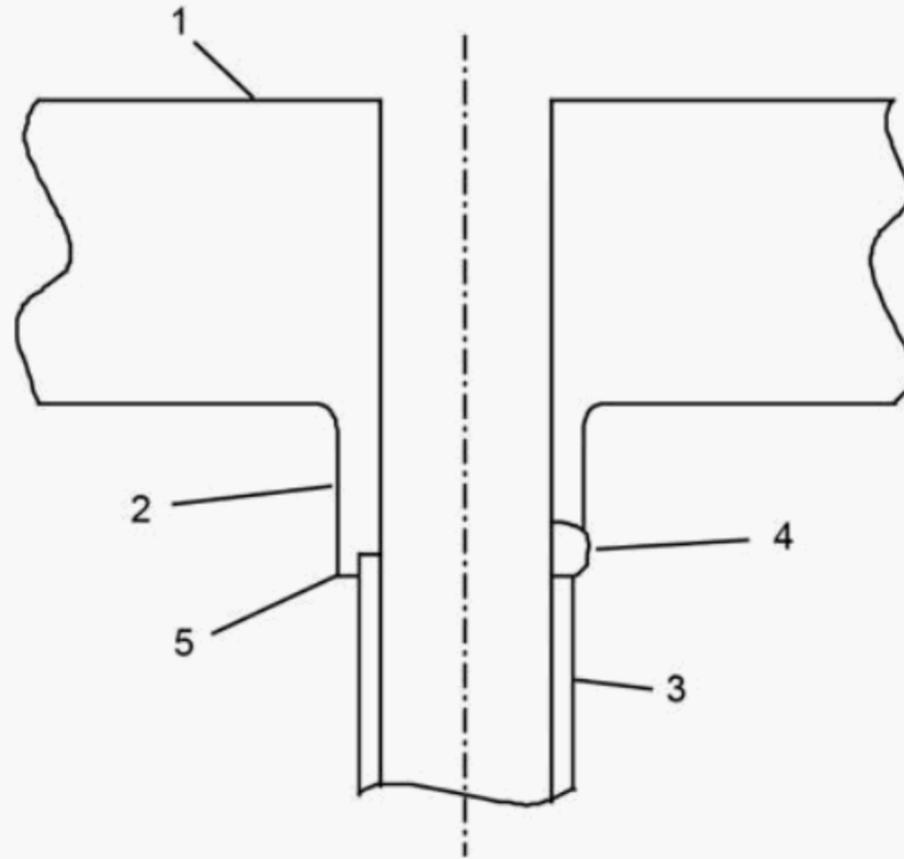
3. Tube
4. Tubesheet

Figure E.9—Conventional Strength Welds**Key**

1. Weld
2. Weld groove

3. Tube
4. Tubesheet

Figure E.10—Full Depth Strength Weld



Key

- | | |
|-------------------------|---------------------------|
| 1. Tubesheet | 4. Full penetration weld |
| 2. Butt weld connection | 5. Lip for tube alignment |
| 3. Tube | |

Figure E.11—Back Face (Shell Side) Weld

E.6.4 Thin Tubesheet with Peripheral Knuckle Attachment

E.6.4.1 A thin tubesheet is generally attached to the shell with a peripheral knuckle between the flat (tubed) portion and the point of attachment to the outer shell. See Figure E.12 for typical configurations. The knuckle provides this critical joint with the necessary flexibility to absorb the axial differential movement between tubes and shell caused by operating temperatures and pressures. Proper design of the knuckle is essential for reliable operation of a firetube boiler. All operating and upset cases considered in the design of the tubesheet should be documented; for example, using ASME *Section VIII, Division 1*, Form U-5.

E.6.4.2 The most severe cases are those involving elevated temperature gases with high heat transfer rates and with high steam side pressure. Such conditions impose considerable loads on the knuckles. An example of a severe service application would be reformer effluent in a hydrogen plant used to produce 10,400 kPa(g) (1500 psig) steam. Examples of less severe services include fluid catalytic cracking flue gas and sulfur recovery plant process gas where condensers generate steam at 4140 kPa(g) (600 psig) and below.

E.6.4.3 Knuckle design selection depends on factors that impact the extent of tube versus shell axial movement, such as the following:

- a) Differential thermal growth—Different mean metal operating temperatures and metallurgical differences create the differential thermal growth between the tubes and the shell. Operating temperatures and materials of construction are key factors affecting thermal growth. (See E.9.2.3.)
- b) Other factors creating axial movement between the tubes and shell: steam pressure, process gas pressure, and vertical vs. horizontal orientation.

E.6.4.4 Joints shown in Figure E.12.a) and Figure E.12.b) are used for services that only have minimal stress on the attachment joint; due to the fillet weld attachment and accompanying crevice. Figures E.12.c) through E.12.f) all have a butt-welded attachment to the shell. The flanged construction of Figure E.12.f) permits channel removal. Figure E.12.g) is used for high-pressure steam service and Figure E.12.h) is well suited for vertically installed units.

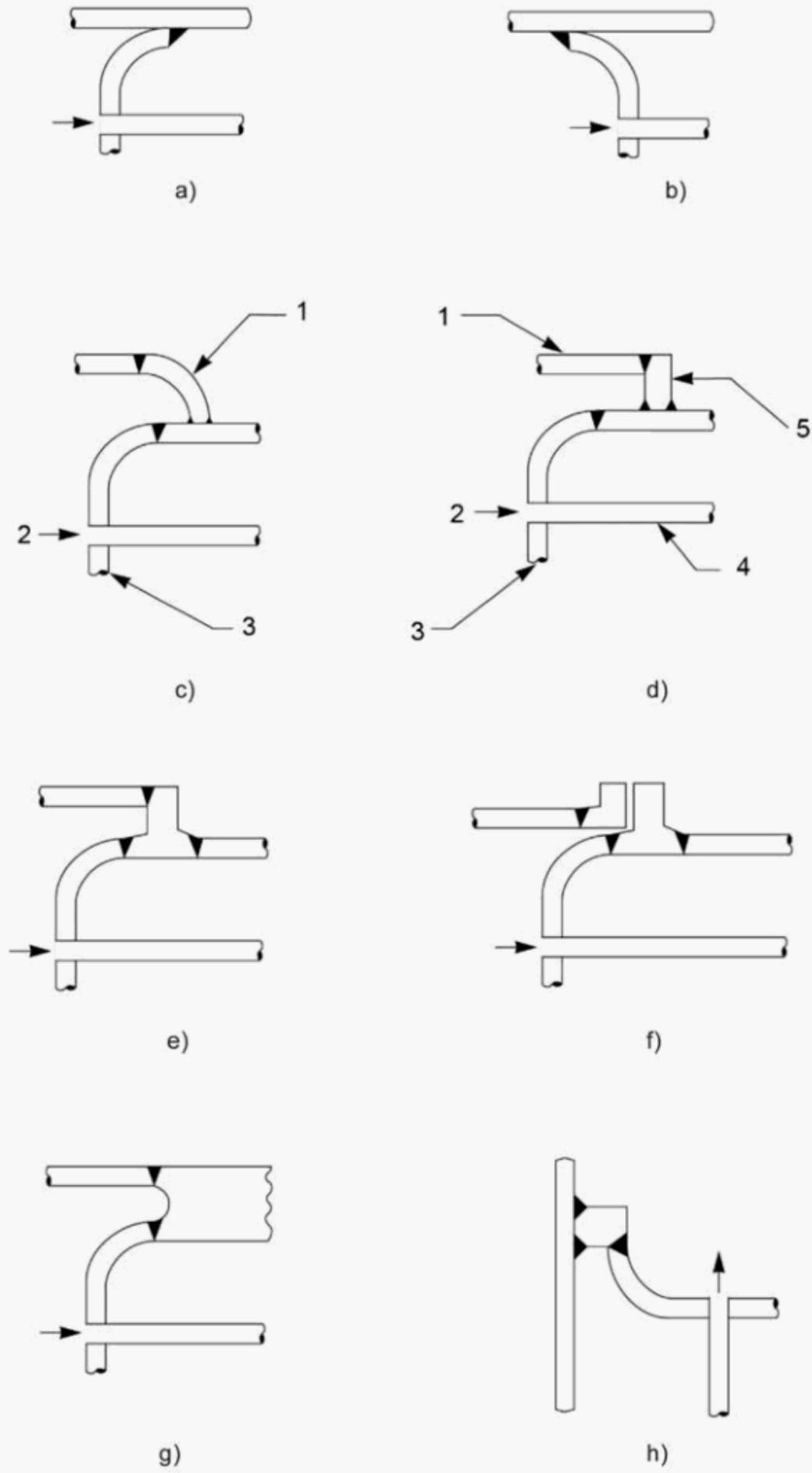
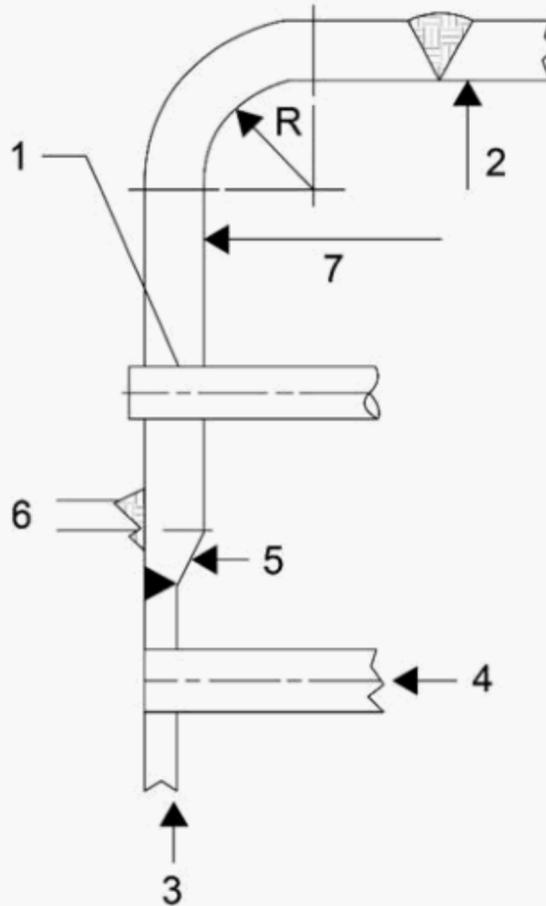


Figure E.12—Typical Channel to Tubesheet-shell Interconnections

**Key**

- | | |
|--------------|----------------------------------|
| 1. Stay rod | 5. Shell to tubesheet transition |
| 2. Shell | 6. Channel |
| 3. Tubesheet | 7. Shell head |
| 4. Tube | |

Figure E.13—Typical Knuckles for Full Diameter Kettles**E.6.5 Tubesheet for a Full Diameter Kettle Configuration**

A full diameter kettle is one which the enlarged portion of the kettle has the same length as the length of the tubes, i.e. no conical reducing section within the shell. Figure E.13 shows a typical knuckle configuration. This firetube design utilizes a stiffened thin tubesheet. Rather than relieving the tube axial loads with flexible knuckles, the loads are transmitted directly to the shell through a stiffening system of stay rods which back up the thin tubesheet. This design may permit the use of longer tubes.

NOTE The differential movement absorbed by the knuckles of conventional firetube HRSG tubesheets is proportional to the tube length. A maximum length limit exists, beyond which the knuckles would be incapable of accepting the imposed loads within stress limits of the material.

E.6.6 Dual-compartment Firetube HRSGs

E.6.6.1 The tube length limitation described in E.6.5 is of significant concern primarily with high-temperature, high-flux, and high-steam pressure equipment. For such cases, the option exists to use two compartments within one shell as shown in Figure E.14. Alternately they can be in separate shells in series as shown within Figure E.15.

E.6.6.2 The two-compartment arrangement may be served by a common steam drum. Advantages of this configuration include the following.

- a) Reduces differential growth between shell and the tubes within each compartment.

- b) Permits optimization of heat transfer surface through utilization of different tube diameters and lengths in each compartment, thereby reducing the total surface required.
- c) Permits locating the internal bypass system, when used, in the second compartment, thereby subjecting the control components to less severe temperature conditions. Internal bypass systems are not always used. See E.6.9 for information on bypass systems.

E.6.6.3 A variation on the dual-compartment design has the second compartment in series but positioned in parallel to the first. They share the same steam drum. This design reduces the plot length of the unit but requires that pipework be provided between the interconnecting channels of the two compartments and a wider plot be available. The steam drum may be over the hot portion or between the two passes. This configuration is shown in Figure E.15.

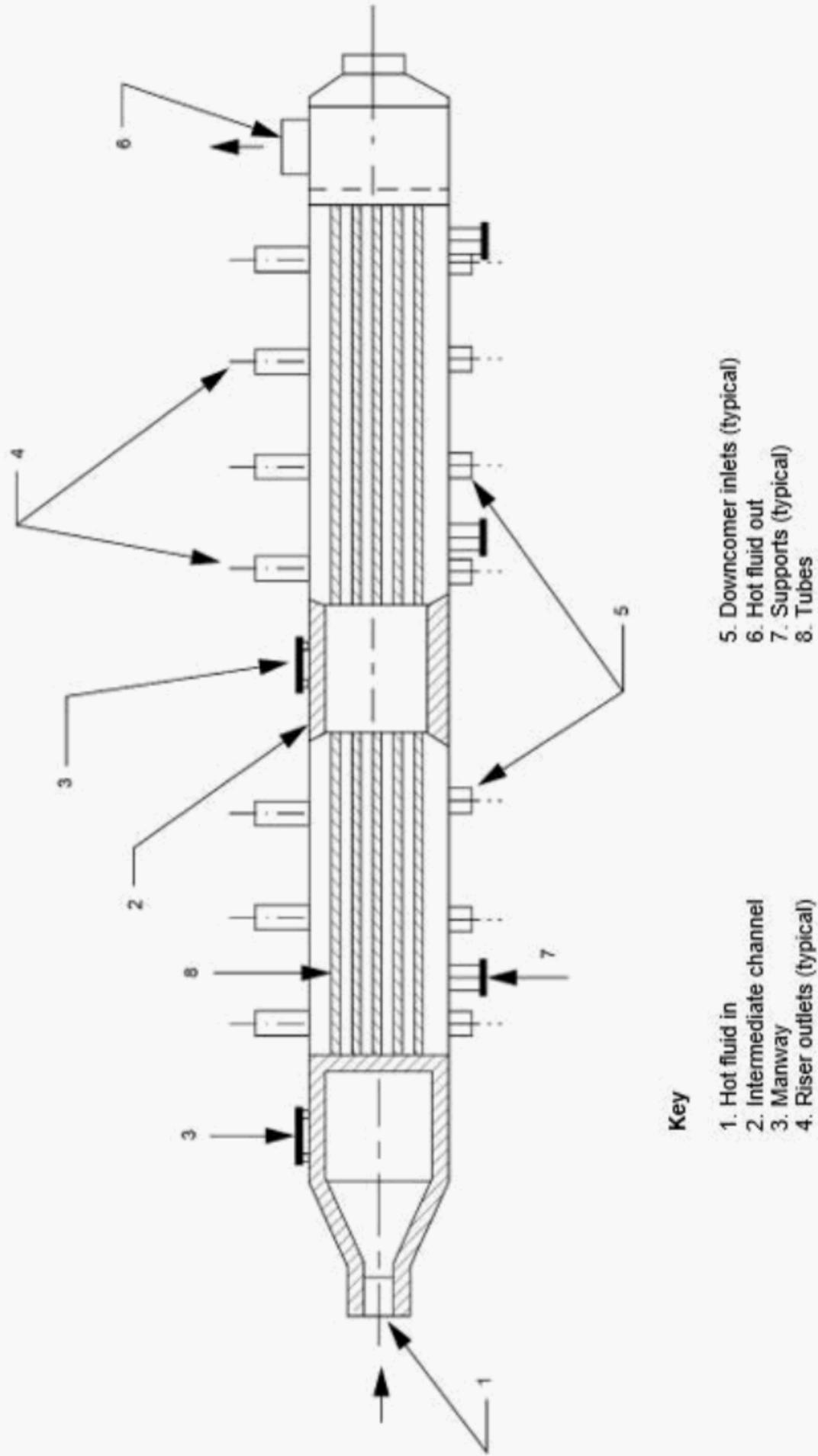
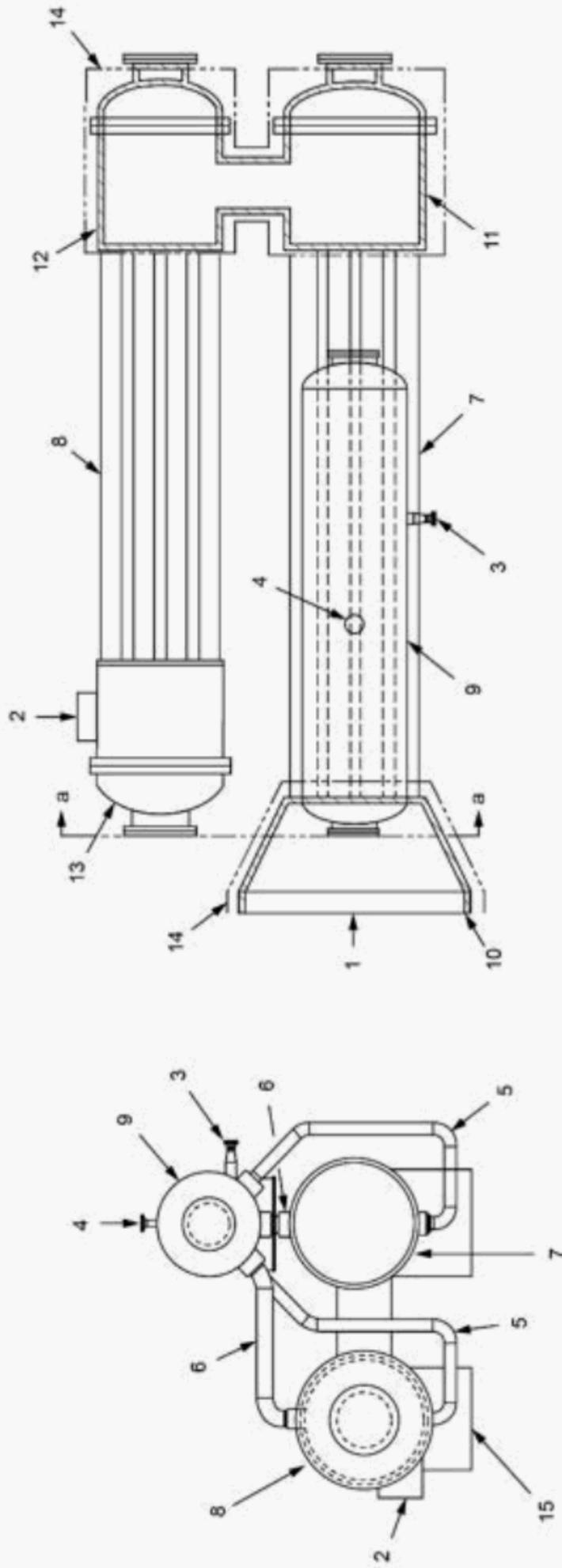


Figure E.14—Typical Dual-compartment Firetube HRSG



Section View (a-a)

Plan View

Key

- 1. Hot gas inlet
- 2. Hot gas outlet
- 3. BFW inlet
- 4. Steam outlet
- 5. Downcomer
- 6. Riser

- 7. First pass
- 8. Second pass
- 9. Steam droom
- 10. Refractory lined transition
- 11. Refractory lined 1st pass outlet inlet channel
- 12. Refractory lined 2nd pass inlet channel

- 13. Second pass outlet channel
- 14. 360° thermal shroud
- 15. Structural support (typical)

Figure E.15—Typical Two-pass, Separate Shells, Firetube HRSG

E.6.7 Tubes

E.6.7.1 Typical tube diameters in high-temperature firetube HRSGs range from 32 mm (1.25 in.) to 100 mm (4 in.). Use of relatively large tubes permits the following.

- a) Low-pressure drop application typical of low-pressure process gas streams such as tail gas of sulfur recovery plants.
- b) Thermal design at lower heat fluxes.
- c) Installation of tube inlet ferrules without over-restricting the flow area available at each tube entrance.
- d) Limits the potential for plugging of tubes in services prone to fouling.

E.6.7.2 The minimum tube wall thickness is governed by applicable code rules. Except for cases involving very high process gas pressures, the steam pressure, which acts externally generally controls the minimum tube thickness. A corrosion allowance should also be considered in selection of tube wall thickness.

E.6.7.3 Tube arrangement and spacing considerations include the following.

- a) Tubes are normally arranged on a triangular pattern to provide the smallest shell diameter, although square layouts may also be used.
- b) The selection of tube pitch should address the following concerns.
 - i) The maximum allowed heat flux is a function of the tube pitch to diameter ratio. Decreasing the pitch to diameter ratio reduces the allowable design flux.
 - ii) The tubesheet metal temperature is dependent on the tube pitch. Decreasing the pitch increases the metal temperature.
 - iii) A minimum tubesheet ligament width between adjacent tubes is required for welded tube ends to physically accommodate the tubesheet J-groove weld preparations. This is particularly significant for full-depth welded joints.

E.6.7.4 Most high-temperature process firetube designs are of single tube pass construction. However, multiple pass tubes may be considered for processes involving near atmospheric pressure gases used to generate low-pressure steam. The low-heat transfer coefficients characteristically associated with such gases result in tube metal temperatures which very closely approach the steam saturation temperature. Therefore, the metal temperature difference and differential thermal growth of tubes of different passes are minimal. Hot pass tubes are typically larger diameter than subsequent passes to optimize heat transfer within pressure drop constraints. Special consideration should be applied to the design of the inlet channel and tubesheet due to the different tube diameters and metal temperatures inherent in this configuration. Figure E.16 illustrates a two tube pass, high-temperature firetube steam generator.

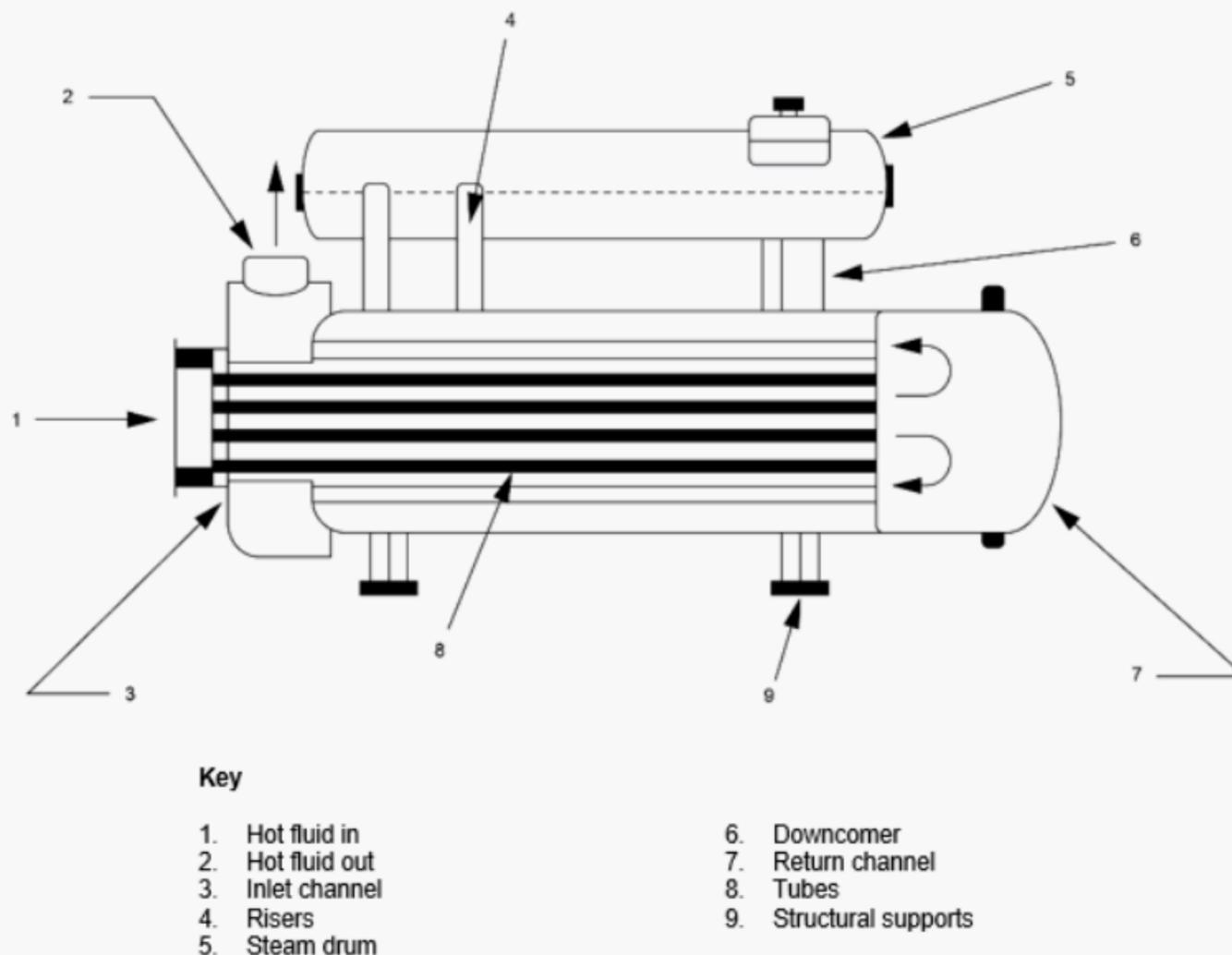


Figure E.16—Typical Two Tube Pass Firetube HRSG

E.6.8 Baffles and Tube Supports

E.6.8.1 In vertical firetube steam generators, it is important to select a type of baffle that does not restrict the flow of water. If the flow of water is blocked, the underside of the baffle could become blanketed with steam and cause the tube surfaces to dry out. This could lead to overheating or corrosion of the tubes. Rod baffles, egg-crate type baffles, or other suitably designed conventional baffles, may minimize these issues.

E.6.8.2 In horizontal firetube steam generators, tube support design and placement should not inhibit the proper water flow to all parts of the tube bundle, to avoid steam blanketing of the tubes and cause the tube surfaces to dry out.

E.6.9 Gas Bypass Systems

E.6.9.1 Gas bypass systems for process side outlet temperature control may be external or internal. Internal bypasses are commonly used because they take advantage of cooling the bypass pipe with water in the shell. The bypass pipe may be internally insulated to assure that the metal temperature is maintained close to the water temperature. In high-pressure steam applications, the pipe may be attached to a transition knuckle in each tubesheet to absorb axial loads. The pipe is located in the center of the tube layout to provide for axisymmetric distribution of loads.

E.6.9.2 An automatically-controlled valve is furnished at the outlet end of the gas bypass pipe to control the bypass flow rate. To reduce the size of the pipe and valve and to increase the flow control range, a plate with adjustable dampers may be installed in the outlet channel. By closing the dampers, the additional pressure-drop imparted to the main gas stream encourages flow through the bypass. The outlet channel should be refractory lined or provided with internals to preclude the possibility of impingement of hot bypass gas on the channel wall. A typical internal bypass system is shown in Figure E.17. Other systems are available.

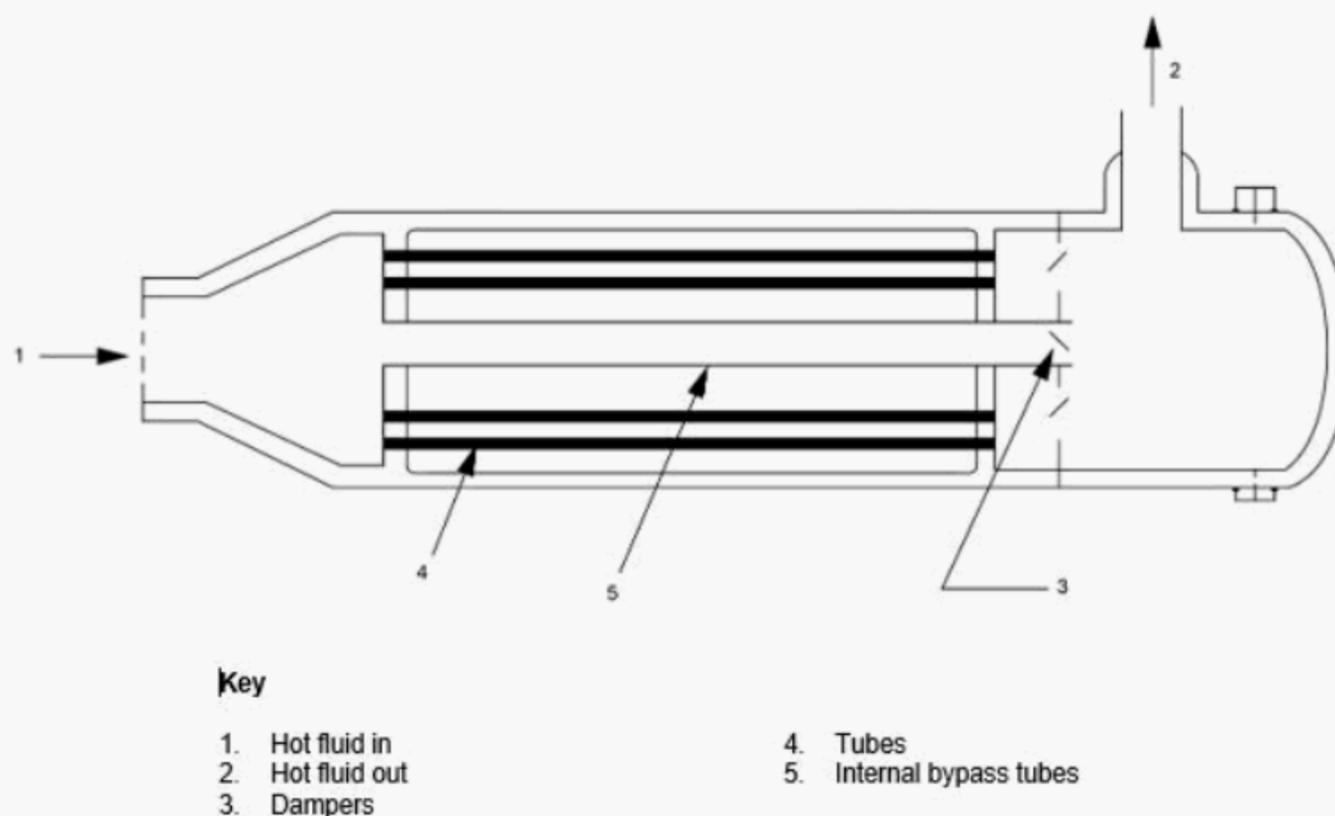


Figure E.17—Typical Internal Bypass System with Valve and Damper

E.6.10 Risers and Downcomers

E.6.10.1 Adequate quantity and size, and proper location of risers and downcomers are essential for reliable operation of high-temperature, high-flux firetube HRSGs. Setting the steam drum elevation, sizing the interconnecting circulation piping, and positioning the connections are an integral part of the design.

E.6.10.2 Riser and downcomer design and connection positioning depend on the orientation. Horizontal firetube designs are usually furnished with multiple risers and downcomers. Connections are positioned to serve zones of equal steam generating capacity. For single pass configurations, the connections tend to be more closely spaced at the hot end, due to the high temperature differential and high heat transfer rates at this location. This is where a significant portion of the steam is generated. A high circulation ratio is desired in this region to avoid “vapor locking” due to unstable two-phase flow. At least one riser and downcomer pair should be located as close as possible to the hot tubesheet, but the actual number and size of risers and downcomers should be selected in conjunction with the available pressure drop to give the correct circulation flow.

E.6.10.3 The system should be designed for the design flow rate and the specified turndown flow, normally 30 %.

E.6.10.4 Vertical units have one or more downcomer connections located at the bottom of the steam generator shell. Of greater significance, however, is the construction at the top which must ensure ample and continuous wetting of the entire shell side face of the tubesheet. The following construction options may be considered to help avoid vapor blanketing beneath the upper tubesheet.

- Multiple riser connections installed around the full circumference as high on the shell as possible.
- Reverse knuckle tubesheets to permit further elevation of the riser connections relative to the tubesheet (see Figure E.12.h).
- Special baffling under the tubesheet to direct water across the back face of the tubesheet.
- Special formed or machined upper tubesheet with a slight taper from the center upward to the periphery.
- Installation of the entire steam generator slightly canted from true vertical such that the tubesheet slopes slightly from horizontal upward toward the risers which are located on that side.

E.7 Kettle Steam Generators

E.7.1 General

Kettle steam generators are horizontally installed units with an enlarged shell side boiling compartment diameter relative to the tube bundle. The bundle penetrates through either a port opening in a conventional head, or the small end of an eccentric conical transition, the latter being more common. Liquid disengagement occurs within the enlarged shell diameter above the liquid level, and natural circulation patterns occur within the tube bundle. They are typically designed to the requirements of API Standard 660.

E.7.2 Tube Bundle Construction

E.7.2.1 Tube bundles may be removable or fixed. Removable bundles offer certain advantages. The bundle may be removed for inspection, cleaning, repair, or replacement. Also, removable bundles avoid the differential axial thermal expansion stress which occurs in fixed tubesheet designs.

E.7.2.2 Removable bundles may be of either U-tube or floating head construction. For fluids prone to fouling or erosive process fluids that may require mechanical cleaning or inspection, the floating-head type is preferred.

E.7.3 Tube Size, Arrangement, and Number of Passes

E.7.3.1 Typical tube diameters are 19.05 mm ($3/4$ in.) and 25.4 mm (1 in.), although larger sizes are considered for process fluids prone to high fouling or viscous process fluids, such as in sulfur condensers. Tubes are arranged on either a square or triangular pattern. The square arrangement is used for better circulation or if cleaning of the outside tube surface is anticipated, as could be the case for generating low-pressure steam from poor quality boiler water. In such cases 6 mm ($1/4$ in.) minimum width cleaning lanes are maintained between tubes. Otherwise, a pitch to diameter ratio of 1.25 is normally used, unless heat flux considerations requires a larger spacing.

E.7.3.2 Multiple tube passes may be used for all bundle types described in E.7.2, except for cases with extremely long hot fluid cooling ranges which may experience severe thermal stress. Single pass tubes are normally limited to fixed tubesheet construction.

E.7.4 Channel Construction

The selection depends primarily on the anticipated frequency of opening the unit for inspection or cleaning. If frequent access is required, a channel with bolted cover plate is desirable. Channels may be according to any of the TEMA designated types.

E.7.5 Shell Construction for Disengagement

E.7.5.1 Disengagement of liquid is achieved in the steam space above the liquid level. The effectiveness of this volume is a function of the free height available, vapor velocity, droplet size, and water quality. A typical minimum height is 500 mm (20 in.) in steam generating equipment. Units which produce very low-pressure steam or operate at relatively high flux tend to need additional height. The water level is normally maintained at between 50 mm (2 in.) and 100 mm (4 in.) above the top of the uppermost tube row to ensure the tubes are submerged. Simple dry pipe devices are sometimes used to enhance separation.

E.7.5.2 Proper water distribution over the entire tube bundle should be provided. Internal perforated distributors or multiple inlet nozzles may be used. For high-temperature services, one boiler feedwater inlet nozzle is typically located adjacent to the hot tubesheet. Tube support design should not inhibit water distribution.

E.7.5.3 The instrumentation and controls of a kettle steam generator are similar to that of a steam drum in a thermosiphon boiler system.

E.7.5.4 A properly sized kettle shell produces steam of adequate quality and purity for most process and heating applications. Higher purity steam may be achieved by the installation of separators in the vapor space above the liquid level, within a dome welded to the top of the kettle, or in the exit vapor line. Types of separators include dry pipes, demister pads and vane type separators.

E.7.5.5 Kettles with a knuckle type tubesheet may use the configuration shown in Figure E.13.

E.8 Other Types of Firetube HRSGs

E.8.1 There are many other types of firetube HRSGs designed for a variety of services. They may be further classified as follows.

- a) Proprietary designs developed for specific process applications.
- b) Steam Generators designed with thick (TEMA type) tubesheets and external drums. The steam generators may be installed in the horizontal or vertical position.

E.9 Other Mechanical Design Considerations

E.9.1 Channel Construction

Firetube HRSG channels are typically designed in accordance with the pressure design code when required by the jurisdiction.

E.9.2 Corrosion Resistance

E.9.2.1 Materials selected for use in firetube units must be compatible with the process fluid, the boiler water and steam with which they will come into contact. The materials must also exhibit mechanical properties consistent with the design requirements of the equipment. Each process fluid from which heat is being recovered has its own composition and may, therefore, have its own unique requirements for construction materials. An important factor in materials selection is often resistance to hydrogen attack, because many high-temperature process gas streams have significant hydrogen content. The specification of materials must also account for the possibility of gas cooling below its dew point, and the corrosive acids which may be formed. Cold metal surfaces can cause local condensation, even though the bulk gas may be above the dew point.

E.9.2.2 Pressure components wetted by boiler water, including tubes and tubesheets, are normally fabricated from ferritic materials. Steam generator shells are generally carbon steel. Materials subject to stress corrosion cracking, such as austenitic stainless steels, are normally avoided and are prohibited in the evaporator by ASME *BPVC, Section I*.

E.9.2.3 The relative differential expansion between the shell and the tubes resulting from the operating temperature differences is of considerable significance to firetube HRSG design. Materials with similar coefficients of thermal expansion are beneficial. This is another reason for avoiding the use of austenitic tubing.

E.10 Operational Considerations

E.10.1 General

Safe and reliable operation of firetube HRSGs depends on the development and use of good operating procedures, specific to the process and HRSG design.

E.10.2 Process Side Operation

E.10.2.1 New refractory lining may require a special heating sequence on start-up to affect proper dry-out.

E.10.2.2 Firetube units must not be subjected to hot gas flow without the tube bundle fully covered by boiler water.

E.10.2.3 The rate of temperature change during transients should be controlled to minimize the potential for thermal shock.

E.10.2.4 All modes of operation should be evaluated during the design phase, particularly with regard to the ability of the steam generator components to withstand the primary and secondary stresses during cyclic operation.

E.10.2.5 All start-up shut-down conditions as well as all production levels (full rates, half rates or less) can have drastic impacts on the performance of these units. Very high heat flux (the hottest tube side gases in the tubes with water on the shell side at ambient temperature) can create large thermal differential stresses that must be managed.

E.10.2.6 For a kettle type HRSG, lowering the water level to facilitate steam separation can result in the top tubes of the bundle becoming uncovered with water resulting in them becoming much hotter than the remainder of the tube bundle. This may cause high thermal loads sufficient to cause tube failure.

E.10.2.7 At maximum rates, the steam exit velocity could cause water carry over and low steam quality. Consider the impact of poor-quality steam on downstream users. For example, poor steam quality can lead to steam turbine failure.

E.10.3 Steam Side Operation

E.10.3.1 Reliability of BFW Supply

Of primary importance to the successful operation of firetube HRSGs is the reliable supply of boiler water to the heat transfer surface. In the event of BFW supply failure, the control system must shut off the hot stream flow to the HRSG.

E.10.3.2 BFW Treatment

For example, BFW chemical treatment must be such that steam generator components are protected from water side corrosion. Improper treatment, or upsets, may cause premature failure. Water treatment specialists are normally consulted. American Boiler Manufacturers Association (ABMA Boiler 402) and ASME's Center for Research and Technology Development (CRTD 34) guidelines are commonly followed. Refer to API Recommended Practice 538 for additional information.

E.10.3.3 Continuous Blowdown

Blowdown must be used in conjunction with BFW treatment to assure boiler water impurities are maintained at or below their recommended maximum concentrations. Continuous surface blowdown is normally accomplished through a perforated collector pipe located just below the water-steam interface. Continuous blowdown from kettle HRSGs should be extracted primarily at the end opposite the feedwater inlet where impurities would be most concentrated.

E.10.3.4 Intermittent Blowdown

Intermittent blowdown acts to remove settled accumulations of boiler water solids. Connections are located at low points in the shell, particularly in the most stagnant regions. Blowdown valves are operated at prescribed intervals, depending on the effectiveness of boiler water treatment. Intermittent blowdown valves should be quick opening type and frequent short burst give better sludge removal than long infrequent opening.

Annex F (informative)

FCCU/RCCU Process Units Heat Recovery Systems

F.1 CO Boilers

Fluid catalytic cracking units (FCCU) and residual catalytic cracking units (RCCU) process heavy fractions of the barrel of crude oil such as heavy gas oils or residual oils and crack them utilizing fluidized catalyst. The hydrocarbon components are broken into small, more useful, hydrocarbon molecules. Some of the hydrocarbons are left on the surface of the catalyst as coke. The coked catalyst is transferred to a separate vessel where it is exposed to air. The oxygen in the air burns the coke off the catalyst. If there is insufficient oxygen present, some of the coke only burns to carbon monoxide instead of the complete combustion product of carbon dioxide. The reason for not fully combusting the coke is to limit the temperature in the combustion vessel to allow the use of lower alloy materials of construction. The clean catalyst is returned to the fluidized bed and the process is repeated. The CO rich flue gas is then sent to a CO boiler where additional oxygen from ambient air and possibly more fuel is added. Oxygen enrichment of the ambient air may also be used to lower the flue gas velocity through the boiler. The hot flue gases are used in the CO boiler to generate steam at a variety of pressure levels depending on the needs of the plant. Supplemental burners may be required to heat up the ambient air, maintain the required CO combustion temperature, or increase the steam production from the boiler. These systems typically include a steam drum, circulating pumps and desuperheaters besides the actual heat transfer surface(s).

NOTE Guidance on the design of this type of CO boiler can be found in API Recommended Practice 538.

F.2 Flue Gas Coolers

F.2.1 General

If there is sufficient oxygen present, the coke burns to the complete combustion product of carbon dioxide. The flue gas is then sent to a steam generator often referred to as a flue gas cooler. The hot flue gases are used in the steam generator to generate steam at a variety of pressure levels depending on the needs of the plant. There are two configurations of this type of steam generators. They can be either watertube or firetube types. These systems typically include a steam drum, circulating pumps (sometimes natural circulation is used) and desuperheaters besides the actual heat transfer surface(s).

F.2.2 Watertube Steam Generators

The hot flue gases pass over the outside of the tubes in the steam generator and the steam is generated on the inside of the tubes. Guidance on the design of this type of HRSG can be found in API Recommended Practice 538.

F.2.3 Firetube Steam Generators

F.2.3.1 The hot flue gases pass through the inside of the tubes in the steam generator and the steam is generated on the outside of the tubes. Refer to Annex E for information on firetube HRSG design information.

F.2.3.2 The flue gases entering this type of HRSG is typically at high temperatures on the order of 650 °C to 815 °C (1200 °F to 1500 °F). It also is likely to contain solid particulates which can be erosive. High amounts of sulfur can elevate the dew point of these gases. As the source of the flue gases is often elevated, it is common

that this type of HRSG is vertical. That takes advantage of using the piping and allows for gravity to help prevent buildup of particulates in the tubes.

F.2.3.3 The design of this type of HRSG is typically controlled by the low heat transfer coefficient in the tubes compared with the higher boiling coefficient of the water/steam mixture on the outside of the tubes. This often results in increasing the tube side velocity as high as possible without causing erosion of the tubes, tubesheet and their joint. Inlet velocities to the tubes may be on the order of 30 m/s (100 ft/s). Ferrules (see E.6.2) are often used to minimize erosion and very high heat flux rates at the top of the HRSG.

F.2.3.4 In some cases, a dual-compartment HRSG (see E.6.6) may be used where one compartment generates steam and the second superheats the steam.

F.3 Catalyst Coolers

F.3.1 In FCC/RCC units, it is also possible to reclaim waste heat using a specialized watertube design to directly remove heat from fluidized catalyst to generate steam. Mechanical reliability is achieved by locating the cooler in the dense phase of the regenerator where higher heat transfer coefficients are obtained in lower velocity catalyst and fluidization air areas. This minimizes the potential for erosion.

F.3.2 The catalyst cooler tube bundle is inserted into a refractory lined shell on the side or bottom of the regenerator. The catalyst flows on the outside of the tubes and the water/steam are in the inside of the tubes. Typically, upward pointing bayonet type tubes are used so that the water and steam enter and exit from the same end of the cooler. The boiler feedwater enters the inner tube and the isothermal steam and water mixture flows in the outer tube. The catalyst flow in the cooler is fluffed to create turbulence and enhance the heat transfer rate. At the same time the gentle aeration air rate is controlled to avoid erosion of the metallic components. The water flows via a pumped circulation system from the steam drum to the cooler and back to insure sufficient water is present to keep the tube surfaces properly wetted. The tubes need to be properly supported to avoid vibration due to the catalyst flow.

Annex G

(informative)

Process Heater Convection Sections

G.1 Process heaters can improve their energy efficiency by putting heat transfer surface into a convection section. The fluid used in the convection may be the same as used in the radiant or it may be different. It could be a different process fluid, hot oil, or steam (superheater, generation, feedwater heating), among others. These steam systems typically include a steam drum, circulating pumps and desuperheaters besides the actual heat transfer surface(s).

NOTE Guidance on the design of this type of convection section can be found in API Standard 560.

Annex H

(informative)

Gas Turbine Exhaust Heat Recovery Systems

H.1 Gas turbines are used in plants for generating steam, driving various types of rotating equipment (compressors, pumps), and generating electricity. The gas turbine generates a large quantity of flue gases at fairly high temperatures that is typically oxygen rich. The flue gases can be routed to a steam generator to recover heat. The steam generator may have a burner at the inlet of the steam generator to increase the temperature of the flue gas. This will increase the quantity of steam being generated or the temperature of the superheated steam. The steam generator may have multiple steam pressure levels. The flue gases could also be routed to a recovery heat exchanger. The heat can be picked up by hot oil, glycol systems or other systems.

H.2 These systems typically include a steam drum, circulating pumps, a duct burner, diverter valves, and desuperheaters besides the actual heat transfer surface(s). A duct burner is a special configuration burner to evenly distribute heat across the cross section of the ductwork at the inlet of the heat recovery system.

NOTE Guidance on the design of these types of HRSG and other utility systems heat recovery equipment can be found in API Standard 534.

Annex I (informative)

Thermal Oxidizers Heat Recovery Systems

I.1 General

Thermal oxidizers, also known as incinerators, are used to thermally oxidize (combust) all or only a portion of a gaseous, liquid, or solid feed. The general characteristics of all thermal oxidizers are fundamentally the same. This type of equipment is also covered in API Recommended Practice 538.

I.2 Target Constituent

The majority of thermal oxidizers are designed to destroy the targeted constituent(s) of the feed to a specified level reflecting the allowable residual concentration of the constituent(s) permitted to be released into the atmosphere. For example, if the targeted constituent was H₂S, the specified oxidation (destruction) efficiency might be 99 %, meaning that only 1 % of the initial concentration would be allowed to pass through the oxidizer to the atmosphere. Other common constituents that require destruction include such things as mercaptans, benzene, toluene, phenol and other sulfur compounds.

I.3 Destruction Efficiency

Since the destruction efficiency specified for the oxidizer must typically be guaranteed, the design, construction and operating controls of the oxidizer must assure that this guarantee will be met under all stated operating conditions. Thermal oxidizers are typically source tested for compliance with the permitted destruction efficiency and often subject to continuous monitoring for compliance. The oxidizer should also be designed to conform to regulatory flue gas emission requirements.

I.4 Design Characteristics

I.4.1 Thermal oxidizers come in various sizes and configurations. Those handling gaseous or liquid waste streams tend to be cylindrical in cross section and arranged for either vertical upflow or horizontal flow of the effluent gas.

I.4.2 Thermal oxidizers usually employ carbon steel outer shells that are internally insulated. Externally insulated, stainless steel or other high-temperature alloy outer shells are typically used in smaller units where the operating temperature is relatively low (under 800 °C [1500 °F]). Internal insulation may be firebrick, castable refractory or ceramic fiber. Insulation thickness is often much greater than that found in conventional fired heaters and boilers, especially when firebrick or high density castable refractory is used to address the erosive properties of the high velocity effluent gas.

I.4.3 One or more burners may be provided with the thermal oxidizer. Burner design can vary depending on the nature of the waste gas stream and on the specific burner technology provided by the oxidizer supplier. Separate burners may be used for the waste gas stream and for the supplemental fuel input, if required. Often the physical size of the oxidizer may dictate the use of multiple burners to ensure thorough mixing of the combustion constituents in a volume that provides the desired residence time. Continuous pilots are typically used in thermal oxidizers; however, sparger type ignition lances are sometimes provided.

I.4.4 Air is commonly used as the oxygen source for thermal oxidizers, although in some processes oxygen enriched air is employed. The air supply is often split into two or more entry locations on the oxidizer. The amount of air supplied is in excess of that stoichiometrically required for the oxidation of the waste gas constituents

targeted for oxidation and the supplemental fuel needed to maintain the required operating temperature in the effluent gas. The amount of excess air is typically greater than that employed in conventional fired heaters and boilers but not sufficient to create a residual oxygen level in the effluent gas that could support supplemental fuel burning without additional augmenting air. In certain oxidizers operating at very high temperatures, a dilution air stream is introduced after the desired residence time is met to lower the effluent temperature to a level where it can be used in a heat recovery unit without the need for very expensive heat exchanger and insulation materials.

I.4.5 The design of the stack (natural draft operation) and of any mechanical draft equipment (forced or induced draft operation) is based solely on the needs of the thermal oxidizer. The inclusion of any heat recovery downstream of the oxidizer will dictate the inclusion of a greater pressure differential for the mechanical draft equipment. When the effluent gas contains particulate matter, corrosive compounds or other toxic material, downstream effluent gas treatment may be required (e.g. baghouse). The combined resistance of a heat recovery unit and one or more post-combustion effluent gas treatment processes must be taken into consideration in the design of the mechanical draft equipment.

I.4.6 Waste heat recovery units typically use watertube configurations as described in Annex D. They may use any type of utility fluid, e.g. steam, water, hot oil.

I.5 Combustion Air and Waste Gas Preheat

I.5.1 Combustion air or waste gas preheat are commonly employed on thermal oxidizers to reduce the amount of supplemental fuel required for the design operating conditions of the oxidizer. Combustion air or waste gas preheaters may be of either the recuperative type or the regenerative type. The effect on flue gas emissions should be considered when adding a preheat system. A balanced-draft system, consisting of both forced and induced draft fans, is often required to overcome the pressure drop across both sides of the preheater.

I.5.2 Recuperative preheaters can be very simply constructed devices. Often, they consist of only a shroud constructed outside the oxidizer/stack shell creating an annulus through which the combustion air can pass before entering the oxidizer. Recuperative preheaters can often be combined with additional heat recovery units.

I.5.3 Regenerative preheaters consist of a rotating heat wheel exposed to both fluids. They are more complicated than recuperative preheaters both in design and in operation. These devices typically can be used to preheat both combustion air and the waste gas stream. Additional heat recovery units downstream of regenerative preheaters are not typically used due to the fluctuating temperature, potential air leakage, and other operating characteristics of the regenerative system.

I.6 Instrumentation and Control

Instrumentation and control of the thermal oxidizer is based on the targeted destruction of the specified compounds in the waste gas stream. The overall control system designed to ensure the performance to be guaranteed for the oxidizer must not be compromised by additional controls that may be supplied to manage any downstream heat recovery applied to the oxidizer. When the heat recovery process is managed independently of the oxidizer operation, either through supplemental heat input or flow management of the heat recovery medium, the operation of these controls must not disrupt the basic operation of the oxidizer.

I.7 Thermal Oxidizers

I.7.1 Thermal oxidizers require the following inputs to operate successfully:

- a) a feed stream, often referred to as the waste gas or waste stream;
- b) a source of oxygen sufficient to oxidize the targeted components;
- c) any supplemental fuel required;

- d) devices promoting the mixing of the waste stream with the source of oxygen and other products of combustion;
- e) a source of ignition for the waste stream or supplemental fuel input; and
- f) a containment vessel suitable to maintain the mixture at a specified temperature for a certain minimum period of time.

1.7.2 Common gaseous streams that are often routed to thermal oxidizers include:

- a) sulfur plant tail gas,
- b) acid gas removal unit flash gas, and
- c) acid gas enrichment unit off gases.

1.7.3 Thermal oxidizers may operate under either natural draft conditions or with some form of mechanical draft. Mechanical draft operation is by far the more prevalent. Natural draft operation severely limits the ability to recover heat from the hot effluent gases due to the limited ability of the available natural draft to overcome the pressure loss through a heat recovery unit. Mechanical draft is typically provided by a forced draft fan or blower. The fan or blower can usually be designed with sufficient discharge pressure to handle a downstream waste heat recovery unit. If necessary, an induced draft fan can be added downstream of the heat recovery unit to overcome the pressure losses of larger heat recovery units.

1.7.4 Thermal oxidizers may operate continuously, intermittently, or on demand as part of a batch process, as follows.

- a) In continuous operation, the oxidizer is always in operation as long as the process unit or plant the oxidizer serves is in operation. Even though the operation is continuous, the waste flow and composition may vary. This may cause variations in the effluent gas flow rate with little variation in the temperature of the effluent gas. The type of heat recovery employed must be tolerant of these variations unless augmenting oxygen (air) and supplemental fuel is used to normalize the waste flow variations.
- b) If operation is intermittent, the flow of waste may be temporarily interrupted. This might be the situation with a ship or tank car fuel loading operation where the process (loading) is continuous, but the waste stream (ship or tank vapor content) varies considerably or is reduced to near zero. Such operations make heat recovery from the oxidizer effluent gas very difficult to manage, even with augmenting fuel input.
- c) Batch type operations are usually associated with special manufacturing processes. This type of operation is not usually accommodative of heat recovery from the effluent gas.

1.7.5 While most thermal oxidizers operate at pressures that are near atmospheric, there are some that, due to the upstream process, must operate at substantially higher pressures. Heat recovery units of the watertube type typically have gas side enclosures designed for very low pressures. Designing the enclosure for pressures substantially above atmospheric may require an enclosure employing cylindrical surfaces with wall thicknesses much greater than the 6 mm ($1/4$ in.) thickness usually associated with flat plate, rectangular enclosures. Shell and tube type heat exchangers are more adaptable to heat recovery from high-pressure thermal oxidizers.

1.7.6 The effluent gas from a thermal oxidizer combusting a gaseous waste stream may contain compounds that are either corrosive or erosive. The oxidation of certain components when combined with the water vapor typically present in the effluent gas may result in an acidic condensate at temperatures not uncommon in heat recovery units. The waste gas stream may also contain particulate matter (e.g. catalyst fines) that can be highly erosive in a heat recovery unit at flowing velocities conducive to high heat transfer coefficients. A complete characterization of the effluent gas from a thermal oxidizer is necessary in the determining the type of heat recovery unit that may be employed.

I.8 Liquid Thermal Oxidizers

I.8.1 Thermal oxidizers handling liquid waste streams are different from gaseous oxidizers in that they must first convert the liquid stream to a gaseous state, or atomized into a near gaseous state, and then oxidize (destroy) the targeted compounds of that stream.

I.8.2 Atomization requires a significant energy input. All conventional forms of atomization are used: mechanical, air, steam, or other gaseous streams. The use of combustible gaseous agents (e.g. natural gas) for atomization is attractive for liquid thermal oxidizers.

I.8.3 Heat recovery units placed downstream of liquid thermal oxidizers may require special accommodations (e.g. sootblowers) for removal of precipitates on the heat exchanger surfaces. Sometimes the nature of the precipitates is such that they can only be removed when the heat recovery unit is taken offline. Inclusion of bypass ducting for such procedures may be warranted. Special precipitate collection hoppers located beneath the heat recovery unit may also be warranted.

Annex J (informative)

Process/Flue Gas Heat Recovery Systems

J.1 General

J.1.1 Process/flue gas heat recovery units are a specific type of WHRU. These services are generally firetube type (hot gas inside the tubes) equipment. In these units, heat is recovered from very-high or moderate temperature gas; the recovered heat is often used to generate steam with boiler water present on the shell side of the heat exchanger or some other type of utility fluid. The cold utility fluid absorbs heat from a hot gas passing through the tubes. The hot fluid is often a high-temperature gas resulting from combustion or other chemical reaction.

J.1.2 This type of WHRU is typically designed in accordance with API Standard 660 as a shell and tube heat exchanger.

J.2 Heat Recovery by Steam Generation

High-temperature severe service firetube HRSGs are supplied with boiler water at a high circulation ratio. Natural (thermosiphon) or forced (pumped) circulation systems are employed. BFW is introduced to an overhead steam drum, which provides for water storage and steam-water separation in addition to the static head driving force for natural circulation systems.

J.3 Heat Recovery from Process Gas

Process gas may be used as a heat medium in conventional shell and tube type heat exchangers. The heat from the process gas may be typically transferred to a utility stream.

J.4 Application

J.4.1 High-temperature/High-flux HRSG Units

J.4.1.1 Firetube HRSGs with high-temperature gases [exceeding 480 °C (900 °F)] resulting in high boiling flux rates (in excess of) 94,600 W/m² (30,000 Btu/hr-ft²) are considered severe service applications. Gas temperatures exceeding 1090 °C (2000 °F) and flux rates to 315,500 W/m² (100,000 Btu/hr-ft²) can be accommodated in firetube HRSGs. Mechanical features as described in E.6 are required for these severe services.

J.4.1.2 The following processing applications are typical of those which often make use of severe service firetube HRSGs:

- a) steam reformer effluent (hydrogen, methanol, ammonia plants),
- b) ethylene plant furnace effluent,
- c) coal gasifier effluent, or
- d) sulfuric and nitric acid reaction gases.

NOTE Typical steam-side operating pressures range from 1050 kPa(g) (150 psig) to 12,400 kPa(g) (1800 psig) when used for ammonia and ethylene facilities.

J.4.2 Moderate-temperature/Low-flux HRSG Units

Firetube HRSGs which handle hot gas temperatures that are normally less than 480 °C (900 °F) with flux rates of 94,600 W/m² (30,000 Btu/hr-ft²) and below have a wide range of process applications. Any hot vapor stream with a sufficient temperature above the steam saturation temperature can be utilized. Typical process applications include the following.

- a) Secondary ethylene effluent cooling.
- b) Miscellaneous refinery hot oil and vapor streams.
- c) Sulfur recovery condensers.

NOTE Typical steam side operating pressures range from 350 kPa(g) (50 psig) to 4150 kPa(g) (600 psig) for these applications.

J.4.3 Moderate-temperature Process Heat Exchangers

Heat recovery heat exchangers also handle hot gas temperatures that are normally less than 370 °C (700 °F). The cold side can be a process or utility fluid. These units are often constructed as a typical shell and tube heat exchanger. Some applications may be:

- a) Tertiary ethylene effluent cooling
- b) Miscellaneous refinery hot vapor streams
- c) Preheating for BFW or hot utility oil or water

J.5 System Consideration

J.5.1 Process Heating Medium

The thermal-hydraulic performance and mechanical construction of the equipment to a large degree are dependent on specific characteristics of the hot process fluid. Each fluid has its own unique aspects which must be accounted for in the firetube steam generator or heater design to assure reliable operation. For example, increased fluid hydrogen content may significantly increase the heat flux.

J.5.2 Fouling

Fouling of the tube inside surface in firetube HRSGs is largely a function of the specific process fluid. It is also dependent on velocity, residence time, tube size and orientation, and wall temperature. Examples of specific concerns include the following.

- a) Ethylene furnace effluent quench coolers are subject to coke deposition due to continuation of the cracking process at elevated temperature. Therefore, high gas velocities resulting in minimum residence time at temperature are used.
- b) Hydrogen plant steam/hydrocarbon reformer effluent HRSGs are subject to silica fouling when improper refractories are used in the upstream secondary reformer (for ammonia facilities), transfer lines, or steam generator inlet channels.

- c) Sulfur recovery plant condensers are subject to the accumulation of partially liquid sulfur in the bottom of the tubes.
- d) Water side fouling can be caused by operating at too high a heat flux, poor boiler water maintenance, improper blowdown, low-circulation rates, or any combination of these issues. When water scale deposits on the outside surface of high-temperature tubes, the fouling actually acts as an insulator to the heat transfer on the water side which causes the tube temperature to approach that of the hot gas temperature. This is one of the known causes of failure in severe service firetube HRSGs.

J.5.3 Velocity

The fluid velocity inside the tubes must meet certain minimum criteria for the specific processes. There are also maximum velocity limitations with respect to the erosive nature of particulate bearing streams. In most cases, however, the velocity is set by maximum pressure drop or by maximum allowable heat flux limits which must be considered in design. The range of acceptable velocities should be specified.

J.5.4 Pressure Drop

Pressure losses across the tube side of a firetube HRSG are limited by overall system considerations. For instance, the performance of an olefins plant cracking furnace is penalized by excessive backpressure imposed by downstream firetube quench coolers. The typical allowable pressure range for each application should be specified.

J.5.5 Pinch Temperature

The degree to which the heating medium is required to approach the steam saturation temperature strongly affects the HRSG. As the design pinch temperature is reduced, the mean temperature difference (MTD) decreases, and the surface area requirement increases. HRSGs with large pinch temperatures tend to use larger diameter or shorter tubes than those with small pinch temperatures. The typical pinch temperature range is 8 °C to 14 °C (15 °F to 25 °F).

J.5.6 Outlet Temperature Control

J.5.6.1 Certain processing applications require close control of the hot fluid's outlet temperature. For instance, secondary reformer effluent in an ammonia plant enters a CO to CO₂ shift reactor after being cooled by the firetube HRSG. Overcooling by the HRSG adversely affects the shift reaction catalyst. For this reason, such firetube HRSGs incorporate a hot gas bypass system, which may be either internal or external to the HRSG. Refer to E.6.9 and L.2.4.8 and Figure E.17 for further details.

J.5.6.2 The amount of gas bypassed is a function of turndown, extent of fouling, and the design temperature approach. The equipment tends to overcool the hot fluid when run at reduced throughput and when clean. HRSGs with large design approaches tend to overcool due to the large approach (serving as thermal driving force) at the outlet end. Such units require large bypass systems for temperature control.

J.5.7 Gas Dew Point

Process fluid gas streams which may reach the dew point of one of the gas constituents require special attention. Condensation can occur on cold surfaces such as the tubes and refractory lined walls even though the bulk gas temperature may be above the dew point. Condensation should be avoided and the selection of materials of construction should be reviewed.

J.5.8 BFW/Steam

J.5.8.1 General

API Recommended Practice 538 (which addresses watertube type HRSGs) provides general information with regard to the BFW/steam system. Additional considerations unique to firetube equipment are included within this section.

J.5.8.2 Heat Flux

Maximum allowable heat flux rates for firetube HRSGs are a function of equipment construction details, steam pressure, recirculation rates, and water quality. Specific construction features which affect flux limits include the following.

- a) Tube quantity, diameter, and pitch; in general, flux limits are lower for increasing tube quantity or decreasing pitch to diameter ratio.
- b) Quantity, size, and location of risers and downcomers.
- c) Clearance between bundle and shell.

NOTE Firetube HRSG design should account for increased hot process fluid heat transfer coefficients due to tube entrance effects.

J.5.8.3 Boiler Water Circulation

J.5.8.3.1 Critical service, high-temperature firetube HRSGs are furnished with elevated steam drums, from which boiler water is supplied with high-circulation rates. Systems may be either natural or forced circulation, with the former being most common.

J.5.8.3.2 Low-flux HRSGs may also be furnished with an external drum. However, such HRSG equipment more commonly makes use of an expanded shell side compartment with the tube bundle submerged in the boiler water (referred to as kettle type). Liquid disengagement occurs above the established liquid level within the expanded shell. Such a unit is commonly referred to as a kettle type boiler. Natural circulation patterns occur within the kettle shell. A water-steam mixture rises through the tube bundle; the vapor rises through the steam/water interface to the steam space above; and the boiler water recirculates back down each side of the bundle to the bottom of the shell. The kettle HRSG shell serves the purposes of a steam drum in a conventional boiler system. It differs from a conventional drum in that the HRSG heating surface is self-contained, different connections are required, and steam/water internal flow patterns are different. Saturated steam generated in kettle HRSGs is normally used for noncritical services so that the requirements for purity and quality may be relaxed. Therefore, separation is commonly achieved by such things as deflector plates, dry pipes, or external steam/water separators.

J.5.8.3.3 The kettle type HRSG should include a level control system to ensure that the tubes are always fully submerged and not subjected to dry conditions, as this will create excessive tube wall temperatures and high tube-to-tubesheet joint stress and potential tube failures.

Annex K

(informative)

Process Liquid Heat Recovery Systems

K.1 General

K.1.1 If a hot process liquid stream is not used for direct process heat interchange, it may be hot enough to be used as a source of heat energy to a utility stream.

K.1.2 Crude, vacuum, coker, hydroprocessing units may be used to heat hot oil, hot water or to generate steam.

K.1.3 These systems are typically treated as conventional heat exchangers like those covered by API Standard 660. Other types of heat exchangers such as API Standard 663 or API Standard 664 may be used.

K.1.4 Process fluids that are solid-liquid slurries have special considerations for minimum and maximum velocity. Minimum velocities are established to avoid settling of the solids while maximum velocities are established to avoid erosion of the exchanger.

K.1.5 Slurry velocities at specified process flowing conditions typically should be between 1.4 m/s (4.5 ft/s) and 2.5 m/s (8.0 ft/s). The minimum velocity should be maintained at turndown conditions.

Annex L (informative)

Sulfur Recovery Unit Heat Recovery Systems

L.1 General

L.1.1 In a sulfur recovery unit (SRU), thermal reactor steam generators (TRSG) and sulfur condensers are common types of waste heat recovery equipment.

L.1.2 SRU designs are often based on the modified Claus process that uses one non-catalytic conversion stage (also known as a thermal reactor or reaction furnace) followed by two or three catalytic conversion stages in series to convert hydrogen sulfide (H_2S) to elemental sulfur. SRU process gas contains hydrogen sulfide (H_2S), sulfur dioxide (SO_2) and elemental sulfur and operates at near ambient pressure. The Claus reaction is highly exothermic, releasing heat energy that is typically recovered as high- and low-pressure steam, or provides preheat to boiler feedwater or heat to other heat transfer fluids, in waste heat equipment following the conversion stages to maximize sulfur recovery.

L.1.3 The mechanical design requires the use of appropriate pressure design codes. The two sides of the equipment may require the use of different pressure design codes, or possibly none.

L.2 Thermal Reactor Steam Generators

L.2.1 General

L.2.1.1 TRSGs (also known as waste heat steam generators) are firetube type exchangers which recover heat from the combustion gases exiting the thermal reactor at temperatures in the approximate range of 925 °C to 1540 °C (1700 °F to 2800 °F). The shell side typically generates steam in the pressure range of 2400 kPa(g) to 4500 kPa(g) (350 psig to 650 psig). These are typically either thermosiphon, kettle, or partially tubed horizontal configurations, although alternate firetube type exchanger arrangements can also be used.

L.2.1.2 Applications using other HTFs other than water/steam can be used but are still referred to as TRSGs in this section.

L.2.1.3 The TRSG should be free draining and may be sloped on the process gas side down toward the outlet channel to ensure self-drainage of liquid sulfur in the tubes.

L.2.1.4 TRSGs are typically of single tube pass construction with fixed tubesheets, as illustrated in Figure E.1 and Figure E.3, or two-pass construction with fixed tubesheets as shown in Figure E.16.

L.2.1.5 Since the bundles are not removable, inspection openings (typically 150 mm to 200 mm [6 in. to 8 in.]) installed on the shell are recommended to allow inspection of the exterior of the tubes and the shell side of the tubesheets. The shell side of the hot tubesheet is normally prioritized.

L.2.2 Thermosiphon Steam Generators

A thermosiphon steam generator is provided with an elevated steam drum located above the steam generator, for boiler water storage and steam-water separation. Water from the steam drum is fed to the heat exchanger, as described in E.5.2.a) and shown in Figure E.1. Setting the steam drum elevation, drum operating and alarm levels, and selecting the adequate quantity, size, and location of risers and downcomers is essential for reliable

operation of thermosiphon type TRSGs. See E.6.10 for more information on risers and downcomers. See Annex M for a discussion on heat flux and circulation ratio, which is applicable to all types of HRSGs.

L.2.3 Kettle Type Steam Generators

Any slope of the TRSG tube bundle should be considered when establishing the liquid and alarm levels. For more information on kettle type waste heat exchangers, see E.5.1 item c), Figure E.3, and Figure E.13.

L.2.4 Mechanical Design Considerations

L.2.4.1 Channels

L.2.4.1.1 In most applications, the inlet channel of the TRSG is directly connected to the thermal reactor. It is internally refractory lined and also has an external thermal protection system (also known as a thermal shroud or the misleading term of rainshield). Refer to API Standard 565.

L.2.4.1.2 Outlet channels should be fully refractory lined when the process gas temperature exceeds the sulfidation limit for carbon steel in this service, typically 340 °C (650 °F). If the TRSG outlet channel is not fully refractory lined and the process outlet nozzle(s) are at the bottom of the channel, then partial refractory lining can be provided on the bottom segment of the channel and graded toward the process gas outlet nozzle or liquid sulfur drain nozzle to allow drainage and prevent the accumulation of liquid sulfur.

L.2.4.1.3 The selection of refractory thickness and materials, and their installation method, must be compatible with the hot side service conditions. For more information, see E.6.1.2.

L.2.4.1.4 Manway access should be provided for the inlet and outlet channels to permit inspection, cleaning, repair or replacement of the refractories, ferrules, tube-to-tubesheet joints, and tubes. The outlet channel may be fitted with a removable end plate rather than a manway.

L.2.4.2 Tubesheets

L.2.4.2.1 TRSGs are high-temperature, high-flux (severe service) HRSGs and should be thin fixed tubesheet exchangers with ceramic ferrules as described in E.6.2.

- a) The inlet tubesheet is usually made of carbon steel plate, although sometimes a corrosion resistant alloy layer is applied to the hot face when sulfidation corrosion is a concern at the operating temperatures.
- b) The thin tubesheet design is based on the use of the stays to provide the necessary support for the tubesheets in sections without tubes. Thin tubesheet stays are most commonly dummy tubes or rods. See E.6.2.2.

L.2.4.2.2 Sufficient cooling of the inlet tubesheet is a priority design factor as described in E.6.2.3 and E.6.2.4. Steam blanketing at the inlet tubesheet can cause excessively high temperatures at both the tubesheet and the hot end of the tubes resulting in tube failure.

L.2.4.2.3 Design considerations to mitigate and reduce the potential for a TRSG tube failure and to prevent steam blanketing at the hot tubesheet include, but are not limited to, the following (see also E.6.2.3, E.6.7.3, and E.6.10.2):

- a) High circulation ratio at the hot tubesheet of a thermosiphon TRSG by sizing the downcomer and riser and steam drum elevation. Typically, a suitable circulation ratio is between 15:1 and 25:1.
- b) Minimize distance of downcomers and risers to the hot tubesheet within a thermosiphon.

- c) Design the TRSG to operate within heat flux parameters (mass flow and temperature) and with adequate steam and water circulation to ensure and maintain nucleate boiling instead of steam blanketing for all operating conditions. Typically, the maximum heat flux occurs at the outlet end of the ferrule.
- d) Increased tube pitch and layout above the TEMA minimum requirement, typically 1.4 to 1.6 times the tube outside diameter as a minimum, dependent upon process and geometric considerations. This also reduces the potential for high-temperature sulfidation of the carbon steel tubesheet.
- e) Suitable disengagement space, above the liquid level, to allow vapor escape from the bundles within a kettle. See E.7.5.
- f) Design of insulating ferrules and tubesheet refractory. Design the TRSG tubesheet refractory protective system (including ceramic ferrules) to avoid high-temperature sulfidation of the carbon steel tubesheet (consider tube mass velocity limits) as described in E.6.2.4. For details of the refractory lining, anchors or ceramic ferrule inserts see API Standard 565.

L.2.4.3 Tube-to-Tubesheet Joints

L.2.4.3.1 The tube-to-tubesheet joints provide positive sealing between the tube side combustion gas stream and the higher-pressure shell side water-steam mixture under pressure and thermal loads in all operating conditions.

L.2.4.3.2 TRSG design considerations to mitigate and reduce the potential for a tube failure include but are not limited to the following.

- a) Construction of the tube-to-tubesheet welds with full-depth strength welds (or equivalent) at the inlet tubesheet and conventional strength welds at the outlet tubesheet per pressure code. See E.6.3.2.b) and Figure E.10 for details on full-depth strength welds. See E.6.3.2.a) and Figure E.9 for details on conventional strength welds. Internal bore welds (back face weld) may be used. See E.6.3.2.c and Figure E.11.
- b) If the tube joint is expanded in addition to strength welded, the tube should be expanded through the thickness of the tubesheet to seal the crevice. However, the expansion should not encroach within 6 mm ($\frac{1}{4}$ in.) from the weld and 3 mm ($\frac{1}{8}$ in.) from the back side (shell side) face of the tubesheet. Overexpansion of the tubes should be avoided.

L.2.4.4 Tubes

L.2.4.4.1 Typical tube outside diameters in TRSGs range from 38 mm to 100 mm (1.5 in. to 4 in.), although other size tubes have been used. They may be supplied as piping or tubing. If piping is used, attention to tolerances should be considered to ensure compressibility of the ceramic paper wrap between ferrule and pipe.

L.2.4.4.2 Tubes of this size offer the following advantages:

- a) accommodate the low-pressure drops requirements in the acid gas streams;
- b) keep heat fluxes at the beginning of the tubes low to prevent local overheating of the hot tubesheet, tubes, and tube-to-tubesheet joints;
- c) accommodate the installation of tube inlet ferrules without restricting the flow area of the tubes; and
- d) limit the potential for plugging of tubes.

L.2.4.4.3 The selection of tube diameter should consider the operating range of the process gas mass velocity. It is suggested to maintain process gas mass velocities in the desired range of 10 kg/(m²-s) to 22 kg/(m²-s) [2.0 lb/(ft²-s) to 4.5 lb/(ft²-s)]. The mass velocity is only one of several parameters that are critical to the design and

reliable operation of the TRSG. The TRSG can be successfully and reliably operated outside this range, but other process and heat transfer parameters would require further evaluation and monitoring.

L.2.4.4.4 The minimum tube wall thickness is governed by applicable code rules. In most cases, the shell side design pressure controls the minimum tube wall thickness.

L.2.4.4.5 Construct TRSG tubes from seamless carbon steel tube/pipe.

NOTE See E.6.7.3 for tube arrangement and spacing considerations.

L.2.4.5 Tubesheet Peripheral Knuckles

L.2.4.5.1 Each thin tubesheet is generally attached to the shell with a peripheral knuckle between the flat (tubed) portion and the outer shell, as shown in Figure E.12, types c), d), e), and g). The knuckle provides this critical joint with the necessary flexibility to absorb the axial differential movement between tubes and shell caused by operating temperatures and pressures.

L.2.4.5.2 As an alternative, TRSGs can be designed using a stiffened thin tubesheet instead of a peripheral knuckle configuration, as described in E.6.5.

L.2.4.6 Baffles

Baffle design should leave a peripheral gap between the baffle outside diameter and the shell inside diameter to allow axial flow and distribution of water and steam throughout the shell. These baffles are not standard TEMA baffles as they will have increased clearance from the baffle, notches, or similar, to the shell.

L.2.4.7 Steam Drum

L.2.4.7.1 Steam drums are normally provided for thermosiphons complete with all the required and applicable connections, manways, and internals (demister, steam/BFW separators, BFW distribution pipe, and blowdown collection pipe).

L.2.4.7.2 TRSG steam drum design considerations to mitigate and reduce the potential for a TRSG tube failure in sulfur recovery units include but are not limited to:

- a) designing the steam drum for additional water storage to accommodate sudden changes in heat load, and
- b) providing adequate instrumentation and controls. See A.3.14 and E.10.

L.2.4.8 Gas Bypass Systems

Gas bypass systems are not commonly used for a waste heat steam generator in SRU thermal reactor service. If a gas bypass system is used for TRSG process side outlet temperature control, then an internal bypass system is typically used. See E.6.9 for details.

L.2.4.9 Overpressure Protection Mitigation Considerations

L.2.4.9.1 Due to the potential of over pressure due to tube failure, the design of the channels should consider this scenario. See API Standard 521 and API Standard 565.

L.2.4.9.2 Design considerations to mitigate and reduce the potential for a TRSG tube failure in new or for significant modifications in existing sulfur recovery units are given below.

- a) Design the instrumented protective system to positively isolate all sources of feed and oxygen to the thermal reactor on minimum required TRSG water level. See E.7.5.1, E.10.2.2, and E.10.3.1.
- b) Design and implement an effective blowdown strategy to ensure that BFW impurities are removed from the TRSG to avoid scale and sludge accumulation. See E.10.3.3 and E.10.3.4.
- c) Install view port(s) in the thermal reactor to allow visual monitoring of the lower portion of the tubesheet to detect water leaks.
- d) Design instrumentation system features to ensure continuous monitoring of the thermal reactor temperature; consider alarms and trend recording.
- e) Design the sulfur plant process side pressure for the higher of that required by NFPA 69 with considerations for deflagration and the expected overpressure resulting from a TRSG tube rupture. See API Standard 565 for more information.
- f) For units with a separate steam drum, a level gauge should be provided on the tube bundle to provide visual verification that the tube bundle remained submerged on loss of steam drum level.

NOTE It is presumed that process gas flow is tripped on loss steam drum level. This additional verification is to determine if the tube bundle has been uncovered. Reintroduction of water is not recommended until the unit has cooled. Failure to do so has resulted in loss in pressure containment.

- g) Protection should be provided on the steam line to limit the back flow of steam and flow of BFW into the process side in the event of a tube failure. See API Standard 521 for additional information.

L.3 Sulfur Condensers

L.3.1 General

L.3.1.1 In a typical SRU, the process to remove hydrogen sulfide (H_2S) and sulfur recovery takes place in 3 or 4 conversion stages. In each stage, a sulfur condenser is provided to recover heat from the low-pressure process gases containing elemental sulfur, usually between 200 °C and 350 °C (400 °F and 660 °F), coming out of the TRSG (in the first stage) and Claus reactors (in the second to fourth stages), to condense and extract liquid sulfur and produce low to medium pressure steam, typically in the range of 300 kPa(g) to 520 kPa(g) (45 psig to 75 psig). Higher pressures may be used in boiler feedwater preheat services.

L.3.1.2 Like the TRSGs, sulfur condensers are horizontal firetube shell and tube exchangers with fixed tubesheets. They are sloped toward the outlet channels to ensure self-drainage of the tubes. The shells of sulfur condensers are typically either kettle type (see Figure E.3) or partially tubed type (see Figure E.13) having the same diameter as the channels with an adequate steam disengagement space provided above the top tubes. Thermosiphon systems with external steam drums can also be used.

L.3.1.3 The sulfur condenser of each SRU process stage typically has its own shell, but in smaller SRUs, the same shell can house multiple condensing bundles in a common shell.

L.3.1.4 The bundles are typically not removable, therefore, handholes should be installed on the shell to allow inspection of the exterior of the tubes and the shell side of the tubesheets.

L.3.2 Mechanical Design Considerations

L.3.2.1 Channels

L.3.2.1.1 Sulfur condenser inlet channels may be fully refractory lined to avoid sulfidation attack or damage from a fire in the first catalytic reactor, as described in L.2.4.1.2 for TRSG outlet channels.

L.3.2.1.2 Whether or not the sulfur condenser inlet channel is fully refractory lined for sulfidation protection, partial refractory lining is installed in the bottom segment of the channel, up to the edge of the bottom row of tubes, to avoid sulfur accumulation at the inlet of these tubes.

L.3.2.1.3 The bottom of the outlet channel is also usually refractory lined to avoid having a pocket for sulfur accumulation. Alternatively weld overlay can be used to protect against sulfidation attack.

L.3.2.1.4 Liquid sulfur outlet nozzle and the outlet channels may be heated. Steam jackets or electrical tracing may be used to maintain a minimum inside wall temperature of in the approximate range of 125 °C (260 °F) to prevent low-temperature corrosion and sulfur plugging.

L.3.2.1.5 Outlet channels are typically extended in length so that a mist elimination pad can be installed between the outlet tube ends and the vapor outlet nozzle. Gravity separation should be the sizing basis.

L.3.2.1.6 Access into the inlet and outlet channels is generally through a manway in large diameter units, or through full access covers in small units.

L.3.2.2 Tubesheets

Knuckled thin tubesheets, or conventional thicker tubesheets can be used in sulfur condensers.

L.3.2.3 Tube-to-tubesheet Joints

Conventional strength welded tube-to-tubesheet joints, see E.6.3.2.a) and Figure E.9, are recommended to avoid potential leaks of BFW/steam into the low-pressure process gas stream. Full depth expansion should be included.

L.3.2.4 Tubes

L.3.2.4.1 Typical tube outside diameters in SRU steam generators range from 25 mm to 50 mm (1 in. to 2 in.), to accommodate the low-pressure drops required in the process gas streams and minimize plugging.

L.3.2.4.2 Tubes are normally arranged on a triangular pattern to provide the smallest shell diameter.

L.3.2.5 Baffles

Baffle design should leave a peripheral gap between the baffle outside diameter and the shell inside diameter to allow axial flow and distribution of water and steam throughout the shell. These baffles are not standard TEMA baffles as they will have increased clearance from the baffle, notches, or similar, to the shell.

L.4 Tail Gas Treatment

In a typical SRU, sometimes the tail gases are treated to reduce the loss of sulfur containing compounds. The removed sulfurous materials may be recycled to the upstream portions of the sulfur plant. In this treatment, it is possible to have an additional heat recovery unit. In other cases, the tail gas may be routed directly to a thermal oxidizer.

Annex M (informative)

Heat Flux and Circulation Ratio

M.1 General

Circulation, heat flux and boiling flow regimes are fundamentals applicable to all HRSGs, regardless of whether the system is forced or natural circulation, watertube or firetube design.

M.2 Heat Flux

M.2.1 Film Boiling

M.2.1.1 Heat flux is the heat transfer rate per unit area of tube surface measured at the surface where boiling occurs. If heat flux is excessive, steam is generated so rapidly that a stable steam film is formed at the tube wall. This steam film prevents water from contacting the tube wall. This phenomenon is known as film boiling or departure from nucleate boiling and results in a sudden increase in the tube metal temperature. This can cause tube failure resulting from high metal temperature.

M.2.1.2 The heat flux at which departure from nucleate boiling occurs depends on several variables including:

- a) orientation and geometry of the surface,
- b) circulation ratio,
- c) steam/water mixture velocity, and
- d) pressure.

M.2.1.3 The actual heat flux must be less than the maximum allowable heat flux determined with the above considerations for all operating cases.

M.2.1.4 Firetube HRSG design should account for increased hot process fluid heat transfer coefficients due to entrance effects, as well as for radiative heat transfer emission in high-temperature services.

M.2.1.5 The ratio between the maximum calculated local heat flux at any point along the tube and the maximum allowable (critical) heat flux should not exceed 0.5 for equipment reliability and unit safety.

M.2.2 Nucleate Boiling

M.2.2.1 Since boiling heat transfer coefficients are much greater than that of the hot gas side, tube metal temperatures approach that of the saturated water. This assumes nucleate boiling where steam bubbles generated at the tube wall are alternately displaced by water rewetting the tube.

M.2.2.2 Steam blanketing can also occur at low-heat fluxes with nucleate boiling if the forming steam bubbles are not continuously removed. Operation at heat fluxes above the values shown in Table M.1 can lead to steam blanketing. At this heat flux even the most vigorous circulation cannot prevent the formation of an insulating steam film on the heating surface and film boiling can occur. Film boiling results in a reduced heat transfer coefficient and high tube temperatures which can lead to accelerated corrosion, fouling, or mechanical failure.

M.2.2.3 For kettle steam generators with tube bundles over 900 mm (36 in.) diameter, consideration should be given to providing vertical clear steam lanes within the bundle, together with large shell-to-bundle clearance to prevent steam blanketing.

M.2.3 Local Heat Flux

Table M.1 shows typical ranges of maximum allowable local heat fluxes. The maximum heat flux should be calculated in the area of highest temperature difference based on fluid properties at that temperature and under clean conditions and with consideration of additional turbulence at the end of ferrules when present. Both tubewall temperature and heat flux should be analyzed to determine the operating limits for HRSG. Many industrial HRSG designs have much lower local heat fluxes than the maximum specified in Table M.1. This may be due to low-temperature difference or low overall heat transfer coefficients.

Table M.1—HRSG Firetube and Watertube Local Heat Flux

| HRSG Type | Maximum Allowable Local Heat Flux W/m ² (Btu/h-ft ²) | Comments |
|---|--|---|
| Firetube | | |
| Kettle | 78,800 to 94,600 (25,000 to 30,000) | Pool boiling; circulation pattern is not well defined. Tube spacing must be carefully considered for larger units. |
| Horizontal and Vertical Natural or Forced Circulation Thermosiphon | 220,700 to 315,500 (70,000 to 100,000) | Separate steam drum, well defined circulation pattern. May not be applicable to transfer line exchangers in ethylene plants. Higher fluxes possible in some proprietary designs. Tube spacing must be carefully considered for larger units. |
| Watertube | | |
| Natural circulation | 315,500 (100,000) | Vertical tubes prevent flow stratification. Need to check circulation ratio at the exit of the hottest tube. |
| Forced circulation | 126,100 to 157,600 (40,000 to 50,000) | Design to avoid stratification in horizontal tubes. Need to control high-steam/water mixture velocity. |
| Forced circulation Once-through [115 mm (4.5 in.) max. tube diameter] for enhanced oil recovery | 126,100 (40,000) | Need to control hardness of water used. |

M.3 Circulation

M.3.1 Circulation Ratio

M.3.1.1 Circulation ratio (CR) is defined as:

$$CR = \frac{\text{Total Steam Mass Flow Rate} + \text{Water Mass Flow Rate}}{\text{Steam Mass Flow Rate}}$$

M.3.1.2 The designer sets the circulation ratio to maintain nucleate boiling for all operating conditions, that is, to avoid departure from nucleate boiling. Two-phase flow regime should be considered in specifying minimum circulation ratio.

M.3.1.3 For tube side boiling applications, bubble flow is the preferred two-phase flow regime for HRSG tubes. Flow regimes are a function of tube diameter and orientation, fluid properties and steam/water mass velocity.

M.3.2 Natural Circulation

M.3.2.1 HRSG risers and downcomers form a flow circuit by connecting the steam drum at the top and a water drum or header at the bottom. During operation, the steam/water mixture in the risers is less dense than the water in the downcomers. Flow occurs within the circuit at a rate where the difference in static head between the risers and downcomers balance the resistance to flow. Natural circulation ratios are typically 15:1 to 20:1. Lower ratios are frequently used in higher pressures.

M.3.2.2 The circulation ratio depends on the static head differences, resistance to flow in the circuit, system pressure, and quantity of steam generated. The designer can increase circulation ratio by a combination of raising the height of the steam drum and by reducing flow resistance, for example, larger downcomers or increased flow area.

M.3.2.3 Two-phase flow regime and stability in the exit piping and risers should be considered to achieve an adequate circulation rate for the system during all operating cases.

M.3.2.4 The circulation ratio should be calculated for the anticipated range of operation. Low circulation ratio can result in departure from nucleate boiling and tube overheating. Natural circulation HRSGs generally use vertical or inclined tubes to allow steam to rise freely.

M.3.2.5 For natural circulation systems, the actual circulation ratio will depend on the operating steam rate as compared with the design steam rate. An example of how the circulation ratio is affected by design load can be seen in Figure M.1.

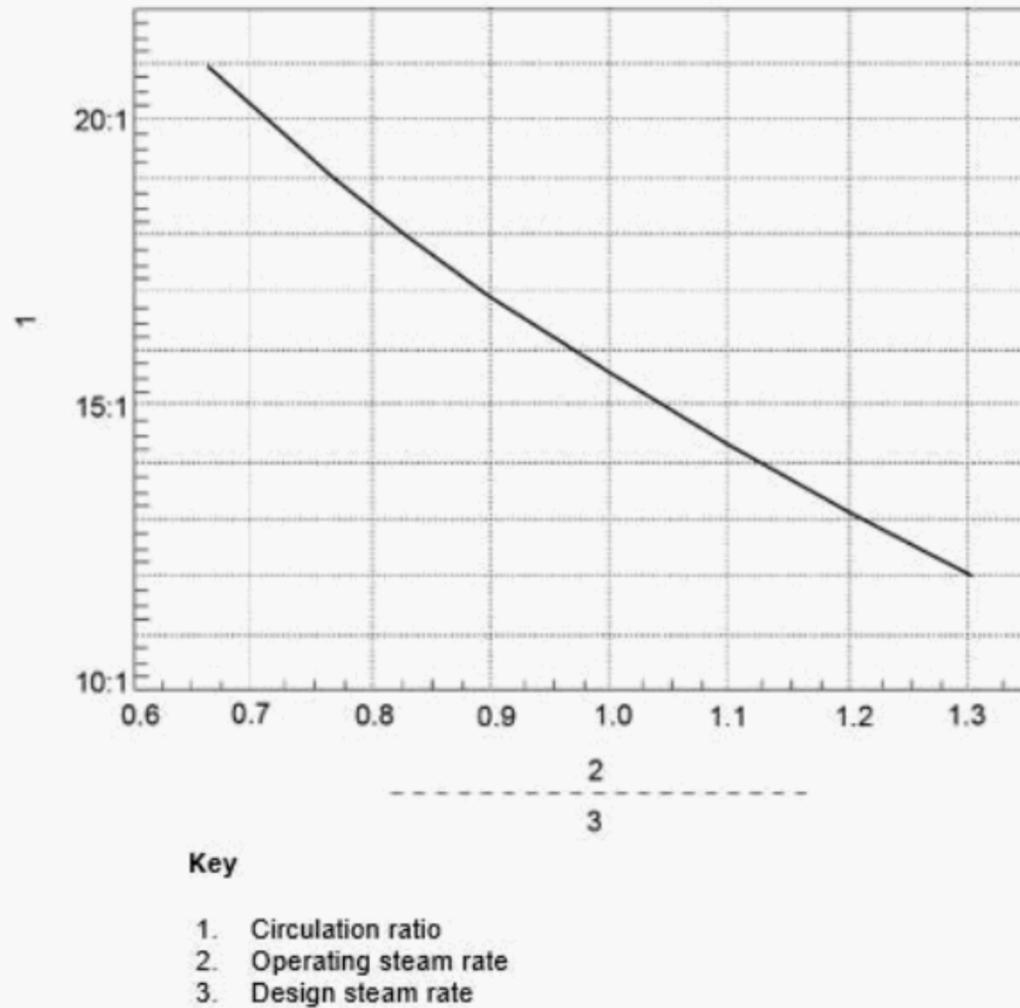


Figure M.1—Example of Circulation Ratio Variability with Steam Rate

M.3.3 Forced Circulation

M.3.3.1 Forced circulation HRSGs use a pump to maintain circulation through the steam generating tubes of the evaporator, steam drum and headers. A minimum circulation ratio of 10:1 should be used. Water is distributed to parallel tube circuits from an inlet header and the exiting steam/water mixture is collected in an outlet header. The steam/water mixture is returned to the steam drum where the steam is separated, and water is recirculated to the evaporator.

M.3.3.2 Steam generator tubes may have any orientation. Fired heater applications are usually horizontal. Tubes are connected in series in a serpentine arrangement to form each single tube pass. With this arrangement water flows upward and improves flow stability between multi-pass parallel circuits. The buoyancy of the two-phase flow assists the forced circulation and minimizes the potential for steam pocketing.

M.3.3.3 Forced circulation HRSGs generally have larger tubes, longer tube circuits and higher flow resistance than natural circulation HRSGs.

M.3.4 Advantages/Disadvantages

M.3.4.1 Natural circulation advantages are:

- a) no pumping systems are required,
- b) less maintenance, and
- c) more reliable.

M.3.4.2 Natural circulation disadvantages are:

- a) usually restricted to vertical or inclined tube applications in watertube applications;
- b) usually installed at grade (more plot space required); and
- c) steam drum location requires higher elevation.

M.3.4.3 Forced circulation advantages are:

- a) horizontal or vertical tube arrangements may be used;
- b) forced circulation arrangements can be installed in vertical heater flue gas ducts;
- c) smaller plot requirements; and
- d) the steam drum location is not restricted.

M.3.4.4 Forced circulation disadvantages are:

- a) pumping systems are required, including standby pump with automatic start;
- b) higher maintenance due to pumps; and
- c) vertical tube arrangements require design expertise.

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