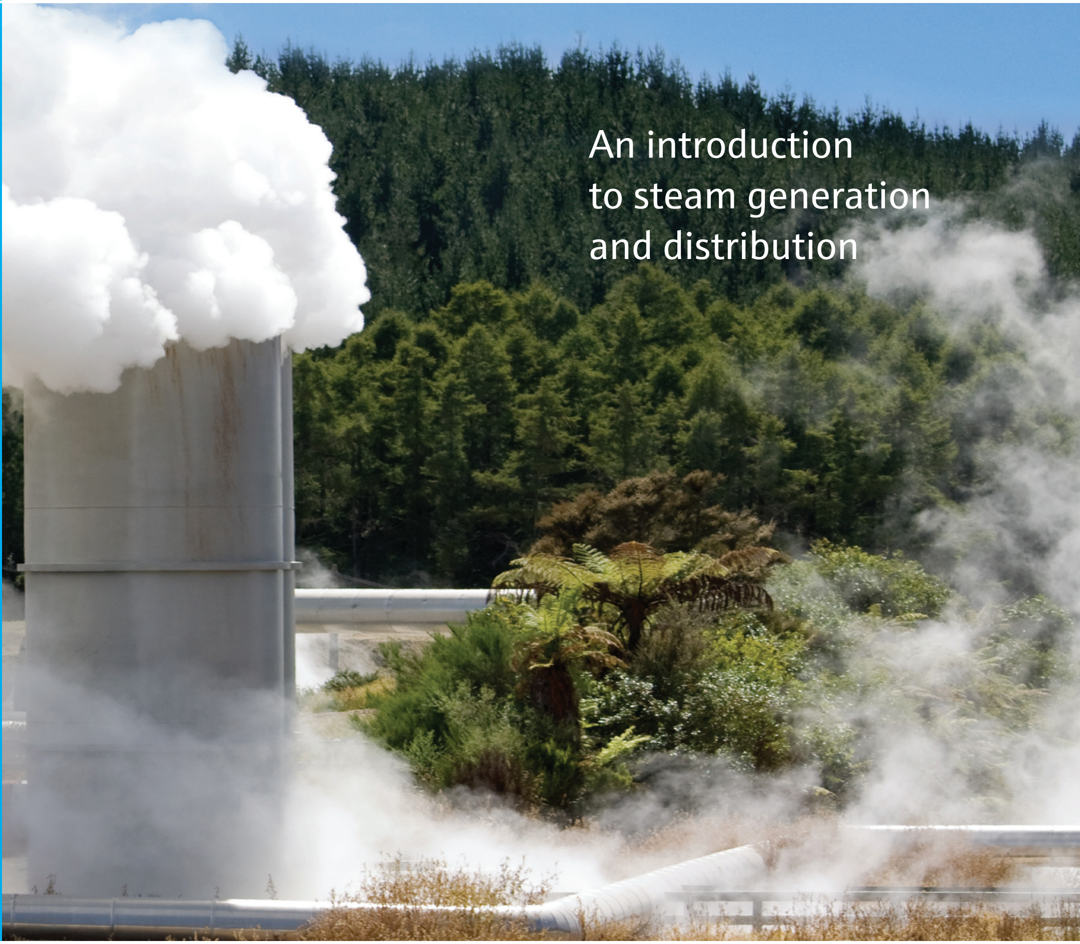
Products Solutions Services

Steam Handbook



An introduction to steam generation and distribution

Endress+Hauser

People for Process Automation

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Dr. Ian Roberts
Phillip Stoor
Michael Carr
Dr. Rainer Höcker
Oliver Seifert

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Editor in chief Thomas Stauss

Editorial team Michael Carr, Dr. Rainer Höcker, Dr. Ian Roberts,
Romeo Rocchetti, Oliver Seifert, Thomas Stauss, Phillip Stoor

Illustrations Kodotec (Lörrach, Germany)

Layout, set Beatrice Meyer

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Foreword

What this document is about?

This document is designed to give a comprehensive technical overview into what steam is in all its forms, how to measure it, where and why it is used for in industrial processes, and to give an insight into unexpected and potentially hazardous situations that may arise from the process configuration and conditions. It uses clear and relatively easy to follow examples, with the minimum of mathematics. However, it does assume a level of interest and curiosity in steam production, distribution and its use, that will be rewarded by taking on-board the principles and practice given in the document.

Who this document is for?

This document is for personnel involved in all aspects of medium sized boiler plant, steam distribution and saturated steam used for both general services and direct process purposes in all industries:

1. Process engineers
2. Energy managers
3. Procurement staff
4. Technical managers
5. Operations managers
6. Instrumentation Sales & Marketing staff
7. Maintenance and application/Support engineers
8. Control and instrument engineers (including DCS)

How to use the document?

For readers who just wish to gain a general understanding, there are sections that can be safely passed over which are included specifically for engineers with a technical interest in understanding the thermodynamics of saturated steam.

This document is not intended to be an engineering design guide, nor is it a commercial guide to a specific manufacturer's equipment. It is intended as both a description of what is objective current best practice – so that readers can make informed equipment design, purchasing and maintenance decisions – and to diagnose/solve problems in existing steam systems.

A short history of boiler designs

Shell and tube-saturated steam boilers of the current packaged form have been manufactured since before the second world war, and their lineage may be traced directly back to the Cornish boilers of the early nineteenth century, invented by the British inventor and mining engineer Richard Trevithick (1771–1833).

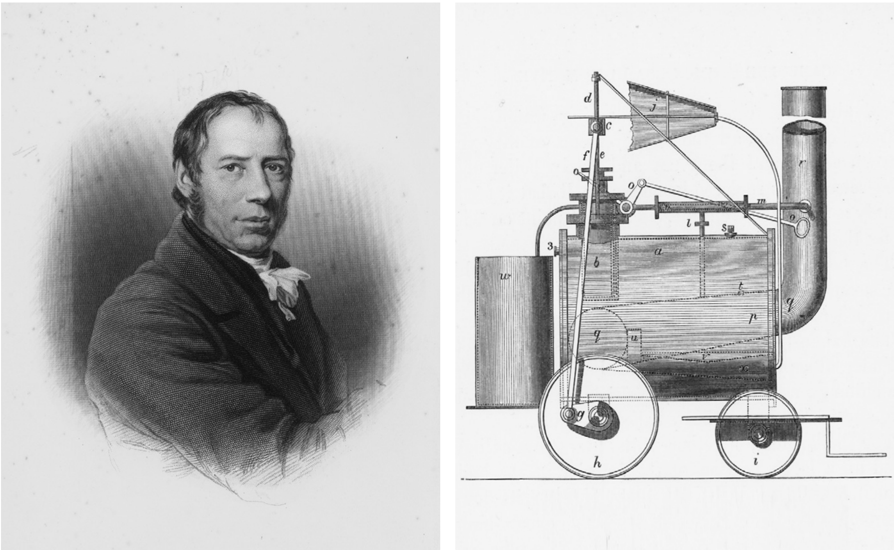


Figure 1: Richard Trevithick (1771–1833). British inventor and mining engineer. Trevithick was the first to successfully use high-pressure steam (then known as "strong" steam). Until ca. 1800, the weakness of existing boilers coupled with the influence of James Watt had generally restricted steam boilers to very low pressures or "weak" steam. Right: Trevithick's "Puffing Devil" – the first steam-powered locomobile drove on streets, not on tracks.

Early steam boilers such as the wagon, the haystack and the egg-ended boiler worked at low pressures (less than 0.7 barg) and were used to drive pumps at mines, particularly the deep mines of Cornwall and the North East via with Newcomen's atmospheric and Watt's condensing engines. These boilers were very inefficient: Murray (1959) reported Brownlie as estimating the efficiency of egg-ended boilers at 34%. As coal was imported to Cornwall by sea and arrived at the tin mines by packhorse, it was a considerable expense to the mine owners. Trevithick realized that if he utilized the power developed from high pressure steam or strong steam as it was then known, the efficiency would be improved.

One of Trevithick's boilers, that manufactured by Hazeldines around 1806 and currently on display at the Science Museum in London, is typical of his designs. It is easily recognizable as the forerunner to today's packaged boilers with the furnace and a single large diameter convective pass mounted inside a cylindrical iron pressure vessel. Trevithick's boilers of this type typically operated at pressures in the range between 1.4 and 3.5 barg. Whilst efficiency data are difficult to find for these early high pressure boilers, that they were significantly more efficient is known. Rolt (1960, p. 168) illustrates this by stating that Dolcoath mine's coal bill reduced from £1000 to £612 per annum following the installation of one of Trevithick's boilers in around 1810.

The Cornish boiler as it came to be known was standardized for larger, stationary boilers in the following arrangement: a single furnace mounted inside the pressure vessel with the hot gases then passing through side mounted external brick-lined flues. This was complemented and then superseded by the Lancashire boiler (Figure 2) patented by Fairbairn and Hetherington in 1844. It utilized twin furnaces arranged side by side inside the pressure vessel. Although this permitted a greater firing rate than that of a Cornish boiler, the volumetric heat release rate was reduced thus reducing the thermal stresses. Crucially, they also permitted a greater depth of water to be maintained over the furnaces thus reducing the risk of a low water incident (Fairbairn and Pole 1910).

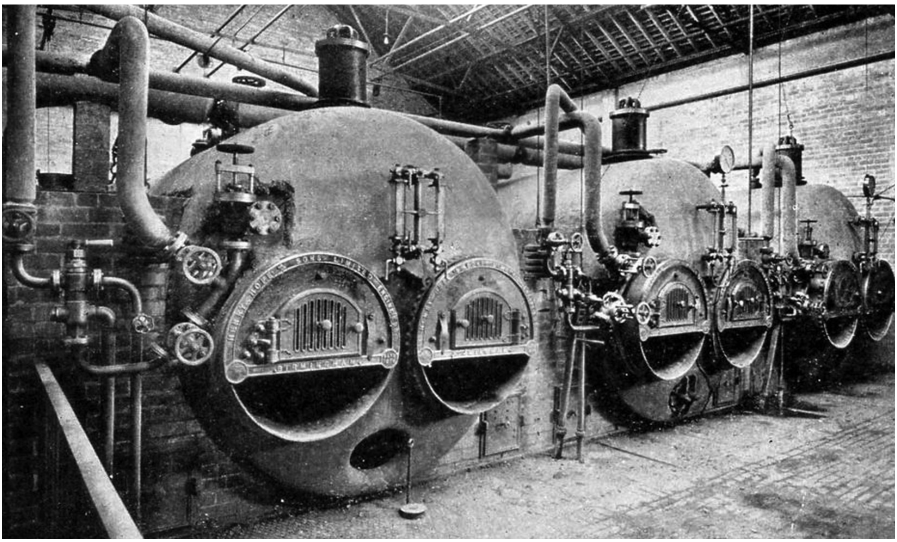


Figure 2: Boiler house with three Lancashire boilers (second half 19th century).
Source: Heritage group of the CIBSE (www.hevac-heritage.org).

In 1858 the *Manchester Steam Users Association for the Prevention of Steam Boiler Explosions* was formed to inspect operational boilers, investigate the causes of explosions and promote fuel economy. A signal feature of the association was that it introduced the concept of insurance to the industry: In the event of the explosion of an approved boiler – for whatever reason – £300 would be paid to its owner (Fairbairn and Pole 1910, pp. 258–268). Whilst the association was formed in part to avoid government interference in the affairs of steam-engine proprietors, it soon became apparent that legislation of some form was required in order to force owners to maintain and operate their boilers in accordance with best practice.

That some form of legislation was required was evident as catastrophic boiler failure and consequent loss of life and production was an everyday feature of steam generation at this time. Hugh Mason M.P. (1882, pp. 1348–1355), during the second reading of the Boiler Explosions Bill 1882 reported that "*one boiler explosion occurred every week, and one person was killed, and two injured, more or less, by a boiler explosion every four days.*"

The success of the *Manchester Steam Users Association* prompted the formation of other similar societies and insurance companies. When Parliament eventually introduced regulations concerning steam boilers, this system of inspection and insurance was adopted and became mandatory.

Simultaneously, various learned societies such as the *American Society of Mechanical Engineers (ASME)* were investigating the fundamental scientific and engineering principles that underlie the use and application of steam. That these were not fully understood, or were plainly wrong, is manifest. For example, Fischer (1874, pp. 311–323), in an address to the Hannover section of the *Society of German Engineers* in *1874*, discussed inter alia the hypothesis that water decomposed into its constituent elements inside the boiler shell thus forming an explosive mixture. The work of these societies led to the introduction of standards for the design and construction of boilers and pressure vessels the *ASME Boiler and Pressure Vessel Code* being the oldest – originally published in 1914 – and possibly best known.

By the end of the nineteenth century, the Lancashire and Economic (a coal fired shell and tube unit) boilers were the dominant types for saturated steam. The packaged boiler concept was imported to the United Kingdom from the United States of America after the Second World War following a visit to that country by a team of experts sponsored by the then Ministry of Fuel and Power (Goodall 1980). A further significant development resulting from this visit was the adoption of oil (replacing coal) from the 1950s onward and supplemented, in turn, by natural gas from the

1960s. This development, in the United Kingdom, was driven by the *Clean Air Act* of 1956 which effectively proscribed the use of coal in cities.

The packaged boiler provided the user with a number of benefits, i.e. boilers that were smaller, cheaper and more efficient than the Lancashire and Economic units which they replaced. They were also supplied with all necessary accoutrements, requiring only the connection of services for them to commence operation, substantially reducing both the time and the cost of installation.

Why use steam?

The thermophysical properties of steam change as its conditions of state change and it is relatively easy to generate steam across a wide range of states. Consequently steam has a very wide range of applications. The following briefly describes a few of the major thermophysical attractions of steam to the engineer. An explanation of the various terms used can be found in the glossary located towards the end of this publication.

- It has a very high heat content, of which a large proportion is contained in the latent heat (enthalpy of change of phase).
- The heat content of the latent heat can be given up and released very quickly.
- Large amounts of steam can be driven through a process very quickly which, combined with the previous point, allows very large amounts of heat to be transferred to the process in a very short time using relatively small heat exchangers.
- The heat content of the enthalpy of change of phase (saturated and wet steam) may be given up in conditions which approximate very closely to constant temperature allowing close process control and process homogeneity.
- Very high steam pressures and temperatures can be achieved which is advantageous for processes such as power generation.
- Steam can be readily distributed and easily controlled.
- Heat may be extracted in a cascade of processes.
- Steam is water and water is required to be added to many foods during their processing.
- Steam is neither toxic nor a fire hazard.
- In many locations, water is cheap, readily available and easily purified.
- In many indirect processes the condensate can be reused in its entirety recovering both the high purity water and its high residual heat content.
- Steam has a wide range of thermophysical properties, the optimum being selected for a given process.

What is steam used for?

We can reduce the answer to this question to three main categories:

- *Power generation:* The steam used for power generation is usually superheated steam as the higher the temperature, the higher the thermodynamic efficiency that can be achieved.
- *Process:* This is predominately saturated steam as it is the latent heat and sometimes the water as well that is required. Moreover the latent heat is given up much more quickly by saturated/wet steam than superheated steam.

  - Indirect processes use just the latent heat of the steam.
  - Direct injection processes consume both the heat and the water.
- *Space heating:* This almost invariably uses saturated steam as the heat transfer rates from saturated steam to the heat exchangers are very much higher than if the steam were superheated.

Where is steam used?

Although the range of applications for steam has reduced e. g. replacement of steam trains, it is still found in a wide range of applications of which just a few are briefly described below:

- *Nuclear power stations:* The nuclear part of these power stations is just a means of generating large amounts of heat. This heat is transferred to water to generate superheated steam which drives a turbine to generate electricity. In fossil fuel power plants, this heat is generated by burning the fossil fuel (e.g. coal).
- *De-commissioning nuclear power stations:* In the United Kingdom, steam is used to keep the pile caps of de-commissioned Magnox stations at a known temperature. (A pile cap can be thought of in very simplistic terms, as the lid of the vessel containing the nuclear fuel).
- *Food processing:* Steam at common pressures (6 to 10 barg) is at temperatures in the range 150 to 180 °C which is ideal for cooking and the steam is very controllable. An example is evaporating milk: steam ejectors are used to evaporate off some of the water fraction of the milk by creating a partial vacuum in the vessel holding the milk thus reducing its saturation temperature.
- *Drinks manufacture e.g. whisky:* In a pot still, used for the manufacture of malt whisky, the distillation process is frequently undertaken using steam in coils as the heat source. This form of distillation is a batch process and the condensate is recovered for re-use. In a continuous distillation process e.g. in a Patent or Coffrey still the steam is directly injected into the still and the steam cannot be recovered.
- *Laundries:* Steam is used for heating ironer beds and heating wash water by directly injection into continuous batch washers. It is also used for conditioning garments prior to folding.
- *Sterilization:* Within hospitals for medical purposes and also for sterilization of any waste which is deemed to require it. Again, the desired temperatures can easily be achieved and maintained in a very controllable manner.
- *Trace heating:* Some fuels such as heavy fuel oil require heating in order to reduce the viscosity of the fuel so that they can be pumped. The same is true for some chemical storage tanks, whereas others require warming for frost protection and other reasons.

A generic steam system

Figure 3 shows a generic steam system. For good reason water is sometimes referred to the "universal solvent" and as such contains a variety of substances a number of which are undesirable to some greater or lesser degree. The make-up water to the boiler therefore needs to be conditioned. It is very important to note that the composition of the make-up water is dependent upon the location and that different localities and water sources will require different treatment regimes. As an example water drawn from a source such as a river will require filtration to remove solid particulates whereas water drawn from a municipal main will not. The primary water treatment system indicated in Figure 3 will have a variety of functions, includ-

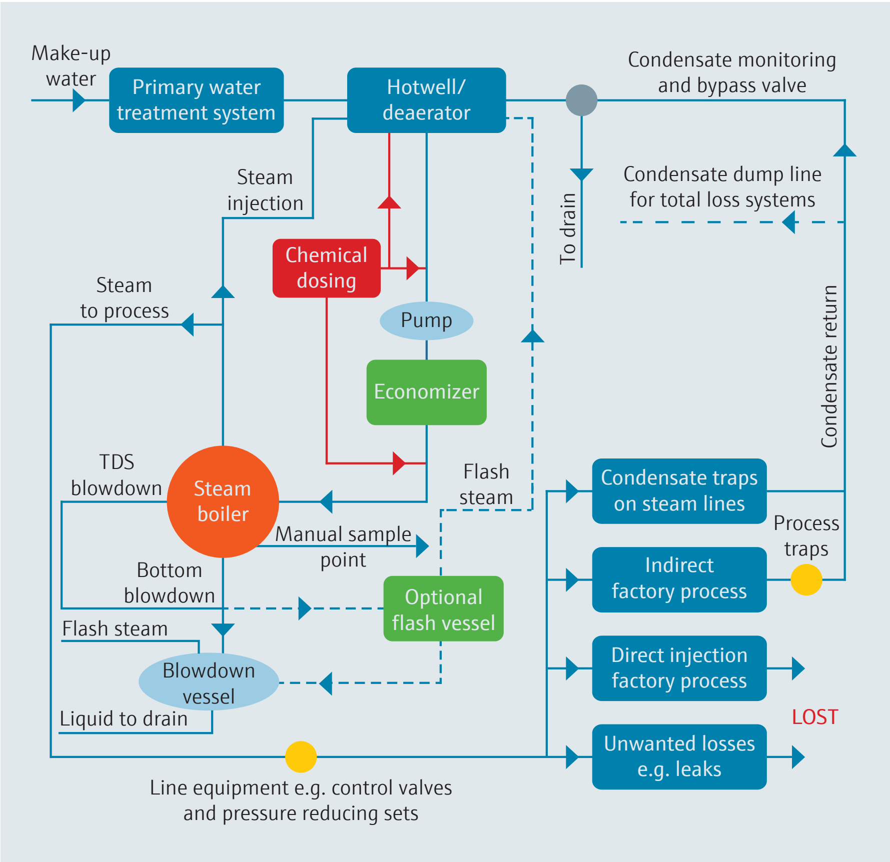


Figure 3: Scheme of a generic steam system.

ing filtration. The most important of these however is the removal of scale forming compounds or their conversion into to substances which do not precipitate out to form hard scale using one of a number of techniques such as base exchange softening or reverse osmosis. This treated water is then conditioned thermally by which its temperature is increased either using condensate return or prime steam injection. This is done to partially remove dissolved oxygen (in an atmospheric hotwell) or to render it a practicable minimum (by use of a deaerator). The increase in temperature also serves to reduce the thermal shock to the boiler which is a significant cause of localized cracking of the pressure vessel. The water is then treated with various chemicals to give it the optimum properties, such as correct pH or required level of oxygen scavenger.

Once conditioned, the feedwater is pumped into the boiler. The function of the boiler is to generate steam by adding heat to the water whilst constraining it both physically and thermally. The pressure inside the boiler is controlled and also limited to a permitted maximum. The water level is also controlled to ensure that the heated surfaces are always submerged. If the level falls below a predetermined set point, limiters are employed to shut the boiler down. The water quality inside the boiler is also monitored and controlled in a variety of ways to ensure that its quality is maintained.

The steam which leaves a boiler contains a very small amount of impurities – much less than the feedwater. This means that solids, both dissolved and suspended concentrate inside the boiler and these must be controlled and removed respectively. Dissolved solids are considered in terms of “Total Dissolved Solids” (TDS) for which there is a permitted maximum which depends upon the type of boiler being used. These are controlled by measuring the amount of TDS in the boiler and then blowing down some of the liquid water in the boiler to remove some of the solids and so reduce the TDS. On modern boilers this is normally accomplished by using an automated system although it may still be performed manually. Suspended solids sink to the bottom of the boiler to form a sludge. These solids are removed using bottom blowdown. This consists of a valve located at the bottom of the boiler which is quickly opened once a day for approximately ten seconds. This sudden discharge removes the sludge from the bottom of the boiler preventing over-heating of the lower part of the furnace. Both TDS and bottom blowdown are normally discharged to a blowdown vessel which safely dissipated its pressure and heat prior to discharge to drain. However, if the quantity of blowdown water is large enough, it may be passed through a flash vessel to recover some of the water and its heat content.

Prime steam leaves the boiler and enters the distribution system by virtue of the pressure difference between the boiler and the various points of use. Some of this steam may be used to heat the hotwell or deaerator, the majority however goes to process. As the steam travels through the distribution system it uses up some of its own energy in doing so, condensing back into the liquid phase and is termed condensate.

Condensate is also formed due to heat losses to atmosphere. As a rule of thumb, an uninsulated pipe will lose at least ten times more heat than an insulated pipe. Condensate is almost without exception, unwanted, and is removed using traps, normally for return to the boiler house where the high-quality water and its residual heat content are recovered.

As the steam approaches, the consumers its pressure is conditioned using pressure reducing valves (PRV) to provide a consistent and controlled pressure onto the process. Steam consumers fall into two main categories: direct and indirect processes. Direct processes consume the steam itself utilising both the water and heat content, so nothing is returned to the boiler-house. Indirect processes utilize just the latent heat content of the steam so that it condenses back to the liquid phase and is normally returned to the boiler house for re-use; although in a few specialized processes the condensate is discarded as a matter of course. In certain processes there is a risk that the condensate becomes contaminated from which dangerous incidents may result. It is therefore very important to assess the risks associated with the potential for contamination of condensate and take appropriate steps such as the inclusion of monitoring systems for condensate and the selection of the most appropriate level controls.

Correct water treatment is vital to the safe, effective and efficient operation of any steam raising plant. Incorrect treatment, failures of equipment and poor operation have resulted in many boiler explosions, accidents and dangerous occurrences over the years.

Types of industrial steam boilers

There are a number of different types of steam boilers. The type preferred for a given installation will depend in the main on the type of steam, steam pressure and output required. Whilst there is some overlap in the capabilities of the different types, it is more usual for a generic type of boiler to "select itself", e.g. multi-tubular horizontal steam boilers (MTHS) in breweries or water-tube boilers for power stations – the purchaser then being restricted to a choice of units from competing manufacturers who may offer different sub-classes within the generic type. The following pages give a brief overview of the existing main boiler types:

- Coil steam generator (once through boiler)
- Water-tube boilers
- MTHS boilers (fire-tube smoke-tube or shell boilers)

Coil steam generator (once through boiler)

Coil steam generators operate with the water contained inside a continuous tube coil (Figure 4). As water is pumped through the coil it is heated by an external flame, and once enough heat has been transferred, the water boils inside the tube to generate steam. The majority of units are supplied to the process industry operating at relatively low pressures (<10 barg) and low capacities (<1.5 MW) although it is possible to generate steam at much higher pressures (>100 barg).

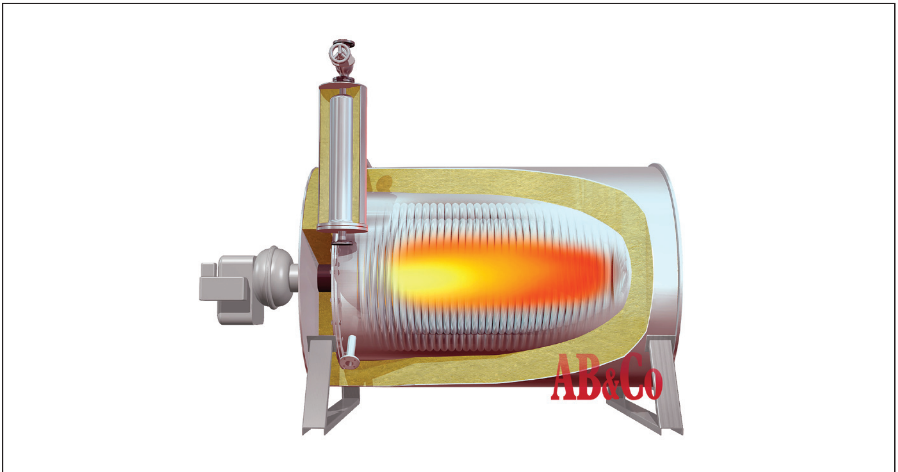


Figure 4: Once-through steam generator (Courtesy: AB&CO – TT BOILERS LTD, Denmark).

The main advantage of a coil steam generator is its ability to produce steam at short notice as they do not contain a large volume of water. Furthermore this small water capacity means that they do not have the explosive capabilities associated with MTHS boilers – although there is, as with all boiler types, the potential for an unwanted, potentially dangerous release of steam. The installation and regulatory environment for these generators are generally less onerous than for MTHS boilers. They are compact units and are often found to be very cost effective where steam demand is infrequent. Against this the steam quality can be low suffering from

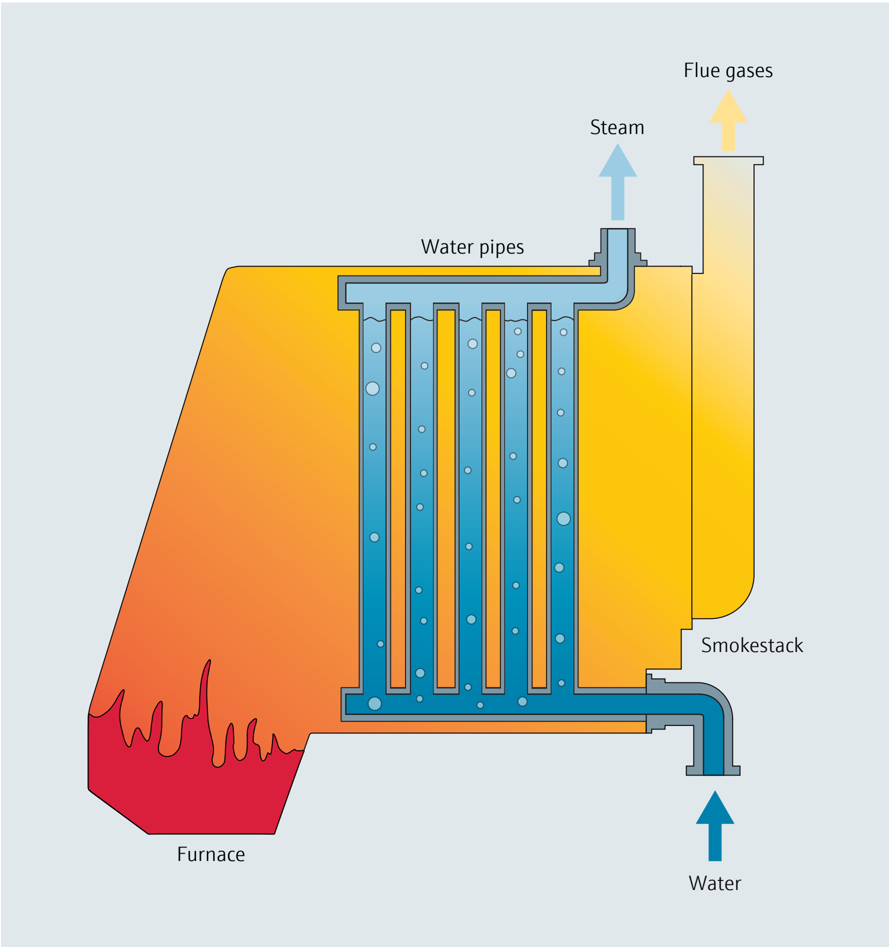


Figure 5: Principle of a water-tube boiler.

problems with wetness and pressure variation as it is difficult to precisely match the burner output to the steam load especially when the demand fluctuates suddenly or at sustained high demand relative to the rated capacity of the generator. They are very sensitive to scale build-up due to poor water treatment and require a relatively high standard of operator training for best performance. Start-up losses are high and steam generators tend to be less economic in situations where high demand is intermittent but frequent.

Water-tube boiler

The water-tube boiler was patented by Babcock and Wilcox in 1867. Figure 5 and Figure 6 show the basic principle of operation: The water is injected into a drum known as the water or mud drum and then flows through the tubes and into the steam drum located at a higher level boiling to form steam. Very high outputs and pressures

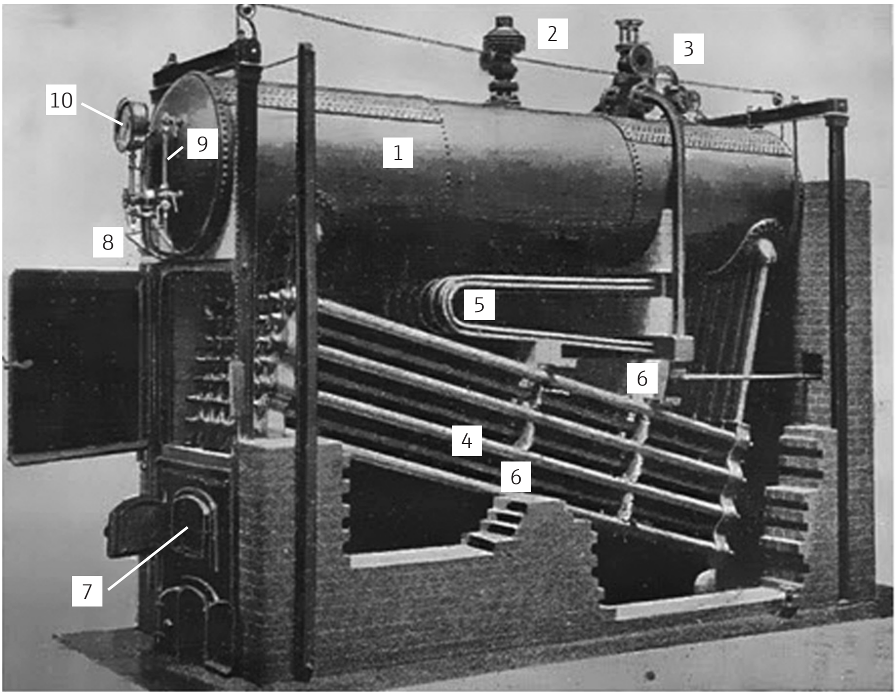


Figure 6: A Babcock and Wilcox water-tube boiler. From: Ripper, William C.H. (1923). Heat Engines, London, p. 207.
1 = Drum (for water and steam), 2 = Safety valve, 3 = Main stop valve, 4 = Water tubes,
5 = Superheater tubes, 6 = Baffle plates, 7 = Fire door, 8 = Feed valve, 9 = Water level indicator,
10 = Pressure gauge

(including supercritical steam) can be obtained from this type of boiler. Power stations operating this type of boiler generate approximately half of the world's electricity. Water-tube boilers are used almost exclusively in the generation of electricity due to the high temperatures (and therefore efficiencies) which can be obtained.

Smaller units, including mobile packaged versions, are also available and are more frequently found where the regulatory environment is not conducive to the use of MTHS boilers (see next section). These smaller boilers are generally used to generate steam or hot water for industrial process applications rather than heating. As a general rule, water-tube boilers have higher capital costs than other boiler types. However this is only a consideration at lower outputs and pressures where they compete with MTHS boilers.

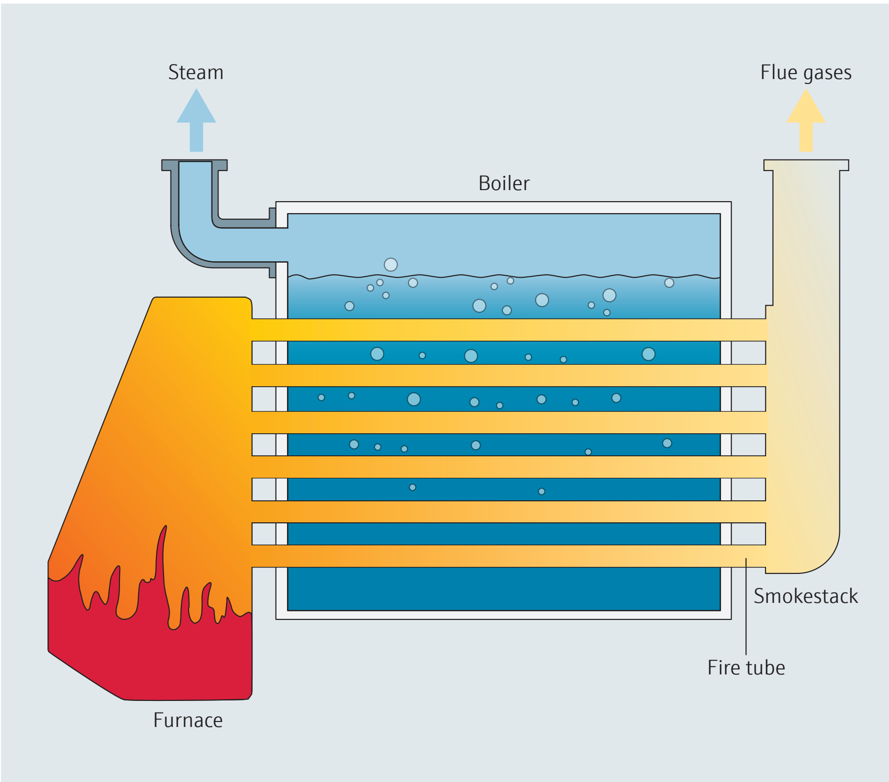


Figure 7: Principle of a fire-tube boiler.

MTHS boilers (fire-tube or smoke-tube boilers)

The modern MTHS boiler (Horizontal Multi-tube Steam Boiler, Figure 7) can trace its lineage directly back to Trevithick's Cornish boiler of the early nineteenth century via the Cornish, Lancashire, Scotch Marine and Economic boilers before arriving at the modern packaged boiler which was first introduced in the twentieth century in the United States of America. As the name suggests, products of combustion rather than water or steam flow inside the tubes.

There are a multiplicity of MTHS boiler designs, and the efficiency of a boiler is determined by the arrangement and sizing of the convective heat transfer tube passes, on the general principle that the greater the convective heat transfer surface area the greater the efficiency, albeit at greater cost and complexity. Two common designs, the three-pass wetback (Figure 9) and the reverse flame MTHS boilers (Figure 10) will be considered in more detail below. Figure 8 shows a modern three-pass wetback MTHS combination boiler.

Figure 8: Byworth Yorkshireman series combination steam boiler (Courtesy of Byworth Boilers).

MTHS boilers are commonly found working at pressures between 6 and 18 barg although units rated at up to 32 barg are available from a limited number of manufacturers. MTHS boilers supply saturated steam. Infrequently some superheat can be generated with the inclusion of an external super-heater, often mounted in the front smokebox or the furnace reversal chamber.

MTHS boilers are often compared using a rating known as from and at 100 °C, e.g. a 1000 kg/hr F&A 100 °C. This is not a measure of steam output but rather of the evaporative capacity of the boiler. This is equivalent to net energy release capability of the boiler: 1 kg/hr being equivalent to a net heat release rate of 2257 kJ/hr (627 W). This measure originated in the nineteenth century and is still used to compare boiler ratings today.

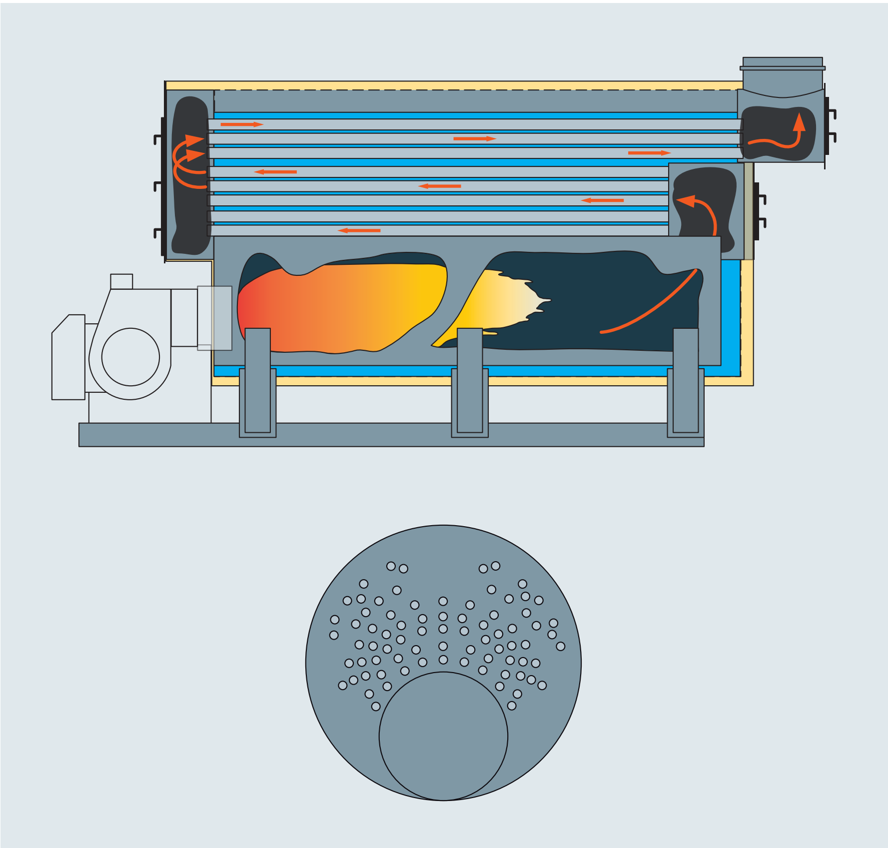


Figure 9: Cutaway of a three-pass wetback MTHS boiler.

Three-pass wetback MTHS boiler general arrangement

This type of boiler utilizes a cylindrical pressure vessel which serves to constrain the water, both physically and thermally (Figure 9). All of the directly heated surfaces are contained within this pressure vessel and, crucially, must be submerged at all times when the boiler is operating in order to prevent them from overheating. The uppermost portion inside the pressure vessel contains steam and is known as the steam space.

Combustion takes place inside the furnace which can be a plain cylinder in very small boilers. The furnace in larger boilers however, needs to be able to accommodate thermal expansion and in order to do so is either fully or partially corrugated or contains a number of expansion joints known as bowling hoops. Combustion should be completed inside the furnace and approximately 50% of heat transfer takes place there. Following on from the furnace are two banks of convective tubes which extract around 30% additional heat. These are named convective passes as the predominant form of heat transfer is convective. In order to minimize the size of the boiler, the tubes are arranged in banks, the gas flow being reversed at the end of the furnace and each tube bank.

Reverse flame MTHS boiler

The reverse flame MTHS boiler (Figure 10) differs from the three pass wetback in that the furnace is blank ended: the flame travels down its centre and the hot prod-

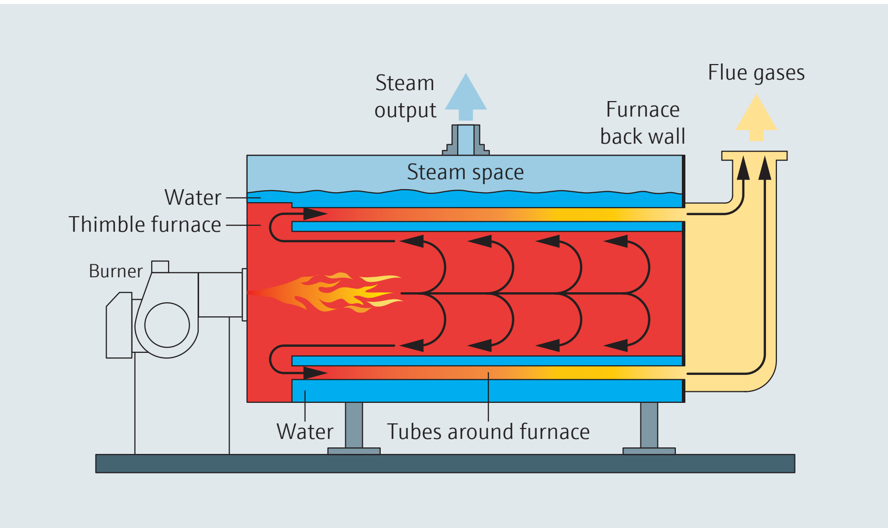


Figure 10: Cutaway of a reverse flame MTHS boiler.

ucts of combustion are deflected back along the walls to exit at the burner end. This type of boiler is restricted to the smaller end of the market, generally with a capacity of less than 2 MW. They are less efficient than three pass wetback MTHS boilers but are cheaper to manufacture.

Reverse flame boilers usually have only one convective pass. The heat transfer within this convective pass is enhanced using turbulators, also known as retarders. Turbulators are metal springs or twisted/formed metal strips which increase the heat transfer rate albeit at the expense of additional pressure drop.

Important shell design features for both types of boiler are:

- steam space volume,
- height between the liquid water surface and the crown valve,
- liquid water surface area,
- volume of water within the shell.

If the first three are too small then the boiler will have a propensity to discharge wet steam, carryover, and prime. If the volume of water in the shell is too small, maintaining the correct water level can be more difficult and thermal shock (due to the operation of the feedwater pump) will be more likely. If the volume is unnecessarily large, the boiler will be oversized, and more expensive to construct and operate.

The products of combustion, having passed through the shell are vented to atmosphere through a chimney. Installations firing natural gas may first pass the gases through an economizer, a gas to liquid indirect heat exchanger, used to pre-heat the boiler feedwater before it enters the shell. Standard economizers are not preferred for use with fuels containing sulphur as they operate at pressure vessels and their operation can cause the exhaust gas temperature to fall below the acid dewpoint of sulphuric acid.

Basic overview of boiler controls with reference to EN 12953

Steam is exported from the boiler solely by virtue of a pressure difference between the boiler and point of requirement. The rate of steam mass flow is thus determined by the process calling for the steam. The quality of the steam, e.g. pressure, wetness and cleanliness are affected by the rate of demand from the process, rate of change of demand, design and operating characteristics of the boiler supplying the steam and the condition of the distribution and condensate return systems.

Pressure controls

Boilers manufactured in accordance with EN 12953, a commonly used standard in Europe, are fitted with a pressure control which regulates the burner during normal operation. A separate high pressure limiter is also fitted which, when deployed, forces the burner to shut down and lock out requiring manual reset. For new boilers the limiter must be of a type which fails safe and complies with part 9 of EN 12953. As a further control measure, steam boilers are also fitted with a independent safety valve which acts in the event of failure of both the pressure control and limit devices.

Level controls

It is crucial to maintain a sufficient quantity of water within a boiler in order to prevent the heated surfaces becoming exposed to steam when the burner is firing (a dangerous low-water condition). A dangerous low-water condition places the boiler pressure vessel at risk of catastrophic failure. If the boiler is manufactured in accordance with EN 12953, the level controls comprise of a control unit for the pump and two independent level limiters. The level controller regulates the feedwater pump. In the event of some failure of the controller, the two limiters are independently connected to the burner to force its shutdown and lock out requiring manual reset. These controls are required to be fail safe, high integrity and self monitoring. An additional high-water alarm and cut out may also be provided.

Flame failure device

Flame failure devices (also known as photo-cells or magic eyes) are used to prevent dangerous occurrences whereby fuel continues to enter the furnace in the event that the flame unexpectedly goes out during normal operation. Modern devices detect a specific frequency within the electro-magnetic spectrum which is emitted by a flame. When activated they force a shut down and lock out of the burner requiring manual reset. As with level limiters they should be fail safe, high integrity and self monitoring.

Water treatment

Proper control of boiler water composition is of vital importance. Incorrect water treatment and management may result in the following problems:

- False positive water level indication increasing the risk of a potentially dangerous low-water incident, e.g. due to foaming with certain types of level control.
- Scale build up inside the boiler reducing efficiency with the potential for overheating of the heat transfer surfaces which can lead to catastrophic failure.

- Internal corrosion of the boiler shell and heating surfaces again with attendant risk of failure of the pressure vessel.
- Carryover of dissolved solids into the steam system.
- Increased risk of priming.
- A build up of sludge causing overheating of the metal surfaces with the possibility of cracks forming at tube ligaments or even furnace collapse.
  Note: A ligament is the joint at which a tube, or the furnace, is connected to the end plate of the boiler.

Care must also be taken to prevent steam being drawn into the boiler through the crown valve when the boiler is cooling following a shut down. Although a non-return valve should be incorporated into the system and is mandatory under common design codes, there have been instances where contaminants such as acid have been drawn into a boiler, due to a fault in the steam distribution system, causing severe internal corrosion.

Amongst other things, raw make-up water can contain many impurities such as dissolved organic and inorganic compounds, particulates and dissolved gases. It is not "pure" in the chemical sense of the term. The more important impurities with respect to steam boilers which are present in water are:

- Salts of calcium and magnesium, which are responsible for hardness in water
- Chlorides

External treatment method	Purpose
Filtration	The removal of organic and inorganic solid particles
Base exchange softening	To convert calcium and magnesium salts, by ion exchange, to sodium salts. This removes hardness which cause the formation of hard scale.
Reverse osmosis (RO)	To reduce total dissolved solids (TDS) and silica by membrane filtration.
De-alkalization	To reduce the total dissolved solids (TDS) by the removal of alkalinity.
Demineralization	To reduce/remove total dissolved solids (TDS) and silica by ion exchange.

Table 1: Common water treatment systems.

- Ammonium salts and their products of oxidation, nitrites and nitrates
- Traces of heavy metals such as lead or copper
- Dissolved gases, such as oxygen and carbon dioxide

The water treatment process for a steam boiler is usually in two stages – external or internal treatment:

External treatment is used to condition the feedwater supply to the boiler for a variety of reasons, one such being softening to remove compounds which form hard scale. De-aeration, primarily the removal of oxygen, is also very important and is achieved by a combination of thermal and chemical conditioning of the feedwater. The feedwater is further conditioned by the addition of various compounds such as alkalinity builders to prevent corrosion, and sludge conditioners. Table 1 gives a brief description of the more common primary water treatment systems in use.

Internal treatment: As steam boilers export relatively pure steam, the various compounds in the water and chemicals added to it concentrate inside the boiler with time either in solution or suspension. Too great a concentration of total dissolved solids (TDS) will cause foaming, carryover and, depending upon the water level control system may cause false positive level conditions. The level of dissolved solids is controlled by automatic or manual blowdown in conjunction with appropriate testing of the TDS.

Some of the solids in the boiler deposit as sludge at the bottom of the shell. This is removed by a bottom blowdown valve normally operated once a day. Failure to bottom blowdown the boiler may cause the bottom of the furnace to become filled with sludge, which will then build up and insulate the heating surfaces which may cause localized damage or more seriously a furnace collapse.

Deposits on the steel surfaces inside the boiler may occur due to defective water treatment or to some form of contamination of the returned condensate. Control of these deposits is of high importance in order to maintain the efficiency of the heat exchange surfaces within the boiler and reduce the risk of a dangerous occurrence. A layer of scale on the heating surfaces of a boiler, for example, would serve to insulate them as the thermal conductivity of scale is about one hundredth of that of steel. This would cause metal temperature of the heated surfaces of the pressure vessel to be higher than intended. This more commonly reduces the efficiency of the boiler and may require expensive remediation work. In extreme cases, however, the scale can be so thick that the metal temperature of the furnace in particular increases to the point where it becomes plastic and it collapses due to the external waterside pressure.

Contamination of condensate

Condensate is returned to the boiler house for re-use because it should be water of high purity with a significant heat content. However it is possible for condensate to become contaminated with undesirable consequences. There are two frequently encountered scenarios:

Scenario 1: The contamination is known and accepted. An example from industry could be the recovery and re-use of condensate from steam ejectors used to evaporate milk. The “evaporate” from the milk mixes with the prime steam and is returned to the boiler house in the condensate. This condensate will contain some contamination such as fats from the milk. This is accepted due to the high monetary value of the condensate. However, there is consequently a risk of foaming inside the boiler due to the presence of the fats in the condensate. Dependent on the exact conditions (contaminant and controls) there may be a contingent risk that the level controls are fooled into measuring the foam level rather than the true water level resulting in a low-water event. In this scenario selection of the most appropriate level controls and close control of the total dissolved solids (TDS) inside the boiler are of critical importance.

Scenario 2: Contamination due to some failure within the steam plant. An example could be a heat exchanger used for cleaning in place (CIP). On one side of the heat exchanger is caustic soda and steam on the other. If the heat exchanger element fails, caustic soda could be drawn into the boiler through the condensate return system. This again may have undesirable consequences for the pH level and certain types of level controls. Good practice and certain regulatory regimes advise that the risks due to contaminated condensate should be assessed and if necessary appropriate control measures introduced. These could influence the selection of the boiler controls and also monitoring and control of condensate for contamination.

Level controls

Correctly functioning level controls are of critical importance to the safe operation of a MTHS boiler. Low-water incidents due to demand related causes, such as priming, are relatively frequent in industry. Whilst they rarely pose a danger to the boiler itself, they may result in the burner being shut down requiring manual reset with a potential consequent loss of production. Of greater significance is a complete failure of the level controls which can cause a furnace collapse or even a catastrophic failure (explosion). Dangerous occurrences and the much less common catastrophic failure due to a low-water level event can have multiple causes. Such causes can be fairly simple such as faulty repair, or more complex, such as equipment failure causing contamination in the process, which in turn defeats the controls.

There are a number of different commonly used level control types, each of which has its own strengths and weaknesses. One of the most common level control systems in Europe comprises fail safe, high integrity self-monitoring conductance and capacitance arrangements. With these types of control the boiler shell forms part of the electrical circuit. These have largely displaced float type controls housed in external chambers which previously dominated the market. Developments in level controls and limiters have meant that other principles are becoming more common in the boiler industry. Examples are internal displacement floats, guided radar and vibrating fork. The precise type of controls used on a given site should be determined by the end user at specification through suitable risk analysis and be capable of defending against all the identified threats.

Feedwater pump arrangements

On-Off controls

As the name implies the water pump is either running at maximum capacity or is switched off. These are generally only fitted to small boilers due to problems associated with thermal shock in larger boilers. They are not recommended for use where large fluctuations in load may be experienced. The injection of a large quantity of relatively cold water in combination with a sudden demand change may cause the steam bubbles under the surface of the water to collapse, lowering the overall level and triggering a low-water cut-out of the burner. They must not be used in conjunction with economizers.

Modulating controls

There are several methods of modulating the feedwater flow rate to try and match the steam output. The simplest method is to fit a variable speed drive to the electric motor of the feedwater pump. This is however frequently limited in the amount of turndown that can be provided due to the types of pump normally selected. An increase in turndown is often achieved by the inclusion of a control valve into the system which regulates the flow. If an economizer is fitted to the boiler, a spillback to the hotwell or deaerator may be required.

A further improvement to the feedwater management can be obtained through the use of three-element control. In addition to the level controls, inputs may be taken from feed water meters, steam meters and pressure transmitters. Three element systems attempt to anticipate the water demand and result in a more even flow of feedwater to the boiler, reduce thermal shock and improve the response of the boiler to changes in demand.

Burners

MTHS boilers can be designed to fire almost any solid, liquid or gaseous fuel with an economically extractable calorific value. Common fuels are natural gas, diesel and light fuel oil. Heavier grades of fuel oil are more commonly found in more remote installations especially those not connected into a natural gas supply system. Heavy fuel oil plants are associated with higher capital costs, handling difficulties and more intensive operational requirements and are becoming less common in Europe due to increasing regulatory demands in relation to pollution. Coal, refuse/waste-derived fuels, bio fuels and alternative grades of gaseous and liquid hydrocarbon fuels are also used, generating important niche fuel markets for the boiler industry.

MTHS boilers are usually fitted with either single or dual-fuel burners, whilst burners with a multi-fuel capability are available they are much less common. Depending on the specification, very close control and highly consistent operation may be achieved by the incorporation of fuel-air-mapping systems coupled with servo-motor controlled modulating damper systems and oxygen trim with a variable speed drive on the motor. These can offer turndown ratios of up to 10:1 on natural gas, and 6:1 on lighter grades of oil whilst being able to consistently maintain excess oxygen levels below 15%.

Care should be taken to match a burner to a given boiler furnace. A correctly matched burner–furnace combination will deliver improved performance and economy whilst minimising the emissions of unwanted pollutants. This can be especially important for burners designed for very low emissions, primarily oxides of nitrogen, for which design considerations such as the furnace shape, burner head design, flame shape, air/fuel staging and internal re-circulation are balanced with achievable minimum excess air, carbon monoxide levels and flame stability throughout the turndown range.

In its simplest form the burner firing rate is a function of the boiler pressure only. If we consider a boiler which is at its upper set point, maximum working pressure, with the crown valve closed then the burner will not be firing. If the crown valve is opened and steam exported the pressure will start to fall inside the boiler. When this pressure falls to a pre-determined set point the burner will commence its ignition sequence and fire up. The burner will then remain firing until it reaches maximum working pressure at which it stops firing. If the pressure inside the boiler again falls the burner will re-start automatically and the process will keep repeating itself.

If the burner is capable of modulation, it will try to maintain a set point just under the upper set point so that it remains firing continuously unless the steam demand is so low that it is below the burner's minimum turndown thus forcing the pressure to reach maximum working pressure which turns the burner off.

If the steam demand is so great that it exceeds the burner's ability to add heat and thus maintain pressure, the burner will keep firing at its maximum firing rate but the pressure will not be able to recover and will fall. The pressure will keep falling until the rate of heat removed in the steam equals that added by the burner. This means that lower pressure steam (lower than is required) is distributed with potential consequences which can include steam starvation to the end processes, excessive steam bulk velocity and carry-over of liquids and solids.
The principal safety features associated with burners are:

- Pre-fire checks such as proving the valves on the gas train during the start up sequence
- Air proving which checks that the volume of air passing through the burner at a set fan speed and damper setting is correct.
- A pre-fire purge during which air is blown through the boiler removing any combustible gases
- A flame detector which, in the event of a flame out event, shuts the burner down requiring manual reset (a lock-out)
- Fail-safe slam shut valves on the fuel lines which deploy in the event of a dangerous occurrence such as a fire
- In addition, the waterside limiters such as pressure and level will, when deployed, force the burner to shut down locking out the boiler requiring a manual reset.

Basic steam distribution

Steam moves from a high pressure location to a lower pressure location. In doing so it uses some of its own energy which is seen as pressure drop. Additionally, heat losses during distribution causes a fraction of the steam to condense to form condensate. This is fundamental to understanding the operation of steam systems and how they are designed.

As steam, by its nature, is generated at elevated pressure and temperature, the whole system has to be rated for the maximum design pressure and temperature. This is normally achieved by designing or selecting parts and equipment that conform to published design standards.

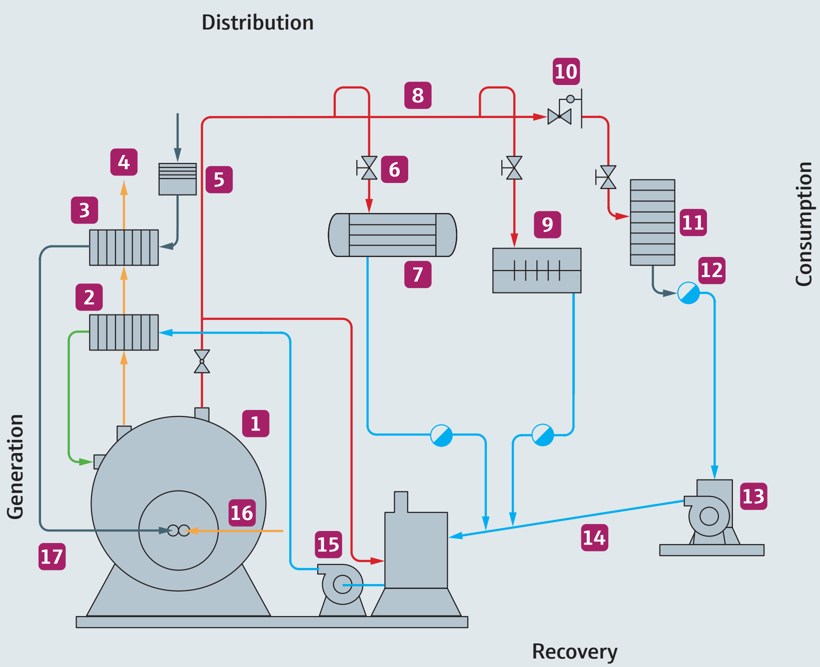
A further very important consideration is the design bulk line velocity of the steam which is limited to prevent unduly fast erosion, for steam is a very aggressive fluid. As a "rule of thumb" steam bulk velocities are normally limited for saturated/wet steam to between 25 and 40 m/s for short lines and 15 to 20 m/s for longer lines; the precise limiting velocity being chosen by the individual design engineer. Once the limiting velocity is selected the line sizes can be calculated based on the steam pressure (and hence its density) and the required mass flow rate for the application.

The condensate which forms as a result of primarily heat loss and secondarily the motion of the steam must be removed. This is achieved by using a steam trap – a device to separate liquid water from steam. Within the distribution system, steam traps are fitted at regular intervals in so-called "dirt-legs" – a vertical pipe leg which allows the denser condensate to collect at the bottom from which it is removed by the trap. To aid removal of condensate, steam lines should always be fitted at a slight downward angle in the direction of flow. As a rule the minimum fall should be 1:100. In order to minimize the rate of condensate formation, steam lines should always be insulated. Insulation of steam lines will also reduce the surface touch-temperature to a point where it will not cause burns to skin on contact.

There are a number of items of basic line equipment which are common to all steam systems (Figure 11). The following list includes only the most basic types of line equipment:

Safety valves

These are valves which discharge to atmosphere in the event that they encounter a pressure higher than their set pressure. They are very important as they prevent overpressure either on vessels such as steam boilers, pressure vessels or downstream of pressure reducing valves.



- 1 Boiler
- 2 Economizer
- 3 Combustion air preheater
- 4 Combustion gases
- 5 Forced draft fan
- 6 Isolation valve
- 7 Shell and tube heat exchanger
- 8 Steam system header
- 9 Process heater
- 10 Pressure reducing valve or back pressure turbine
- 11 Process heater
- 12 Steam trap
- 13 Condensate receiver tank and pump
- 14 Condensate piping
- 15 Feedwater heater pump
- 16 Fuel
- 17 Combustion air

Figure 11: Components of a steam generation and distribution system.

Isolation valves
These are simply valves which permit or deny flow. They can be either manual or automatic in operation and are used extensively in steam plants. During normal operation they should always be either fully open or fully closed. If a system is being warmed up, steam valves should be fractionally opened to permit a small amount of steam through so that the lines are heated slowly and gently as this reduces the risk of water hammer events. Valves which are fractionally opened for this purpose should always be under supervision. Isolation valves are vital when plant is being isolated for work on the steam system. The selection of appropriate systems of plant isolation is extremely important and care should be taken to ensure that an isolation system used for a particular job is adequate. Under no circumstances is it appropriate to work on a steam system using a single closed isolation valve.

Control valves
These are valves which regulate the flow of steam. They may be manual or automatic in operation, although in industrial settings the vast majority are automatic.
Note: valves designed or installed only for isolation should not be used for control purposes.

Pressure reducing valves (PRV)
Steam moves by giving up some of its own energy. Consequently it is almost impossible to design a system with a known, controllable pressure at the point of consumption unless pressure reducing valves are incorporated. Pressure reducing valves are used to reduce the pressure to that required close to the point of consumption. Pressure gauges and safety valves should always be fitted downstream of them. The complete set of valves and other equipment associated with a PRV are frequently referred to as a pressure reducing set or station.

Pressure and temperature gauges
These are found in abundance on steam systems so that the plant operators know the condition of the steam. They may also be used for control purposes, e.g. pressure gauges downstream of certain types of PRVs are used in the feedback loop.
Note: temperature gauges are normally only used on superheated steam systems.

Non-return valves/check valves
Steam and condensate should only flow in one direction – that intended by the designer. These valves are used to ensure that this is so. They are important pieces of equipment as they can prevent unwanted or potentially dangerous occurrences.

Strainers

Steam is aggressive and erodes steam pipes. Strainers are used to filter out any debris that may be contained in the system. They are frequently found located in front of control and reducing valves. They should always be included in trap sets.

Steam traps

As steam moves through a system some of it condenses into liquid water. This liquid water must be removed for a variety of reasons. This is done using steam traps, devices which are capable of separating liquid water from steam and discharging it into a condensate line for return to the boiler house or disposal. They are also fitted to all indirect consumers of steam. Their correct operation is vital to the functioning of a steam system. Traps are normally fitted in conjunction with other equipment such as sight-glasses, strainers, check and isolation valves. Collectively these are known as “trap sets”. It is very important that steam traps are inspected on a regular basis. It is estimated that 15% fail every year, and there are a number of undesirable consequences, both economic and safety, which result from failed traps.

How steam moves (simple explanation)

In the following text the results of calculations are merely stated. The derivations of these calculations can be found on Pages 41 ff.

Suppose we have dry saturated steam held in a reservoir at 10 bara and that this reservoir can be expanded (Figure 13). When the reservoir is expanded – using external work –, the steam also expands and work is performed by the steam in its expansion. As this is a closed system, i.e. there is no energy transferred into or out of the steam, the energy required for the expansion of the steam must come from the steam itself. When the reservoir expands, the pressure decreases, and for this example we shall assume a final pressure of 9 bara. The total enthalpy of the steam at the initial condition is the same as at the final condition. The total enthalpy of the dry saturated steam at 10 bara is 2777 kJ/kg, and its temperature is 179.9 °C (from steam tables). As there is no change in energy, the steam – once the pressure has fallen – still contains the same amount of energy as it had at the start. However, dry saturated steam at 9 bara has a saturation temperature of 175.4 °C and contains 2774 kJ/kg, 4 kJ/kg less than at 10 bara. This difference shows itself as superheat, i.e. the temperature rises above the saturation temperature at 9 bara and 175.4 °C. This temperature rise can be calculated to be 1.5 K of superheat for this situation. This gives a final steam temperature of 175.4 °C plus 1.5 K which equals 176.9 °C.

Why does this happen?

Work and heat are different forms of the same thing: namely energy! If the reservoir is expanded, the steam inside the system must expand with the physical boundary in order to maintain the equilibrium of the reservoir. For the steam to expand, work must be performed, and the energy required for this work must come from the steam itself as it is in a closed system.

Initially the energy of the system can be thought of as potential energy, observed physically through the pressure and temperature of the steam. Since, initially the system is in equilibrium, the kinetic energy of the system is zero. When the system boundary is physically moved, the steam also moves, and some of this potential energy is converted into kinetic energy. However, when the system is again at rest there is no motion and the kinetic energy returns to zero. The energy that was converted into kinetic energy has to be converted into another form of energy. In this case, heat and we observe this heat as superheat. This energy conversion process is: potential energy to kinetic energy to heat energy.

If we consider dry saturated steam initially at 10 bara, and flowing along a pipe until the pressure had decayed to 9 bara, the frictional heating (seen as superheat) determined in the previous example (1.5 K) would be seen if the system was also closed, i.e. isenthalpic. However, in practice heat is lost across the pipe wall, and in reality the processes are not identical. Although frictional heating is still generated it is not seen as superheat due to the cooling effect of heat loss to atmosphere.
Note: Whilst the actual rate of heat loss to atmosphere from a steam pipe is difficult to calculate exactly, we can estimate it quite well for known conditions.

We can illustrate what happens in a more realistic situation: If we calculate the pressure drop resulting from the passage of steam down a pipe, we can estimate the line length required for this pressure drop. For example, a pressure drop from 10 bara to 9 bara at 20 m/s in a DN 100 line will require 848 meters of pipe (making an assumption about the roughness of the wall) giving a frictional heating effect of about 4 W/m.

If we then calculate the heat loss for the above case for an insulated pipe, assuming benign conditions, i.e. still air, we can estimate the heat transfer to the atmosphere to be 126 W/m. This is very large when compared to the frictional heating effect of approximately 4 W/m and so the superheat is not seen. For uninsulated pipes, the heat loss is at least an order of magnitude greater than for insulated pipes.

As you can see, the temperature increase due to frictional heating is very small for conditions typically found in saturated steam systems. When expressed in energy terms, it can readily be seen that superheat generation is easily "masked" by the heat loss from the pipe wall which approximates to an order of magnitude greater than the rate of superheat generation for the insulated pipe case. Given that steam lines are extremely difficult to fully insulate, superheat solely due to the passage of steam through a pipe is therefore highly unlikely to be seen in practice.

So far we have neglected to consider the effect of heat loss through the pipe walls on the steam itself. Let us take our 10 bara steam travelling at 20 m/s down a DN 100 pipe: It requires 848 meter for the pressure to drop to 9 bara due to friction. However, over that length of 848 meter some 106 kW of heat would be lost from the steam to the atmosphere if the pipe were insulated and nearly 6.5% of the steam in the line would have condensed back into condensate (liquid water).

Although it is in practice not possible to observe superheat due to pipe friction losses alone in a steam line, superheat is frequently found in what are normally thought of as saturated steam systems. This is due to the presence of line equipment across which there is a large pressure drop. The most obvious and frequently occurring example of this is a pressure reducing valve (PRV).

There are a number of reasons why PRVs are used in industry. The one which concerns us here is the need to provide a stable and controllable steam system supply to each of multiple consumers from one overall generation source (boiler). The purpose of a PRV is to provide a stable steam pressure down stream of the valve over a range of mass flow rates and with a potentially variable upstream pressure.

Example

Let us suppose a typical situation (Figure 12): Upstream of a PRV the steam is saturated with a pressure of 9 bara at 175.4 °C and the downstream pressure is 4 bara. If we approximate a no loss condition across the PRV, the difference in specific enthalpy of the 9 bara saturated steam compared to the 4 bara saturated steam is so great that some 16 degrees of superheat are generated, and the 4 bara steam is superheated rather than saturated. *Note:* The final temperature of the 4 bara superheated steam is the saturation temperature of saturated steam at 4 bara (143.6 °C) plus the degrees of superheat (16 K) which gives a final steam temperature of 159.6 °C.

Thus, rather than the extremely small rate of superheat generation due to frictional line losses, we may now have a large and measureable amount of superheat immediately downstream of the PRV. It is very important to realize that this superheat is caused by the passage of the steam through the valve. If there is no flow then there is no superheat formation. If the flow rate is extremely low then the heat loss through the valve body would mask this effect to some degree. However, for typical steam bulk velocities, it is quite possible to observe this effect downstream of PRVs in industry simply by measuring the temperature and pressure of the superheated steam and then looking up these values in a set of steam tables.

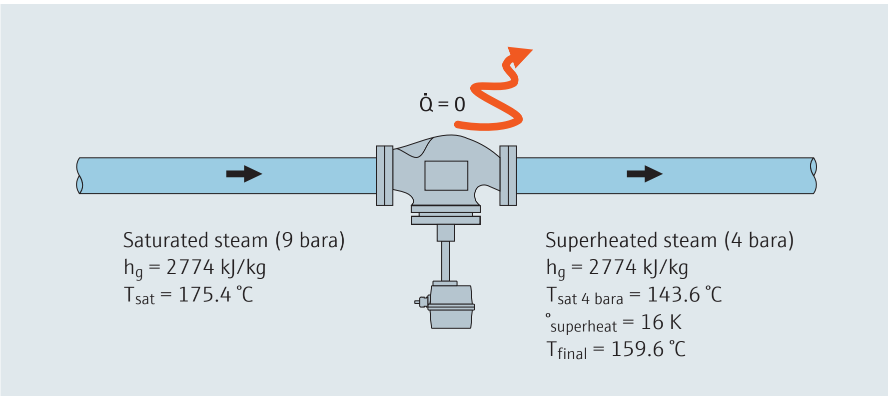


Figure 12: The passage of steam under isenthalpic conditions through a pressure reducing valve (PRV) for known conditions.

Approximation to no loss condition: For the case of 9 bara steam travelling at 20 m/s down a DN 100 line the flow of heat will be greater than 2 MW. Typical casing losses from a suitably sized PRV will be around 2 kW – around three orders of magnitude less than the flow of heat in the pipe. Therefore, the superheat will remain in the steam and be dissipated by heat loss as the steam travels further downstream.

How long will the superheat endure?

Superheat, if generated in significant quantities can persist for a significant distance. Even for an uninsulated line, it is possible in theory for there to be superheat a considerable distance from the PRV, for example, at 20 m/s for insulated pipelines up to 151 meters, and for uninsulated pipelines up to 18 meters.

In practice, as previously noted, steam lines are often not completely insulated and the superheat dissipation length will be shorter. Moreover, any condensate that was in the steam line upstream of the PRV will also have a significant effect, as it can pass through the valve and then a small amount will boil off to quickly dissipate the superheat. This does not happen instantaneously; it is possible for wet steam to pass through a PRV, and downstream for there to be superheated steam in the centre of the pipe with liquid water adhering to the walls. There will then be a mixing length effect before the superheat is dissipated by the line losses and boiling of the condensate.

All line equipment exhibit this effect as there is a much greater pressure reduction through line equipment compared to pipe friction losses. This superheat generation may be unanticipated. An example of a predictable superheat created by losses through equipment other than a PRV would be the bluff body of a vortex flowmeter or the orifice plate of a DP flowmeter.

An example of an unanticipated situation would be a partially opened isolation valve (note: these should be either fully open or fully closed): In this instance there is an unexpectedly high pressure drop across the valve which causes superheat to occur downstream. It is quite possible for this to happen, and for the steam system to operate effectively, because they are intentionally designed with an excess of pressure. Alternatively, a partially opened isolation valve may cause local starvation downstream as the pressure drop is too great to allow the required amount of steam through.

On the motion of steam (detailed explanation)

Let us suppose that 1 kg of dry saturated steam is held in a reservoir at 10 bara (Figure 13, a) and that this reservoir can be expanded using external work so that the steam is under adiabatic, isenthalpic conditions in a closed system with the boundary as the internal wall.
Note: The work required to move the physical boundary is provided from an external source, no work is performed on the steam itself and no heat is transferred to or from the steam.

When the reservoir is expanded, the steam also expands, and work is performed in expanding the steam (Figure 13, b). As this is a closed system, i.e. there is no heat transfer across the boundary and the process is adiabatic, the energy required for the expansion of the steam must come from the steam itself. When the reservoir expands, we see that the pressure decreases and for this example we shall assume a final pressure of 9 bara (Figure 13, c).

We can calculate the physical changes to the steam: the specific enthalpy of dry saturated steam vapor at 10 bara is 2777 kJ/kg and a saturation temperature of 179.9 °C. As there is no change in the total enthalpy of the steam, once the pressure has fallen to 9 bara it must still contain 2777 kJ/kg. However, dry saturated steam vapor at 9 bara has a specific enthalpy of 2774 kJ/kg and a saturation temperature of 175.4 °C. The difference between the specific enthalpy of the vapor at the two pressures – 4 kJ/kg – is exhibited as superheat, i.e. the temperature has risen above the saturation temperature at 9 bara 175.4 °C. This temperature rise may be calculated using the specific heat capacity of the substance according to Equation (1).

$$\delta h = m\ C_p\ \delta T \qquad (1)$$

Taking the specific heat capacity of the steam vapor as 2.7 kJ/kg K and the mass of steam as 1 kg – when Equation (1) is rearranged into Equation (2) –, the temperature rise is 1.5 K of superheat giving a final steam temperature of 175.4 °C + 1.5 K = 176.9 °C.

$$\delta T = \frac{\delta H}{(m\ C_p)} \Rightarrow \delta T = \frac{4}{(1 \times 2.7)} = 1.5\ \mathrm{K} \qquad (2)$$

Why does this happen?

For this fact we require some understanding of both the First and Second Laws of Thermodynamics. Taking the Second Law of Thermodynamics, it may be said that a corollary of this law is that all thermodynamic systems tend towards equilibrium.

Consider our system with the previously described 1 kg of dry saturated steam at 10 bara (Figure 13, a). It is internally at rest and no heat or work is crossing the thermodynamic boundary, ergo it may be said to be in equilibrium. If the reservoir is expanded by doing work external to the system, then, a void would be created if the steam did not expand and the system would be out of equilibrium as there would be two volumes each with a different pressure but no boundary between them (Figure 13, b). Thus the steam inside the system must expand with the physical boundary in order to maintain the equilibrium of the internal system (Figure 13, c).

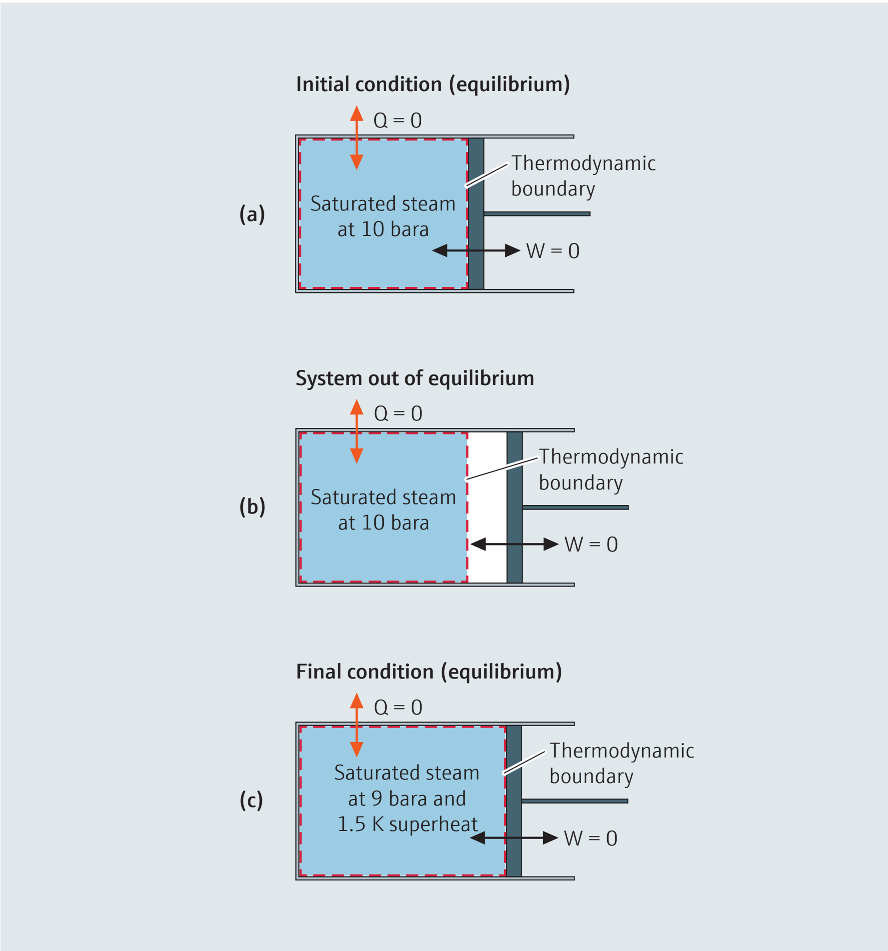


Figure 13: Thermodynamic state of a system – equilibrium during expansion.

From the First Law of Thermodynamics, we can see that work and heat are different forms of the same entity: namely energy. Energy cannot be created nor destroyed, merely changed from one form to another. Also, for the steam to expand in accordance with the First Law of Thermodynamics work must be performed and, as noted previously, the energy required for this work must come from the steam itself.

Initially for the condition shown in Figure 13 (a), we may consider that the energy of the system is in the form of potential energy, observed physically through the pressure and temperature of the steam. As there is no motion the kinetic energy of the system is zero. When the system boundary is physically moved, the steam also moves and some of this potential energy is converted into kinetic energy. However, when the system is again at rest there is no motion and the kinetic energy returns to zero. In order to satisfy the First Law of Thermodynamics, the energy that was converted into kinetic energy is further converted into heat and it is this heat that is exhibited as superheat. This process, i.e. conversion of potential energy to kinetic energy, and then to heat energy, can be derived from consideration of the First Law of Thermodynamics and expressed in terms of Equation (3).

$$\Delta \mathrm{PE} = \Delta \mathrm{KE} = \delta \mathrm{H}_{\mathrm{superheat}} \quad \text{(idealized isenthalpic conditions)} \qquad (3)$$

If we consider saturated steam taken from a boiler or accumulator (a reservoir) which is maintained at 10 bara, and flowing along a pipe until the pressure has decayed to 9 bara, the above results would be also true if the situation was also isenthalpic, i.e. the final condition of the steam would be 9 bara steam with 1.5 K superheat. However, in reality, heat is lost across the system boundary (pipe wall) and the process is not isenthalpic.

We can use standard theory to illustrate what happens in a more realistic situation. *Note:* whilst the actual rate of heat loss to atmosphere from a steam pipe is difficult to calculate exactly, standard theory permits a good approximation to be estimated for known conditions.

The Darcy-Weisbach equation (Kreith 1972, p. 376), Equation (4), is used to estimate the pressure drop resulting from the passage of fluid down a pipe. This can be rearranged and used to estimate the line length (l), resulting in a corresponding pressure drop. Let us suppose that the overall pressure drop is 1 bar for a range of initial pressures of dry saturated steam and typical bulk velocities for a DN 100 bore steam line. For this calculation we shall require a friction factor. In this example this factor is calculated using an approximation, that of Petukhov (Petukhov 1970) modified to calculate a “British” friction factor, Equation (5). The resulting line lengths are shown in Table 2 (for saturated steam). The friction factors calculated for the

Petukhov correlation are valid for smooth bore pipes. For carbon steel pipes which have experienced corrosion and erosion the friction factor will be higher in practice. If applying this methodology to a "real world" situation, care should be taken over the selection of the appropriate friction factor for steam applications. Care should also be taken not to confuse the various correlations for the friction factor such as Petukhov, Fanning, Filonenko and derivations from a Moody chart with the form of the calculation for the pressure drop.

$$\delta \mathrm{p} = 4f \left(\frac{\mathrm{l}}{\mathrm{d}}\right) 0.5\, \rho\, \mathrm{u}^2 \qquad (4)$$

$$f = 0.25\, (1.82\, \log_{10} \mathrm{Re} - 1.64)^{-2} \qquad (5)$$

For a fixed line size, the bulk velocity is the dominant factor with regard to the pipe length required for a fixed pressure drop due to its exponentiation. The thermo-physical properties of steam also have a significant effect on both the density and friction factor.

Once the line lengths are known, we can then calculate the average rates of superheat generation per 100 meters, and for the same range of pressures and bulk velocities for isenthalpic conditions (Table 3 and Table 4). Table 4 presents the average enthalpy available for the generation of superheat expressed as Watt per meter [W/m] for the same base data.

The rate of superheat generation can be compared to the average heat transfer rate per meter due to pipe losses. For these calculations the cases of uninsulated, benign and adverse conditions and insulated pipe (25 mm thickness) for benign conditions only are considered.

To calculate these heat transfer rates, standard theory was used in conjunction with correlations for the dimensionless numbers such as the Nusselt number. The heat loss for the insulated pipe was calculated according to the approximation of Stevens (Stevens, n.d.), Equation (6). For ease of calculation a fixed value of 5 W/m^2 K was used for the local heat transfer coefficient. The heat loss for the uninsulated pipe was calculated in accordance with Holman (Holman 1981, pp. 272–274), Equations (7) and (8) for both benign and adverse conditions. Benign conditions assumed natural convection and radiation losses only, whilst for adverse conditions a constant wind speed of 20 m/s was assumed, all other factors held constant.

Initial pressure [bara]	Final pressure [bara]	Pipe length [m] required for a 1 bar pressure drop Average bulk velocity				
		10 m/s	15 m/s	20 m/s	25 m/s	30 m/s
3	2	8408	4062	2419	1617	1162
4	3	6503	3134	1864	1244	894
5	4	5360	2579	1532	1022	733
6	5	4589	2205	1309	872	626
7	6	4029	1934	1147	764	548
8	7	3602	1727	1024	682	489
9	8	3264	1564	927	617	442
10	9	2989	1432	848	564	404
11	10	2761	1321	782	520	373
12	11	2567	1228	727	483	346
13	12	2401	1148	679	452	323
14	13	2258	1079	638	424	304
15	14	2131	1018	602	400	286
16	15	2018	964	570	379	271
17	16	1919	916	541	360	257
18	17	1829	873	516	343	245
19	18	1748	834	493	327	234
20	19	1673	798	471	313	224

Table 2: Line lengths [m] required for a pressure drop of 1 bar (DN 100 pipe).

Initial pressure [bara]	Final pressure [bara]	DN 100 pipe superheat/100 m line [K/100 m] Average bulk velocity				
		10 m/s	15 m/s	20 m/s	25 m/s	30 m/s
3	2	0.104	0.216	0.363	0.543	0.755
4	3	0.092	0.190	0.320	0.479	0.667
5	4	0.082	0.170	0.287	0.430	0.599
6	5	0.074	0.154	0.260	0.390	0.544
7	6	0.068	0.141	0.237	0.356	0.497
8	7	0.062	0.129	0.218	0.327	0.456
9	8	0.057	0.118	0.200	0.300	0.419
10	9	0.052	0.109	0.184	0.277	0.387
11	10	0.048	0.101	0.170	0.255	0.356
12	11	0.044	0.093	0.157	0.236	0.329
13	12	0.041	0.085	0.144	0.217	0.303
14	13	0.037	0.077	0.130	0.196	0.274
15	14	0.034	0.072	0.122	0.183	0.256
16	15	0.032	0.066	0.112	0.168	0.235
17	16	0.029	0.060	0.102	0.154	0.215
18	17	0.039	0.082	0.139	0.209	0.292
19	18	0.024	0.050	0.084	0.127	0.178
20	19	0.021	0.045	0.076	0.114	0.160

Table 3: Rate of superheat generation for typical average bulk velocities, idealized isenthalpic conditions (DN 100 pipe).

$$\dot{Q}_m = \frac{2\,\pi\,(T_{steam} - T_{air})}{\left[\frac{1}{k}\ln\left(\frac{r+t}{r}\right) + \frac{1}{\alpha\,(r+t)}\right]} \qquad (6)$$

r = steam pipe outer radius
t = insulation thickness
k = thermal conductivity of the insulation
α = local heat transfer coefficient over the insulation

$$\overline{Nu_f} = C\left(Gr_f\ Pr_f\right)^m \qquad (7)$$

$$\dot{q} = \alpha\ A\ \delta T_{LM} \qquad (8)$$

As may be observed from Table 3, the fraction of the initial enthalpy of the steam which is converted to superheat due to its motion results in the generation of very small amounts of superheat. When rate of superheat generation is expressed in energy terms and compared to the rate of heat loss for various conditions (Table 4), it can readily be seen that superheat generation is easily "masked" by the heat loss from the pipe wall. Given that steam lines are extremely difficult to fully insulate, superheat due to the passage of steam along a pipe is very unlikely to be seen outside of a laboratory.

Although, in practice it is not possible to observe superheat caused by friction losses due to the passage of steam down a pipe alone. Superheat is frequently found in what are normally thought of as saturated steam systems. This is due to the presence of line equipment across which there is a large pressure drop; the obvious example being a pressure reducing valve (PRV).

Example

Let us suppose that we have a typical situation. Upstream of a PRV the pressure is 9 bara and downstream it is 4 bara (Figure 14). Moreover we shall assume an isenthalpic or "no loss" condition across the PRV. The specific enthalpy of 9 bara saturated steam is 2773 kJ/kg, however, the specific enthalpy of 4 bara saturated steam is 2738 kJ/kg – a difference of 35 kJ/kg. It is this 35 kJ/kg that is available to superheat the 4 bara steam. Again, we can calculate the amount of superheat generated to be 15.3 K from Equation (9).

$$\delta T = \frac{\delta H}{\left(m\ C_p\right)} \Rightarrow \delta T = \frac{35}{(1 \times 2.29)} = 15.3\ K \qquad (9)$$

Initial pressure [bara]	Final pressure [bara]	DN 100 pipe: enthalpy conversion [W/m] Average bulk velocity				
		10 m/s	15 m/s	20 m/s	25 m/s	30 m/s
3	2	0.26	0.80	1.79	3.4	5.6
4	3	0.32	1.01	2.25	4.2	7.1
5	4	0.38	1.18	2.65	5.0	8.3
6	5	0.43	1.33	3.00	5.6	9.4
7	6	0.47	1.47	3.29	6.2	10.3
8	7	0.50	1.58	3.55	6.7	11.2
9	8	0.53	1.67	3.77	7.1	11.8
10	9	0.56	1.75	3.95	7.4	12.4
11	10	0.58	1.82	4.09	7.7	12.9
12	11	0.59	1.86	4.20	7.9	13.2
13	12	0.60	1.90	4.27	8.0	13.5
14	13	0.61	1.91	4.31	8.1	13.6
15	14	0.61	1.92	4.32	8.1	13.6
16	15	0.61	1.91	4.30	8.1	13.6
17	16	0.60	1.88	4.25	8.0	13.4
18	17	0.59	1.84	4.16	7.8	13.1
19	18	0.57	1.79	4.04	7.6	12.8
20	19	0.55	1.72	3.89	7.3	12.3

Table 4: Rate of superheat generation expressed in W/m compared to casing losses (DN 100 pipe).

Initial pressure [bara]	Final pressure [bara]	Casing losses [W/m]		
		25 mm insulation "benign"	Uninsulated "benign"	Uninsulated "adverse"
3	2	75	626	2642
4	3	88	719	2928
5	4	98	796	3155
6	5	105	863	3347
7	6	112	923	3513
8	7	117	978	3660
9	8	122	1028	3794
10	9	126	1075	3916
11	10	130	1118	4029
12	11	134	1160	4134
13	12	137	1199	4233
14	13	140	1237	4326
15	14	143	1273	4414
16	15	146	1308	4498
17	16	148	1341	4578
18	17	151	1374	4655
19	18	153	1405	4728
20	19	155	1435	4799

(Table 4 continued)

The final temperature of the 4 bara superheated steam is the saturation temperature of saturated steam at 4 bara (143.6 °C) plus the degrees of superheat (16 K) which gives a final steam temperature of 159.6 °C immediately downstream of the PRV.

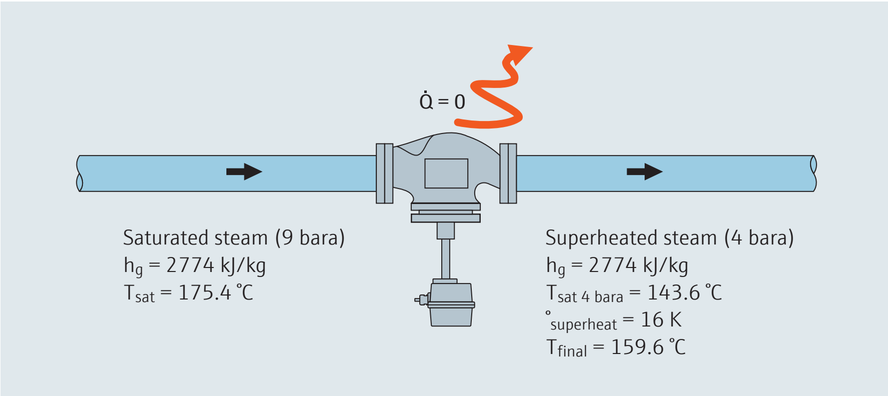


Figure 14: The passage of steam under isenthalpic conditions through a pressure reducing valve (PRV) for known conditions.

Thus, rather than the extremely small rate of superheat generation due to frictional line losses, we now have a large and measureable amount of superheat immediately downstream of the PRV.

Returning to the assumption of isenthalpic conditions across the PRV – is this assumption reasonable? Let us suppose that for the above pressures and degrees of superheat a PRV is fitted on a DN 100 pipe which has an estimated surface area of 0.62 m^2 (derived from a 3-D model of an actual valve). As this PRV is not insulated we can estimate the rate of heat loss from the valve body to be around 1.8 kW, again using standard theory, of the form of the previous calculation utilising Equations (7) and (8).

Table 5 shows the total heat flow across the PRV and the heat flow of the superheat fraction downstream of the valve. As the heat loss across the PRV is small in comparison we can see that it has little effect on the amount of superheat immediately downstream of the valve. For the previously supposed isenthalpic case, the total superheat was calculated to be 15.3 K.

Line speed [m/s]	Steam mass flow rate [kg/s]	Total heat flow across the PRV [kW]	Heat flow of superheat fraction after the PRV [kW]	Heat loss across the PRV body [kW]	° Superheat immediately downstream of the PRV [K]
5	0.19	537.8	6.8	1.8	11.2
10	0.39	1075.6	13.6		13.3
15	0.58	1613.3	20.4		13.9
20	0.78	2151.1	27.1		14.3
25	0.97	2688.9	33.9		14.5
30	1.16	3226.7	40.7		14.6

Table 5: Superheat downstream of PRV accounting for casing losses.

How long will the superheat endure? Having determined the heat flux of the superheat fraction, this is simply divided by the line losses (averaged between 4 and 9 bara). The results are shown in Table 6 and demonstrate that superheat, if generated in significant quantities, will endure for a significant distance. Even for an uninsulated line, it is possible in theory for there to be superheat a some distance downstream of the PRV.

In practice, as previously noted, steam lines are often not completely insulated and the superheat dissipation length will be shorter. Moreover, any condensate that was in the steam line upstream of the PRV will also have a significant effect, as a small amount will boil off to dissipate the superheat. This does not happen instantaneously and it is possible for wet steam to pass through a PRV and downstream for there to be superheated steam in the centre of the pipe with liquid water adhering to the walls. There will then be a mixing length effect within which the superheat is dissipated by the condensate and line losses.

All line equipment exhibit this effect as there is a much greater pressure reduction through line equipment compared to pipe friction losses. This superheat generation may be predictable or unexpected.

Line speed [m/s]	Heat flux of superheat fraction after the PRV [kW]	Insulated line loss [kW/m]	Line length for superheat dissipation [m]	Uninsulated line loss [kW/m]	Line length for superheat dissipation [m]
1.95	0.0		0		0
5	2.8		25		3
10	7.4		67		8
15	12.1	0.111	109	0.918	13
20	16.7		151		18
25	21.3		192		23
30	25.9		234		28

Table 6: Superheat dissipation length, DN 100 pipe for stated conditions.

An example of an unanticipated situation would be a partially opened isolation valve (note: these should be either fully open or fully closed). In this instance there is an unexpectedly high pressure drop across the valve which causes superheat. It is quite possible for this to happen and for the steam system to operate effectively as they are intentionally designed with an excess of pressure. Alternatively, a partially opened isolation valve may cause local starvation downstream as the pressure drop is too great to allow the required amount of steam through.

An example of a predictable superheat created by losses through equipment other than a PRV would be the bluff body of a vortex flowmeter or the orifice plate of a DP flowmeter.

The effect of superheated steam on volumetric meters located in “saturated steam” systems

The presence and potential endurance of superheated steam in what are notionally saturated steam systems is, for many, unexpected. When a steam meter is located in a line with superheated steam one should also ask the question: what possible effect might this have on volumetric meters which are externally compensated by either only pressure or temperature for mass?

Temperature only compensation

Let us suppose that we have a temperature only compensated volumetric steam meter located immediately after the PRV in the previous example, i.e. 9 bara upstream and 4 bara downstream, and that the PRV is not insulated. The temperature of the superheated steam would be used to calculate the density based on a look-up table which assumes that the steam is on the saturation line and the resultant look-up density will be higher than the actual density. This will produce an error in the mass flow rate equal to the over-read error in the density. For the 4 bara case immediately downstream of the PRV, the maximum possible error is shown in Table 7. In practice the induced error is normally less than the maximum due to the effect of other factors which serve to mitigate it. However, errors of an over read of up to 20% are not uncommon in industry.

Pressure only compensation

The above exercise can be repeated for a volumetric meter with mass compensation by pressure only (Table 8). Using the same conditions as the temperature only compensation example, we can see that the error reaches a maximum of 4.1%, i.e. it is an order of magnitude less than for the temperature only compensation.

Actual pressure [bara]	° Superheat [K]	Assumed pressure [bara]	Assumed density [kg/m³]	Actual density [kg/m³]	Error [%]
9.0 before PRV 4.0 after PRV Saturation temperature [°C] 143.6 after PRV	0	4.00	2.160	2.163	0
	2.0	4.23	2.278	2.150	5.9
	4.0	4.46	2.398	2.138	12.2
	6.0	4.71	2.523	2.126	18.7
	8.0	4.97	2.653	2.114	25.5
	10.0	5.24	2.789	2.102	32.6
	12.0	5.52	2.930	2.091	40.1
	14.0	5.82	3.076	2.080	47.9
	14.3	5.86	3.099	2.078	49.1

Table 7: Error caused by temperature only compensation in superheated steam when saturated steam is assumed.

Pressure [bara]	° Superheat [K]	Temperature assumed by the meter [°C]	Density assumed by the meter [kg/m³]	Actual density [kg/m³]	Error [%]
9.0 bara before PRV and 4.0 bara after PRV Saturation temperature after PRV is 143.6 °C	0	143.6	2.163	2.163	0.0
	2.0			2.150	0.6
	4.0			2.138	1.2
	6.0			2.126	1.7
	8.0			2.114	2.3
	10.0			2.102	2.9
	12.0			2.091	3.4
	14.0			2.080	4.0
	14.3			2.078	4.1

Table 8: Error caused by pressure only compensation in superheated steam when saturated steam is assumed.

Recommendation

Due to the possibility of the presence of unexpected superheat in a nominally saturated steam system, it is recommended that volumetric steam meters should be compensated by both temperature and pressure in order to avoid unnecessary induced errors.

Some hazards of steam

Steam is extensively used in many different industries because its properties, from a engineering perspective, are highly advantageous. However, these same properties give rise to a number of hazards associated with steam; principally its expansive properties and elevated temperature. In order for steam to be safely harnessed it is essential that engineers and plant operators have sufficient understanding of these hazards so that they can minimize the risks to themselves and others which naturally arise when steam is used. The following gives a brief description of a number of the principal hazards associated with steam.

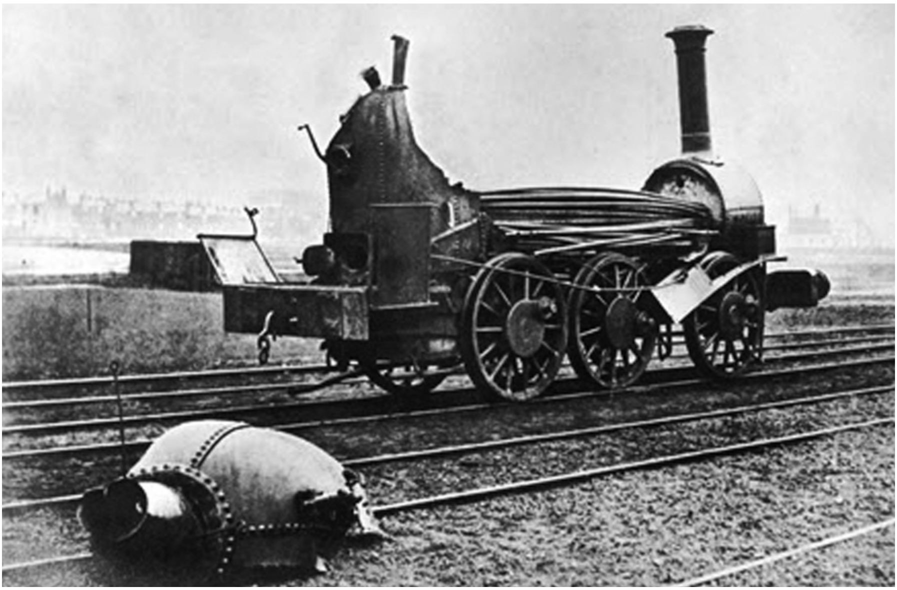


Figure 15: Locomotive boiler explosion and its consequences.
Source: Wikimedia Commons (https://commons.wikimedia.org/wiki/File:Boiler_explosion_1850.jpg)

Boiling liquid expanding vapor explosion (BLEVE)

This occurs when a vessel containing a saturated liquid at elevated pressure fails. When the vessel fails, a fraction of the liquid is vaporized and rapidly expands giving rise to the term BLEVE. BLEVEs are not restricted to steam, however, only steam is under consideration here. Historically BLEVEs are associated principally with MTHS and water-tube boilers. There are several different causal paths for a BLEVE to occur:

BLEVE due to overpressure only:

Whilst possible, there are no recently documented examples in Western Europe of MTHS boilers failing purely due to overpressure (although they were relatively common in the nineteenth century) as this requires a complete failure of both the pressure controls and the safety valve.

BLEVE due to failure of the longitudinal seam weld at normal working pressure:

There are several documented examples of BLEVE involving steam boilers which occurred at normal working pressure. In these accidents the pressure vessels were significantly weakened by corrosion and thermal cycling which caused stress corrosion cracks to form in the longitudinal seam weld. Once the cracks reach a critical size the pressure inside the boiler causes the whole seam to fail resulting in a BLEVE. *Note:* the catastrophic failure of any weld on a pressure vessel can result in a BLEVE. However it is the longitudinal seam weld that more commonly fails.

BLEVE due to a low-water condition:

This mode of failure is predicated by some failure of the level controls which causes the heating surfaces inside the boiler to become exposed to steam vapor whilst the furnace is still fired. The heating surfaces then start to heat up. If the metal temperature becomes too high, the furnace wall becomes plastic and loses its strength. The external pressure on the furnace then causes it to buckle inwards or collapse. In extreme cases the furnace detaches from an end plate resulting in a BLEVE.

BLEVE due to insulation of the heating surfaces:

In this mode the heating surfaces are covered on the waterside with a layer of deposition. This is often due to hard scale caused by poor water treatment although other contaminants can have the same effect. The deposition insulates the furnace from the water which would normally carry the heat away. This results in the furnace overheating with the same results as a BLEVE due to a low-water condition.

Examples of BLEVE also occur in other sectors of industry and examples can be found in the literature or on social media sites.

Column collapse water hammer

This is considered particularly dangerous because it can occur when steam is not being used for process, i.e. when “nothing” is thought to be happening. It requires that a downstream valve be closed which creates an effective dead-leg in conjunction with a U-bend from which the condensate is not removed (Figure 16 showing initial condition). As the condensate collects it gradually fills the bottom of the U-bend and starts to rise up the vertical legs.

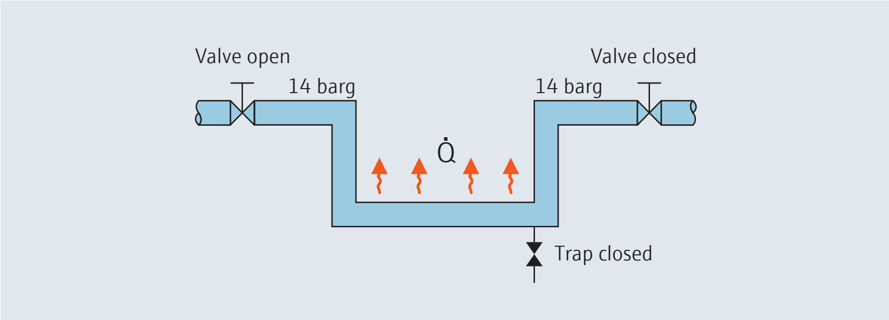


Figure 16: Column close water hammer, starting condition.

At this point a pocket of stream is trapped downstream of the U-bend but it is maintained at pressure by continuity from the live steam upstream, through the condensate (Figure 17). As the condensate rises up the vertical legs it becomes stratified and a temperature gradient forms with the condensate at the saturation temperature at the free surfaces and the sub-cooling to some minimum at the lowest point.

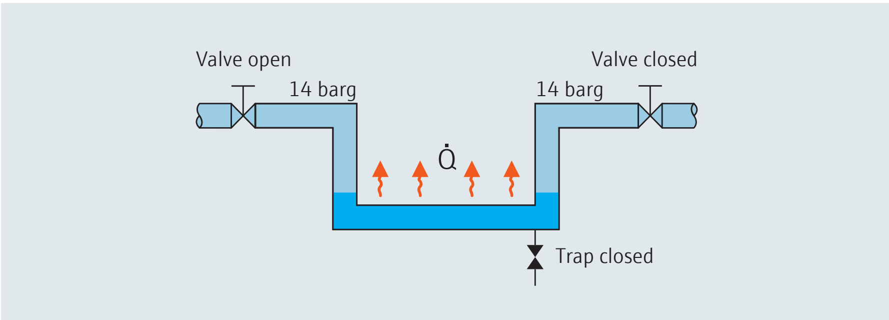


Figure 17: Column closure water hammer, stratification and sub-cooling.

If left in this situation for long enough which, depending on the conditions and line size, could take several days, the water level will rise up (Figure 18) until it starts to flood the horizontal line sections.

As it does so, the downstream pocket of steam will come into contact with sub-cooled condensate and will start to condense. This reduces the size of the steam pocket and draws more condensate up the line exposing the remaining steam to condensate

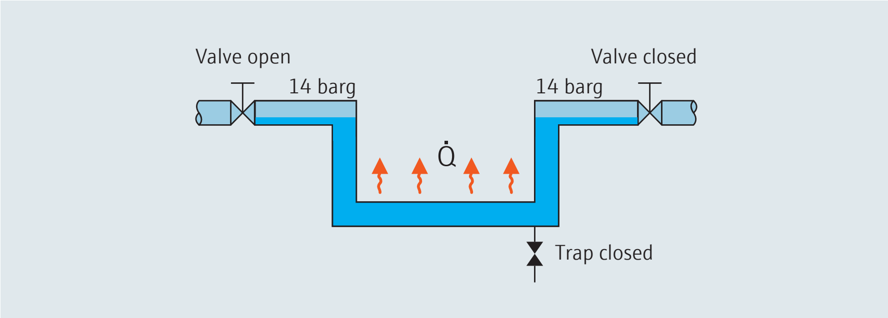


Figure 18: Column closure water hammer, flooding and steam pocket collapse.

which has been sub-cooled to a greater degree. At some point the remaining steam in the pocket will condense almost instantaneously creating a partial vacuum. The column of condensate will then be pushed into this void by the upstream pressure (Figure 19), until it is driven into the downstream valve at which point line rupture occurs. For a detailed explanation of the cause and consequence of an example of column closure water hammer the reader is referred to the UK Health & Safety Executive (HSE) report into the accident at BP Grangemouth in 2000.

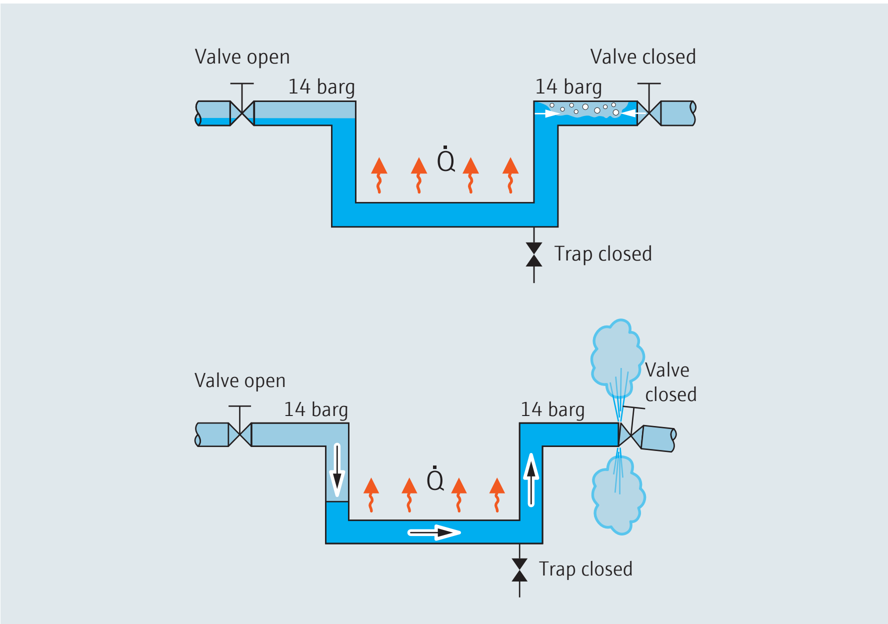


Figure 19: Column closure water hammer, line rupture.

Sub-cooled condensate induced water hammer

There are a large number of different scenarios in which sub-cooled condensate induced water hammer can be induced. The following text illustrates just a few possible scenarios. This type of hazard is commonly associated with cold starts or distribution systems that are operated intermittently. For sub-cooled condensate induced water hammer to occur it is necessary for there to be sub-cooled condensate in the distribution system, pooled or collected for some reason. When live steam comes into contact with this sub-cooled condensate, it condenses, giving up its heat to the condensate. If the live steam is brought on quickly it has a tendency to “jet” into the

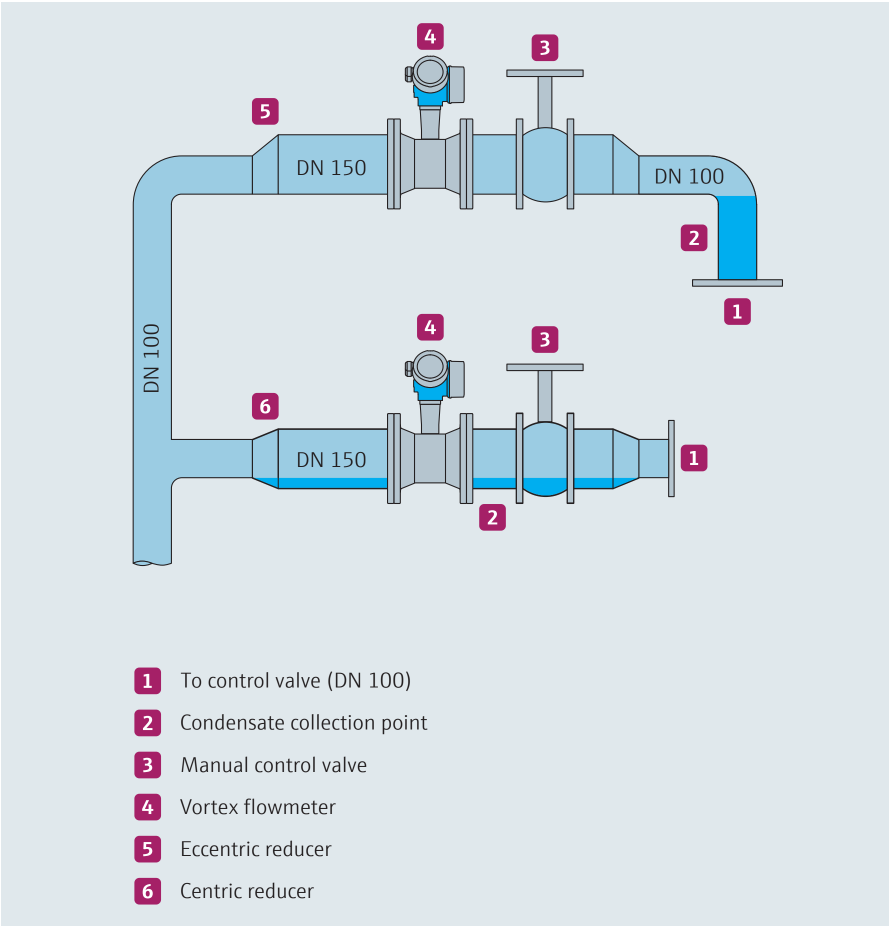


Figure 20: Condensate collection due to trap failing closed.

condensate and steam bubbles become isolated. Because they are surrounded by relatively cold condensate they collapse instantaneously with a popping or banging sound. These collapses result in shockwaves travelling through the system which are potentially very destructive – this form of water hammer can be likened to hitting the pipework with a sledgehammer. This description above represents relatively minor sub-cooled condensate induced water hammer. In severe cases a line rupture is quite possible and there a number of documented instances which have resulted in death or severe injury.

Figure 20 is a representative drawing of part of an actual system which was subjected to repeated water hammer, causing repeated failures of vortex flowmeter sensors. The system was operated throughout the day until eight o'clock in the evening at which point an upstream control valve was closed and the downstream section of the line allowed to cool. At approximately four o'clock in the morning this control valve was opened and live steam at approximately 8 barg entered the system. Looking at the upper line, the pipe dropped vertically down and condensate collected against the top of a closed control valve (Fig. 20/1). Looking at the lower line, concentric joints had been used to increase the line size to DN 150 rather than eccentric ones as in the upper line. This again allowed the condensate to collect with no means of escape.

When the live steam came into contact with the sub-cooled condensate it "jetted" into it and water hammer resulted. The shockwaves which resulted were of suf-

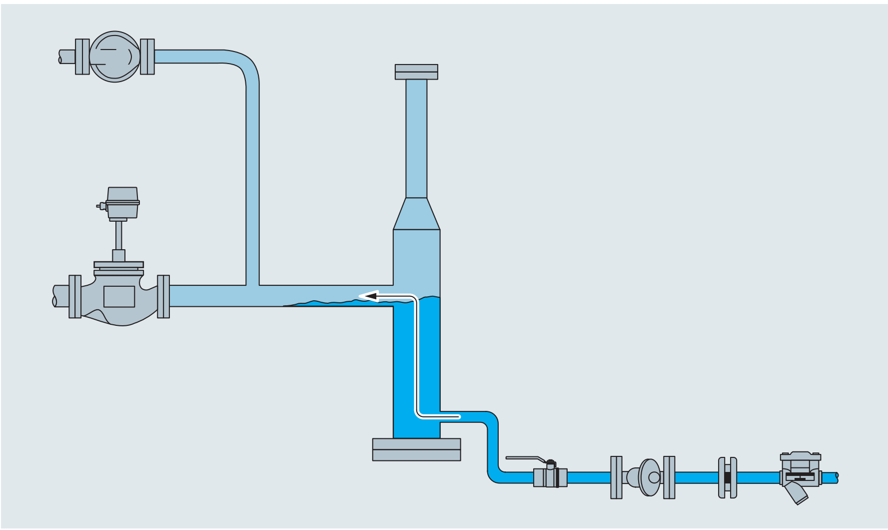


Figure 21: Condensate collection due to trap failing closed.

ficient magnitude to cause failure of the vortex flowmeter sensors on a repeated basis. The pipework was subsequently modified to remove the condensate collection point. However, if it had been left long enough there would have been a risk of line rupture.

Figure 21 shows another way in which condensate can collect – through a trap which has failed closed. This is not uncommon as steam is an aggressive medium. However, the incidence of water hammer events is often exacerbated by the failure of end users to check their steam traps on a regular, frequent basis.

Flash steam explosion

Although uncommon, flash steam explosions still occur in steam boilers and can also be found in other settings. These happen when relatively cold liquid water hits a very hot surface. In doing so, some of the water is instantly vaporized to form superheated steam which expands very rapidly with an accompanying pressure wave. If this occurs inside a pressure vessel such as a steam boiler, it can result in catastrophic failure.

An example of this was a boiler that had a dangerous low-water condition – the level was low enough that the heating surfaces were exposed to steam vapor rather than liquid water whilst the burner was still firing. This situation is sometimes called a “dry-fry”. The boiler was left long enough that the metal of the furnace overheated. The operator who found the boiler in this condition manually injected feedwater into the boiler instead of shutting it down. This caused a flash steam explosion of such magnitude that the pressure vessel ruptured and the boiler exploded. The boiler was forced some 30 to 35 meters away from its original position and debris was spread over a large area.

Overpressure in the distribution system

Steam is generated at a higher pressure than is necessary for the consumers to ensure that there is enough pressure available when it reaches the point of requirement. To provide controllability pressure reducing valves (PRV) are employed near to the consumers. Downstream of the PRV safety valves are fitted as the downstream equipment may be rated to a lower pressure than the boiler. If both the PRV and safety valve fail there is a risk of downstream failure due to overpressure.

Overpressure (inside a pressure vessel)

See section “Boiling liquid expanding vapor explosion (BLEVE)” on Page 55.

Plug flow water hammer

This phenomenon is caused by a slug of water which has filled the pipe cross section and is pushed along by the steam behind it. It can quite easily cause line rupture if the slug impacts on a pipe bend, for example.

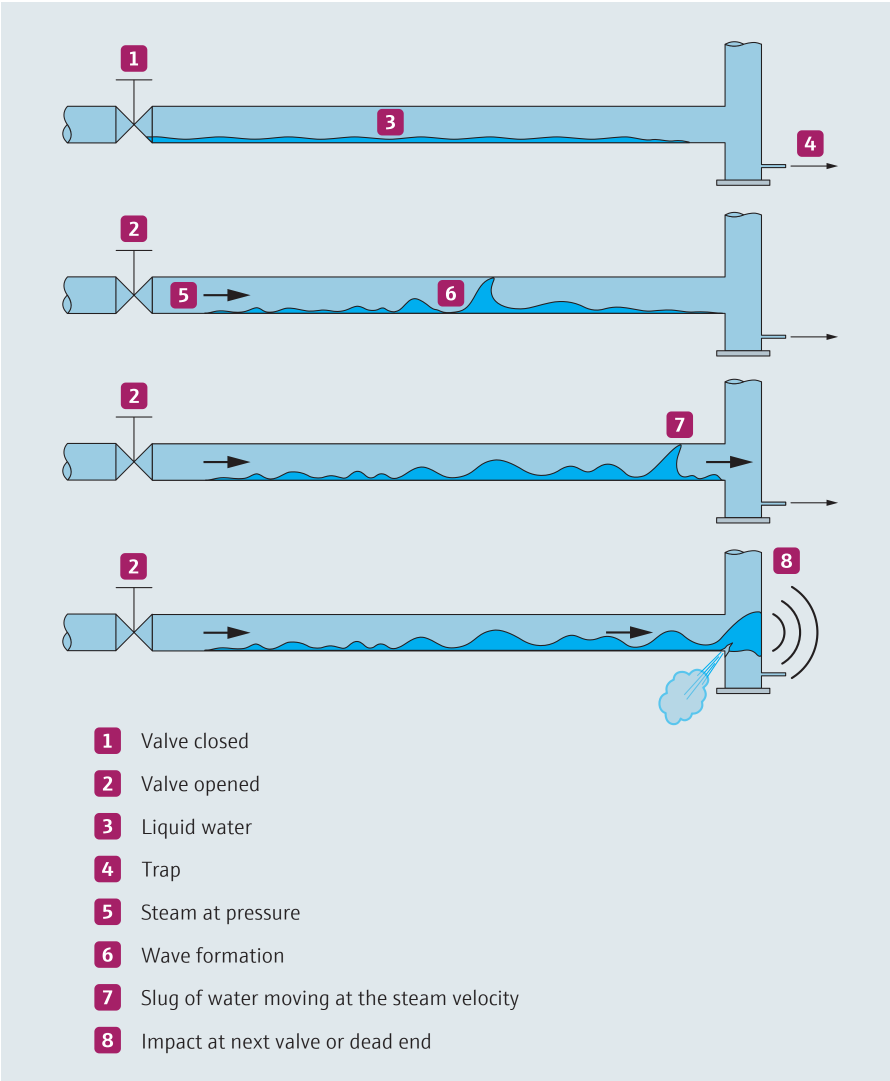


Figure 22: An example of plug flow water hammer.

For plug flow water hammer to occur condensate must be present in sufficient quantity in the steam line. The condensate can be either saturated or sub-cooled. The presence of sufficient condensate is frequently caused by poor design or inadequate maintenance exacerbated by inadequate fall in the lines, incorrect support, failed traps and missing insulation. As with all types of water hammer, the probability of its occurrence is significantly increased if steam is admitted too rapidly into the line.

An example is shown in Figure 22: An upstream valve is closed and downstream of the valve condensate has collected. When the valve is opened the steam rushes down the pipe and as it does so it starts to push the condensate ahead of it. If there is enough condensate and the line is long enough, the amount of condensate being pushed along increases until it completely fills the steam main. At this point the slug of water is travelling at the speed of the steam which could easily be 110 km/h (~70 mph) or more. When the slug reaches the riser required for the steam trap it smashes into it and if the slug possesses enough kinetic energy it will cause a line rupture. The probability of line rupture is increased in older and poorly maintained steam systems.

Steam hammer

This is a phenomenon associated with the rapid opening or closing of a valve. Consider a long steam line which contains a valve which can be rapidly opened or closed. Let us suppose that the valve is open and that the bulk velocity of the steam is 30 m/s (110 km/h or ~70 mph) and that the valve is suddenly closed. However, the steam still possesses momentum. Upstream of the valve this momentum causes the mass of steam to be driven onto the closed face of the valve causing a pressure spike. When the over pressure has reached a maximum, the steam rebounds off the surface of the valve and eventually equilibrium is reached.

Downstream of the valve the momentum of the steam carries it away from the valve resulting in a rapid local pressure reduction. It is the combination of the local over and under pressures that constitute steam hammer. Steam hammer normally does not result in a line rupture or valve failure unless combined with a further risk factor such as badly corroded lines or cast iron valves.

Temperature

Steam at pressure is by its very nature hot. Due to the high thermal conductivity of the associated steel work, its temperature approximates closely to that of the steam inside the pipework even when uninsulated and located in adverse conditions. The temperature of 10 bara saturated steam is 179.9 °C and the external temperature of its steam line will be approximately 175 °C. Steelwork at this temperature will very easily burn human skin. Minimal, effectively instantaneous contact will leave skin with a light burn or blistering. Contact for just a few seconds will very likely cause a full penetration burn – the type that requires a skin graft.

When opening up steam lines for manual work, the operator must not only ensure that there are suitable isolations in place but also that the temperature of the line has fallen sufficiently – preferably touch temperature. The reason for this is that even though a line which has been isolated and a local pressure gauge shows zero, if the line is very hot some fraction of any residual condensate will re-boil at low pressure. If a line is quickly isolated and immediately breached, i.e. still hot, there is a risk of burns to the individual.

Note: The above is **not** a definitive statement concerning the opening up of a steam line. Great care must be exercised to prove the contents of the line and sufficiency of isolations before work is undertaken.

Vacuum draw

The density of steam is much lower than liquid water at the same temperature. Under certain circumstances this can lead to dangerous incidents:

Vessels which are steam-cleaned:

During this process most of the air in the vessel will be evacuated by steam. If, once the process is completed, the vessel is sealed shut the pressure inside the vessel will fall as the steam condenses. This will eventually lead to a large pressure difference between the inside of the vessel (at some sub-atmospheric pressure) and the outside at atmospheric pressure. Certain vessels which are designed to withstand internal pressure, such as thin walled liquid tankers, are not able to withstand significant external pressure. The vessel may then collapse due to the external pressure.

Vacuum draw into a boiler:

This occurs when a boiler is allowed to cool with the crown valve left open. It also requires that a non-return valve is either not fitted or is defective. As the boiler cools, the density of the liquid fraction inside the boiler increases. Once the temperature has fallen sufficiently a partial vacuum occurs. This then draws steam back through the main into the boiler. If some form of contamination has entered the steam main, this will also be drawn into the boiler. In one instance acid was drawn into the boiler which caused corrosion leading to the boiler being condemned at its next inspection. In another incident oxygen was drawn into the boiler. This contributed to the eventual failure of the boiler and a subsequent explosion. In a third incident a steam boiler became fully flooded as did the steam line immediately downstream of it.

Waterlogging

This is not in itself a hazard, however, when a sub-cooled waterlogged steam system is combined with live steam, then condensing water hammer events will occur. It is possible for a single condensing water hammer event to result in a line rupture. However, it is more common for the water hammer events to be of a smaller magnitude. These may then contribute, over time to the failure of line equipment in the vicinity of the water hammer or a line rupture.

Waterlogging may result from poor design in which condensate can collect in a location from which it cannot be removed due to the lack of a steam trap. Incorrect repair work may also result in this situation in an otherwise well designed system. Traps which fail closed or which are rendered non-functional due to blockages. Poor maintenance is another cause.
Note: The above topics are not exhaustive. Poor or missing insulation will contribute to waterlogging as this will greatly increase the rate of condensate formation.

Figure 23: Destroyed factory building caused by a boiler explosion (Meuselwitz, Germany, 1904).

Boiler operation

Modern steam boilers are automatic in operation and it is very important to understand this and its implications for both operators and managers. Their roles are briefly discussed as is the role of technology and a few of the more significant risk factors which affect their safe operation.

The boiler operator

The role of the boiler operator has undergone marked change in recent years due to the ongoing trend to reduce the use of manpower in the boiler house and improvements in technology. On many sites, the boiler house is effectively unmanned on an hour-by-hour basis with the operator only coming in to perform checks as required, the frequency of which might be as little as once every three days on some sites. Furthermore, it should be noted that the operators themselves are often less knowledgeable about boiler operation and steam systems. For many it is a secondary work activity and moreover one that is often outside their core competencies.

Let us suppose that we are bringing a modern steam boiler online. It has adequate water inside it, the burner is switched on, but not firing as it is at its maximum working pressure and the crown valve is closed. Once the crown valve is opened the boiler will operate automatically without human intervention, and in theory could do so indefinitely. This is because the steam is drawn from the boiler solely due to the pressure difference between inside the boiler and in the distribution system. When steam is drawn out of the boiler then its pressure drops. When it falls to a predetermined set point the burner will start automatically and will keep firing until the pressure increases to the maximum working pressure whereupon the burner will stop firing. The burner will not fire again until the pressure again drops to the set point. During normal operation the water level inside the boiler is also automatically controlled and feedwater will be periodically pumped into the boiler, maintaining sufficient level required for safe operation. This set of events could, in theory, go on indefinitely without human intervention once the crown valve has been opened until it is required to be closed. One must therefore ask the question: what is the role of the operator in today's boiler house?

- Oversight: A multi-tubular horizontal shell boiler is a store of energy when at pressure and there is an ever present risk of catastrophic failure. Therefore the checking and testing of controls and safety critical systems is of crucial importance.
- Response to alarms and troubleshooting: It is important that operators can respond to alarms in a timely manner and also to investigate and correctly

identify their various causes in order to prevent the risk of a dangerous occurrence from happening. It is therefore vital that operators are appropriately trained and competent.

- Operation: There is still some operation required of steam boilers. One example is cold starts which must be performed in accordance with the manufacturer's instructions and which also must be performed with an operator in attendance. It should be noted that repeated cold starts which are incorrectly performed have contributed to a number of boiler failures.
- Maintenance (preferably planned rather than reactive): Once maintenance has been performed, it is vital that safety systems are tested as a number of accidents have occurred because checks have not been performed following maintenance works.
- Record keeping: Records are very useful on a number of levels, for example where sites operate multi-shift working the operators may not even see each other on a regular basis so the written records form the main point of contact. Records can also demonstrate that the correct checks and tests have been performed thus reducing the risk of regulatory action in the event of an accident. They are also necessary for risk analysis and can be very useful when troubleshooting.

The manager

Following on from the changes to the role of the boiler operator, that of the manager has also changed. On many sites the boiler and steam system comprise only part of a manager's responsibility and as with the operator, in many instances specific specialist knowledge of boilers may well not form part of their primary core competencies.

In the first instance a boilerhouse must comply with local and national laws and regulations. This is a primary duty. Secondly the installed plant should be operated in a manner which reduces risk as far as is reasonably practicable. (This statement reflects the global trend towards goal-based regulation as opposed to a prescriptive approach.)

In achieving the above the manager needs to ensure that systems of work are fit for purpose and also that manning levels of suitably trained and competent persons are adequate. The manager must then enforce appropriate oversight to ensure that the plant is operated as intended and that any incidents are appropriately investigated. The manager must also ensure that he or she has adequate knowledge so that their duties and obligations can be adequately discharged.

As manning levels have decreased, there naturally follows the question: how may technology be gainfully employed to partially replace/aid the boiler operator in this paradigm? A few examples are given in the text below. The introduction of new technology should be given careful consideration to ensure that it is both appropriate and that the best use is made of it.

The role of technology in the modern boiler house

A partial answer is to understand the role that technology can play in replacing and helping the operator. Within this frame it should be noted that considerations towards safety – bearing in mind the potential consequences of catastrophic failure – should be given primacy over efficient, effective operation. Fortunately a "safer" boiler installation is more likely to be an effective, efficient unit.

Consider Figure 24, a fault tree diagram based on actual boiler incidents which postulates the question: what are the causal factors of a waterside explosion or furnace collapse for a MTHS boiler? As can be seen, there are a number of different factors that can contribute to a failure and in some accidents several factors combine. Therefore reduction in the risk of a single factor in isolation will reduce the overall risk of a failure.
Note: It should not be deemed sufficient to mitigate risk factors on a partial or ad hoc basis, they must be considered in toto although it may be necessary to examine each one individually.

Let us now consider how and where technology may be usefully employed in mitigating individual risk factors, helping the boiler operator and substituting for the operator.

Pressure cycles

The pressure vessel of a steam boiler is constantly subjected to changing temperature and expands and contracts as the temperature rises and falls. This subjects the vessel to fatigue. During the statutory inspection the inspector will estimate the number of full and partial pressures to which the vessel has been subjected and this will influence the predicted life of the boiler. At present pressure cycles are rarely recorded. However, doing so would permit the inspector to make a more accurate calculation regarding the predicted life of the boiler. It would also allow the user of the boiler to make informed decisions concerning the reduction of, in particular, full pressure cycles.

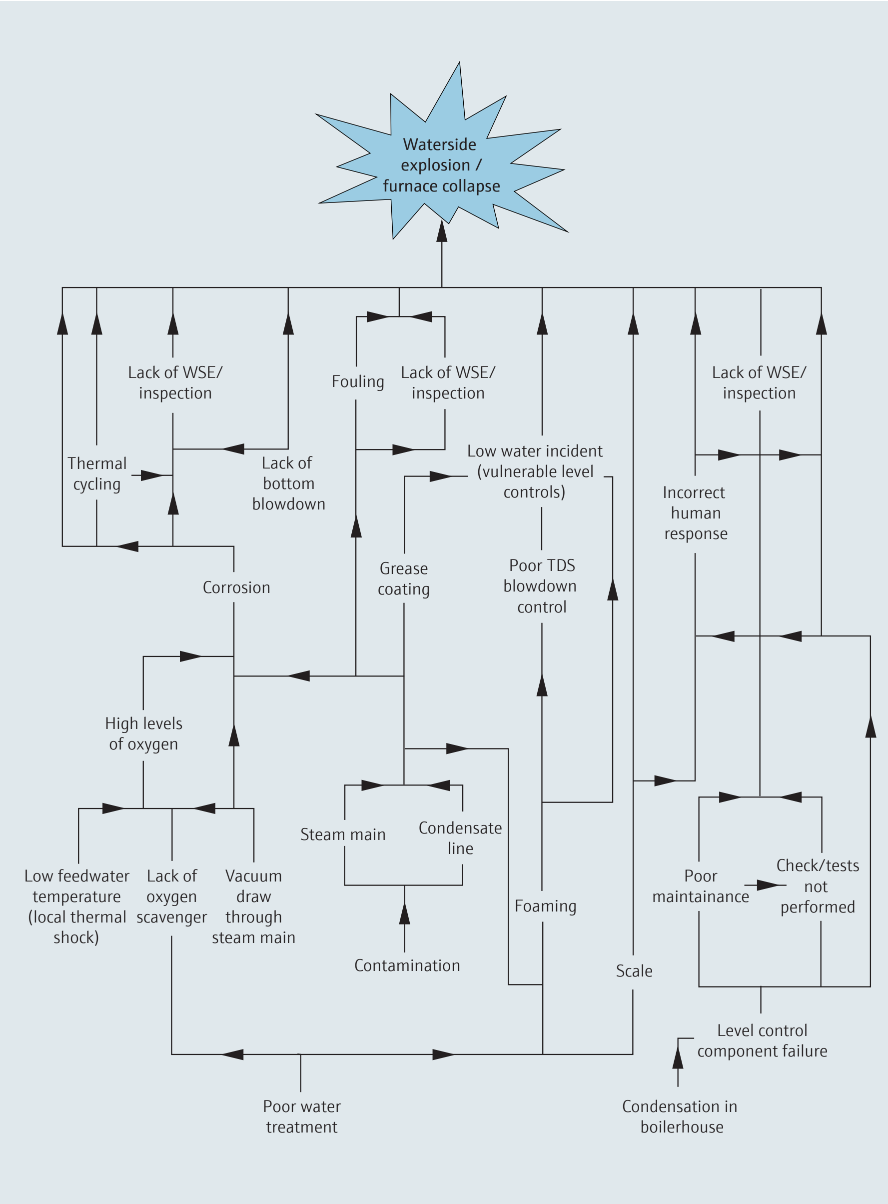


Figure 24: Causal factors of a waterside explosion or furnace collapse for MTHS boilers (although the diagram is based on actual historical accidents, it is not exhaustive). WSE = Written Schemes of Examination.

Over-firing during cold start
Excessive firing during cold start means that the furnace is very much hotter than the rest of the steelwork that comprises the boiler's pressure vessel. As a consequence the furnace thermally expands much more than the rest of the vessel, resulting in a number of very undesirable consequences:

- The vessel is stressed in ways which the designer did not intend. This will contribute to an unexpected degradation of the boiler's anticipated life due to increased fatigue.
- The boiler end plates will tend to bow out. In extreme cases this can cause tubes in the convective passes to partially pull out of the end plate resulting in leaks.
- Cracks may form at welded ligaments.
- It is a causal risk factor for catastrophic failure of the pressure vessel.

Technology can be used to automate the firing sequence of a boiler to prevent excessive thermal strain. *Note:* This does not mean that a cold start can be un-manned. A cold start must always be attended by a trained and competent individual.

Selection of level controls
Many boilers in the United Kingdom are fitted with level controls which comprise a mixture of capacitance and conductivity probes. These are fitted with protection tubes, and in general they are reliable and work well. However, they all rely on the conductivity of the water inside the boiler to function correctly and there is a residual risk in situations where foaming occurs. This may be due to some failure of the TDS controller or unwanted contamination.

Alternative types of probe such as guided radar, vibrating fork, positive displacement and differential pressure are available at reasonable cost. By employing instruments with different principles of operation, diversity is brought into the installation and risk posed by certain known and foreseeable events is reduced. It should however be noted that all controls possess vulnerabilities and selection of appropriate controls should be made on a site-by-site basis.
Note: On sites where very pure water is used to feed the boiler, e.g. from reverse osmosis plants, there have been instances where the capacitance/conductivity systems have failed to operate correctly when the boiler is first filled with water as its conductivity is too low.

Overview of required tests and checks
Although operators in many instances are not required to be in the boiler house at all times, they are required to undertake regular tests which comprise two main categories. There are what are termed daily checks, although – depending on the

equipment fitted to the boiler – it is possible that the frequency is reduced to once every three days. There is also the requirement for e.g. an evaporation test every week.

An evaporation test is one where the water level inside the boiler is lowered to force a low-water event to confirm that the limiters are working correctly. It is very important that tests are completed as required. In the case of the evaporation tests there have been a number of catastrophic failures of boilers due to a primary failure of a safety critical system which was not tested as it should have been.

Technology can be employed to electronically record that tests have been completed, when they are intended to have been performed and evidence this to both management and regulatory bodies.

The above is also important to prevent abuse of systems of work: there have been instances of operators filling out several months worth of daily and weekly records just before an inspector is due to undertake the statutory inspection.

Steam boiler efficiency

There are two methods of assessing the thermal efficiency of a steam boiler: the fuel-to-steam (direct) method and the indirect method (see below). They are not directly comparable, although it may be possible to estimate one from the other under certain circumstances. There are a number of standards which may be applied to these methods including BS 845:1987, BS EN 12953-11:2003 and DIN 4702-8. The most common standard used in the United Kingdom is BS 845 although references to DIN 4702 and BS EN 12953-11 are not infrequent. As with all calculations, the uncertainty will depend upon the care taken with the measurements and calculations. Efficiencies determined using different standards should not be directly compared as they use different methodologies and reference quantities. For example BS845-1 uses 15 °C as the reference temperature whereas EN12953-11 uses 25 °C. Efficiencies should be quoted on either a gross calorific (GCV or HHV) or a net calorific (NCV or LHV) basis although it is not uncommon to see net efficiencies quoted which are greater than 100% wherein a gross basis has been compared to a net basis.

Steam boiler manufacturers will normally only quote indirect efficiencies as these are solely derived from the boiler whereas fuel-to-steam efficiencies are site dependant with local factors such as water quality and blowdown rates having significant effects. Manufacturers all use different calculation bases, e.g. fuel carbon/hydrogen (CH) ratio and calorific values to generate quoted efficiencies. This can make it difficult for the user to compare one boiler against another.

Considering fire-tube boilers, the combustion efficiency is determined by the burner and furnace combination whilst the overall thermal efficiency is determined by the design as this governs the exit temperature of the gases from the boiler for a given firing rate and boiler pressure. A number of configurations have been tried, the United Kingdom market settling on the three-pass wetback to provide the optimum balance between running costs, physical size, complexity and capital cost. Recent innovations in Europe, Southern Africa and North America have renewed interest in enhanced heat transfer techniques, and boilers with improved thermal efficiencies are starting to appear on the market.

The following refers to BS 845 although the basic principles are the same for the other methods.

Indirect method

This is covered in BS 845-1:1987 and is referred to as the concise procedure. The indirect method is suitable for all fuel types under steady state conditions and may be defined as “The determination of thermal performance by the assessment of the thermal losses and the measured thermal input or output”. The calculation determines the various heat losses from the boiler in the form of percentages. These are then subtracted from 100 to give the calculated indirect efficiency. BS 845-1 states a tolerance for the results of ±2% for thermodynamically simple boilers. It should be noted that although boiler manufacturers will quote a BS 845-1 efficiency for boilers fitted with economizers, these do not strictly fall within the definition of “thermodynamically simple”. The calculations are stated in section six of BS 845-1 and the losses may be summarized as follows:

- Loss due to sensible heat in the dry flue gas
- Losses due to enthalpy in the water vapor in the flue gases
- Loss due to unburnt gases in the flue gases
- Loss due to combustible matter in ash and riddlings
- Loss due to combustible matter in grit and dust
- Radiation, convection and conduction losses

It should be noted that the calculation of the heat input used in BS 845 may be manipulated to render it independent of time. This forms the basis of combustion analysers which provide instantaneous efficiencies. Only clean burning liquid and gaseous fuels should be assessed on an instantaneous basis. Tests undertaken strictly in accordance with the standard should last for a minimum of one hour during which time steady state conditions are maintained and six sets of results obtained in that time frame.

The inputs and variables required by the BS 845-1 calculations are:

- Mass of fuel supplied over the duration of the test
- Duration of the test (This may be eliminated for instantaneous calculations)
- Gross or net calorific value of the fuel
- Carbon/Hydrogen (CH) ratio of the fuel
- Loss due to radiation, convection and conduction
- Temperature of the combustion air entering the burner
- Carbon dioxide content of the exhaust gases (percentage, dry volumetric basis). Oxygen content, wet or dry basis may be substituted with suitable modifications to the calculations.
- Temperature of the exhaust gases
- Moisture content of the fuel as fired (considered negligible in cold or temperate climates)
- Carbon monoxide content of the exhaust gases (percentage, dry volumetric basis)

- Quantity of ashes and riddlings per tonne of fuel burnt/hour (dry basis)
- Quantity of dust and grit per tonne of fuel burnt/hour (dry basis)
- Carbon content per tonne of fuel burnt of ashes and riddlings (dry basis)
- Carbon content per tonne of fuel burnt of grit and dust (dry basis).
  (These are not required/negligible for liquid and gaseous fuels under conditions of good combustion.)

Typical peak efficiencies for standard natural gas and light oil-fired MTHS boilers without enhancements are around 80 and 83% respectively when correctly commissioned; the main difference being due to the differing carbon/hydrogen (CH) ratios of the two fuels. The losses of MTHS steam boilers are thus around 17 to 20% of which the flue gas losses are by far the most significant, accounting for over 90% of the losses.

Recent innovations in enhanced heat transfer techniques applied to the convective passes have enabled thermal efficiencies to be increased by up to 2.5% through exhaust gas temperature reduction although the size of the boiler, firing rate and fuel type will have a significant impact on the final improvement in efficiency.

There is however an inherent limit to improvements in boiler efficiency imposed by the saturation temperature of the steam where the pressure vessel is defined as the system boundary. The steam inside a boiler running at 10.3 barg, for example, has a saturation temperature of 184.1 °C, and as the exhaust gas temperature approaches this value, the effectiveness of the heat transfer within the pressure vessel progressively reduces. Moreover, it is not possible for the exhaust gases to be lower in temperature than that of the heat carrier.

Therefore, further efficiency gains are best considered external to the pressure vessel. The commonest example of this is the economizer, a gas to liquid heat exchanger used to pre-heat the incoming feedwater by cooling the exhaust gases. These are often sized to reduce the exhaust gas temperature by a further 80 to 100 °C, simultaneously increasing the feedwater temperature by 25 to 30 °C, resulting in an efficiency gain of 4 to 5%. An alternative is to use the exhaust gases to pre-heat the incoming combustion air.

Of the remaining variables only the carbon dioxide/oxygen content of the exhaust gases can be directly altered (via modification of the burner settings). For typical carbon dioxide concentrations, 8 to 12%, a 1% change will affect the boiler efficiency by around 0.65 to 0.75%. This can also be seen as a change in exhaust temperature due to the change in mass flow rate of the exhaust gases for a given amount of fuel burn.

For a given burner setting the exhaust temperature, as previously stated, is determined by the effectiveness of the convective passes. Fouling of the passes will cause the exhaust temperature to rise and thus the thermal efficiency to fall. Fouling may occur on either the gas or the water side of the convective passes. Fouling on the gas side is normally due to poor combustion and has little effect other than to reduce the thermal efficiency of the boiler. On the other hand waterside fouling is normally caused by a build up of scale on the internal heat transfer surfaces. If the scale forms a thick enough layer over the furnace, it can lead to a collapse of the furnace. Gas side fouling may occur over a very short time frame whereas waterside fouling often takes several months. Whilst the result of the fouling is the same (an increased exit temperature of the exhaust gases) analysis of the rate of change of this temperature may indicate the causal mechanism.

Shell losses are very difficult to measure precisely and the simplest method is to look up estimates, e.g. the appropriate table in BS 845-1. The tables state the shell losses as a percentage of the fuel firing rate at maximum continuous rating (MCR). For firing rates below this, the shell losses must be factored appropriately. For example a boiler rated at less than 2 MW is assumed to have a rate of heat loss of 1% of the heat input at full fire. At half load, while the rate of heat loss is the same when considered in e.g. kilowatts, the percentage heat loss is doubled to 2% as it is a function of the rate of heat input to the boiler.
Note: If the mass of water inside the boiler is known, the unfired standing losses can be estimated by measuring the boiler pressure twice over a one hour interval and calculating there from.

If a modern burner is attached to the boiler and is well set up, the concentration of carbon monoxide in the exhaust gases would be expected at less than 20 ppm and may consequently be ignored. Burners are set up to provide the optimum balance between air-fuel ratio and emissions particularly carbon monoxide and oxides of nitrogen. At high firing rates excess air levls of 15% are consistently achievable in conjunction with low carbon monoxide (CO) and nitrogen oxid (NO_x) levels. At low firing rates, however, excess air levels tend to be around 30% due to potential problems with flame stability.

The effect of variable changes on thermal efficiency are:

Exhaust temperature:	1% per 20 °C change in temperature
Ambient temperature:	1% per 20 °C change in temperature
Excess air:	0.5% per 6% change

(Values are approximate and dependent on fuel type and interaction with other variables.)

	Natural gas Gross (Net) [%]	Diesel Gross (Net) [%]
Loss due to sensible heat in dry flue gases	7.17 (7.96)	8.16 (8.70)
Loss due to enthalpy in the water vapor	11.65 (2.74)	7.60 (1.79)
Loss due to unburned gases in the flue gases	Negligible	Negligible
Loss due to combustible matter in ash and riddlings	Negligible	Negligible
Loss due to unburnt matter in grit and dust	Negligible	Negligible
Loss due to radiation, convection and conduction	1.0 (1.1)	1.0 (1.1)
Total	19.82 (11.81)	16.76 (11.55)

Table 9: Steam boiler losses, running at maximum continuous rating (MCR).

Composition of losses (indirect method)

Table 9 provides typical figures for small (<2 MW) natural gas and diesel fired boilers. All data are taken from Rose and Cooper "Technical Data on Fuel" 7th Ed. 1977 for the following conditions: ambient temperature, 25 °C, oxygen (O_2) content on dry volumetric basis, 2.5%, exhaust temperature 240/255 °C natural gas/diesel and calculated in accordance with BS 845-1:1987.

Variation and uncertainty of efficiency measurements

Figure 25 shows plots of BS 845-1 efficiency for a 2000 kg/hr boiler. The data are extracted from five separate tests, all diesel-fired. The tests were conducted carefully using the same high quality instrumentation for each test.

The data are spread in a band between 82% and 83.5%. As the tests were conducted under the same conditions that it is practical to obtain, it would not be unreasonable to assume that the greater part of the spread is caused by variation in the environmental conditions followed by the small but unavoidable differences in the test conditions rather than instrumentation errors. The major variables being:

- Variation in the ambient temperature will directly affect the combustion efficiency. It will also have a small effect on shell losses.
- Variation in the barometric pressure will affect the combustion in the furnace and the draft through the boiler.
- Variation in the wind speed in the boiler house will affect shell losses.

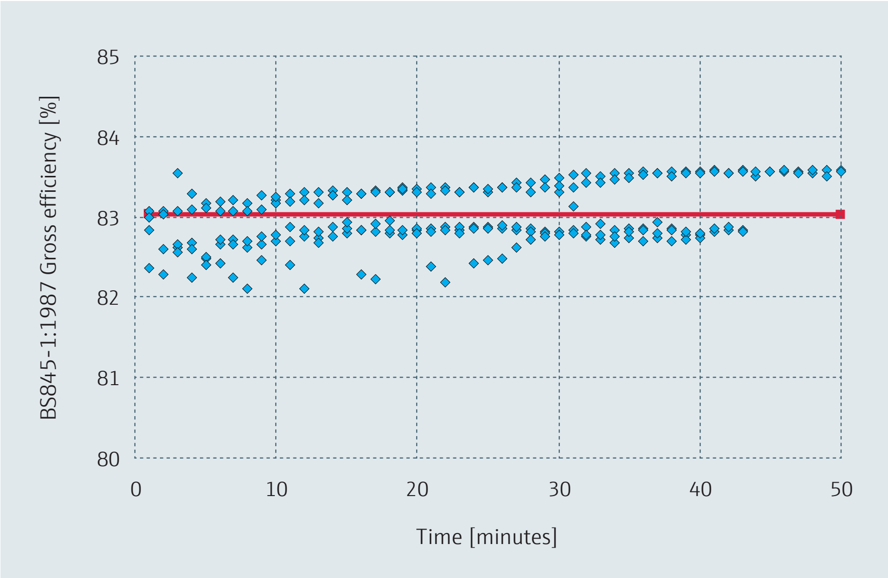


Figure 25: Efficiency measurement according to BS 845-1:1987.

- It is almost impossible to maintain a constant shell pressure and thus shell temperature both during and between tests. Even minor variations in the shell temperature will affect the rate of convective heat transfer in the tube banks.
- The burner used possessed mechanical linkages used to control the damper. This will vary the mass flow rate through the boiler.

A leading independent assessor of industrial boilers states that the most accurate test that they have achieved is ±1.8%, under carefully controlled conditions. When considering an individual test there are a number of potential sources of instrumentation errors. Errors may result from two main sources: inaccuracy of the instrument itself and poor experimental technique.

Exhaust gas temperature measurement error

Measurement of an accurate bulk exhaust temperature is difficult. In order for this to be done with a minimum of uncertainty, cross duct measurements of both local temperature and local velocity are required. An investigation of a 6 MW MTHS boiler revealed a cross duct temperature variation of approximately 50 °C, equating to 2.5% uncertainty in indirect efficiency.

Observational comparison of dial-type thermometers compared to 3-mm-diameter Endress+Hauser K-type insulated thermocouples indicate a temperature difference of between five and ten degrees centigrade; the dial-type thermometer giving the lower reading. Radiation losses may also be significant.
Note: Some boiler makers do not insulate all of their rear smoke boxes, mainly for reasons of economy, however it does have the effect of artificially suppressing the exit temperature and thus increasing the indicated efficiency, as demonstrated in the following example. The error, solely due to radiation, is the equivalent of over 0.5% of the calculated thermal efficiency.

Example: Suppose the exhaust gas temperature in the boiler flue is measured at 240 °C and that the flue is uninsulated and has an internal wall temperature of 75 °C. Then, the temperature loss due to radiation may be calculated as follows, using typical values for emissivity and heat transfer coefficient:

$$\delta T = \frac{\sigma\, \varepsilon\, (T_m^4 - T_s^4)}{h} = 12.4\ \mathrm{K}$$

δT = measurement error (K)
T_m = measured temperature (513 K)
T_s = radiation surrounding temperature (348 K)
σ = Stefan Boltzmann Constant (8.559×10^{-8} W/m² K⁴)
ε = emissivity (0.2)
h = heat transfer coefficient (50 W/m² K)

Thus radiation losses coupled with a less accurate indicator may easily contribute an error approaching 1%.

Exhaust gas composition error

One of the more accurate and common methods of determining the gas composition is the use of a zirconia probe which indicates the percentage (wet basis) of oxygen in the exhaust gases. If calibrated correctly they are considered to have an accuracy within 5% of reading. This would translate into an error of the order 0.1% when factored into the thermal efficiency. However, calibration intervals are often extended and thus zero, and span drift may give rise to much more significant errors. Errors which affect the water content of the exhaust gases will have a disproportionate effect on the overall thermal efficiency error due to the latent heat of the water.

Shell losses

Because of the sheer size and complexity of many boiler installations, accurate determination of shell losses is very difficult. The rate of heat loss will be significantly greater when the boiler is running but this in turn will be affected by the shell temperature. The prevailing atmospheric conditions may also have a significant effect. For this reason shell losses are generally taken from the reference tables of BS 845-1 Appendix B. The most common error when considering shell losses, whether measured or referenced, is failure to factor them for the firing rate.

It should also be noted that Appendix B makes no allowance for possible differences of shell temperature. For example, an 18 barg boiler will have a shell temperature of up to 209.8 °C whilst a 6 barg boiler would have a maximum of 165.0 °C with consequently lower radiation, conduction and convection losses.

Fuel composition and calorific value

In general the calorific value of the fuel is taken from utility bills or provided by the supplier. For liquid fuels the CH (carbon/hydrogen) ratio may be requested although it usually has to be inferred for natural gas. This is usually done on an infrequent basis, and thus variation in the fuel quality will not be known. A change of ±1% in the hydrogen ratio value will result in a change in calculated efficiency of 0.5 to 0.75% depending on fuel type. A similar variation in the carbon ratio value affects the calculated efficiency by less than 0.2%.

The thermophysical property variation of the fuel may also have a small but significant effect on errors, especially in the instance where the fuel is pre-heated (e.g. heavy fuel oil) by up to 60 °C from the reference temperature but is not accounted for in the calculations.

Direct or fuel-to-steam efficiency

This aspect is covered in BS 845-2:1987 and is referred to as the comprehensive procedure. In its simplest form, as determined in the majority of boiler installations, it merely consists of the measured energy output of the steam divided by the energy input derived from a knowledge of the fuel and its flow rate.

$$\eta = \frac{(m_{Steam}\ h_{Steam}) - (m_{Steam}\ h_{Feedwater})}{m_{Fuel}\ \delta H_{Fuel}}$$

m = mass flow rate (kg/s)
h = enthalpy (J/kg)
δH = enthalpy of combustion (J/kg)

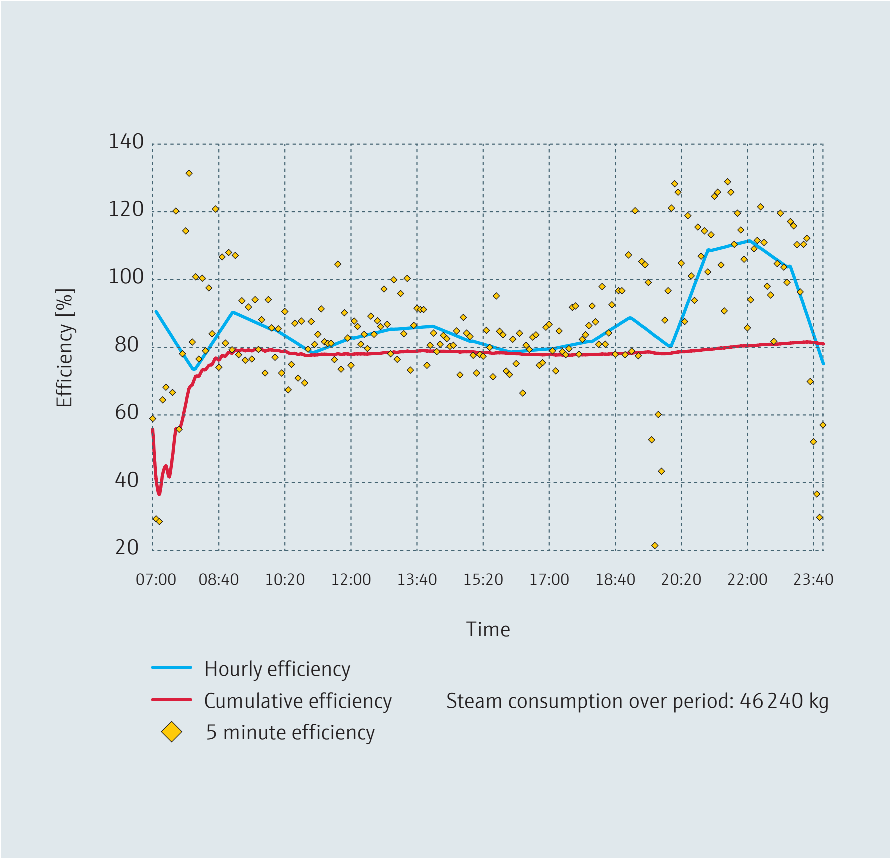


Figure 26: Fuel-to-steam efficiencies with varying time spans.

It is important to note that many processes require the heat content of the steam and not the steam itself. Thus for many installations accurate knowledge of this is as important as knowledge of the steam quality.

The fuel-to-steam efficiency is not suitable for instantaneous or short-time period measurements. This is due to the fact that the energy input rate does not necessarily match the energy output rate. The burner always follows the demand. This is especially true of stored energy units such as fire-tube boilers which are capable of delivering a large amount of steam at constantly reducing pressure – even when the burner is not firing. Figure 26 plots actual fuel-to-steam efficiencies over both five minute and one hour intervals and compares it to the same data used to calculate a cumulative efficiency. The five minute data is spread over nearly the whole range of the vertical axis, i.e. 0 to 140% and is meaningless. The data points in excess of 100% will coincide with periods when steam is flowing but the burner is not firing while the data points with very low values will correspond to periods when steam flow is very low but the burner is firing in order to restore pressure. It should be noted that gas-fired burners may take several minutes to go through the start-up cycle. If the steam demand is high during this period, the efficiency will be vastly overstated.

The hourly interval data is also meaningless as the lag between the demand and the firing is still too great. This results in the wide variation and overstatement of efficiency. The cumulative efficiency however, is of much more value. The data used to create Figure 26 was collected at five minute intervals. Using the cumulative approach, the variation caused by the burner lag is of little importance and a more realistic indication of efficiency is provided.

Data should always be considered in the context of the site from which it was taken. In this instance, the boiler was some ten years old, natural gas-fired and without an economizer. From a knowledge of the expected efficiency determined by the indirect

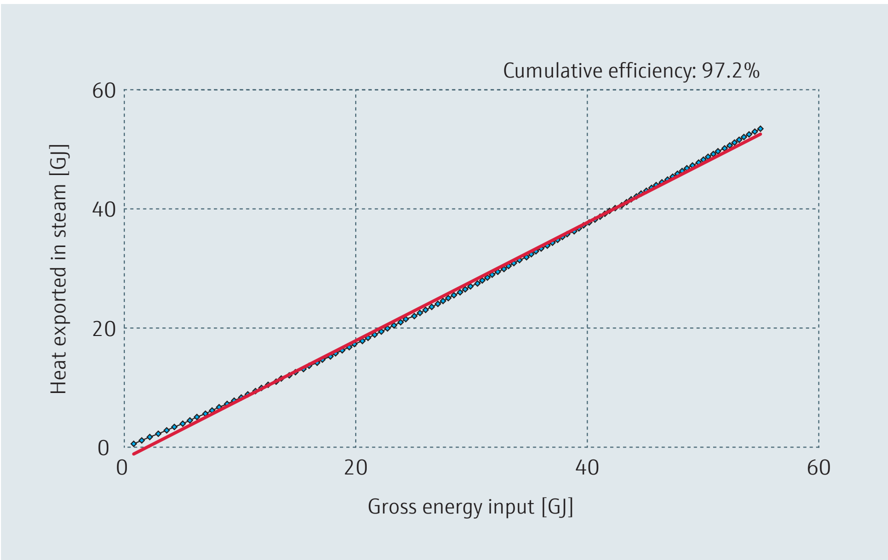


Figure 27: Unrealistic cumulative efficiency data.

method, the fuel-to-steam efficiency would be expected to be no more than about 77%, assuming ideal conditions and high quality feedwater. However, the efficiency of this unit is indicated at around 80%. Figure 27 shows the cumulative efficiency of the same boiler, the data acquired on a different day of the same week. Over this day, the cumulative efficiency is 97%. This in itself is clearly impossible, and is moreover markedly different to the 80% of Figure 26.

Considering first the discrepancy between the two efficiencies, the possible causes would either be related to the boiler or to the metering. In order to achieve a variation of this magnitude, the boiler operating characteristics would have to change by an amount so large as to be completely unrealistic. This then leaves consideration of the metering. It was, in fact, found that there were numerous problems with both the steam and gas meters which rendered the data valueless.

It should be noted, however, that the company from which this data was taken accepted it without question. **This is a common problem: there is a lack of understanding of metering, its installation and interpretation of the data.**

For many companies trending information is more important than absolute accuracy. Thus measurement, if accurate, of just the steam and fuel is adequate for this purpose. The requirement of BS 845-2 to determine the boundaries of the system and accurately measure within them is unnecessary. When analysing data, many companies are concerned with their fuel costs rather than the steam data in itself. They convert and restate the data on an energy and then financial basis. This is a source of error more common than one would suspect.

The thermophysical properties of steam can change markedly as it passes through its distribution system. Some companies do not actually know what the optimum steam quality for their particular purpose is. Figure 28 presents data for an industrial process, the steam being supplied by fire-tube boilers. The demand profile, instantaneous and with very large fluctuations represents a worst case operating scenario for this type of boiler. As can be seen the instantaneous peak demand often exceeds the boiler rated capacity. This was a result of installing extra process equipment without regard to the supply system. The steam supplied after the production increase was extremely wet, and carryover was a major problem to the extent that the water quality inside the boilers was uncontrollable; the boiler also frequently locked out due to priming events. This had the effect of increasing production times with major cost implications for the company.
Note: It is often the case that the downstream cost implications are much greater than the operating costs of the boiler. Gains from improving a distribution system can far outweigh gains from improvements in the boiler house.

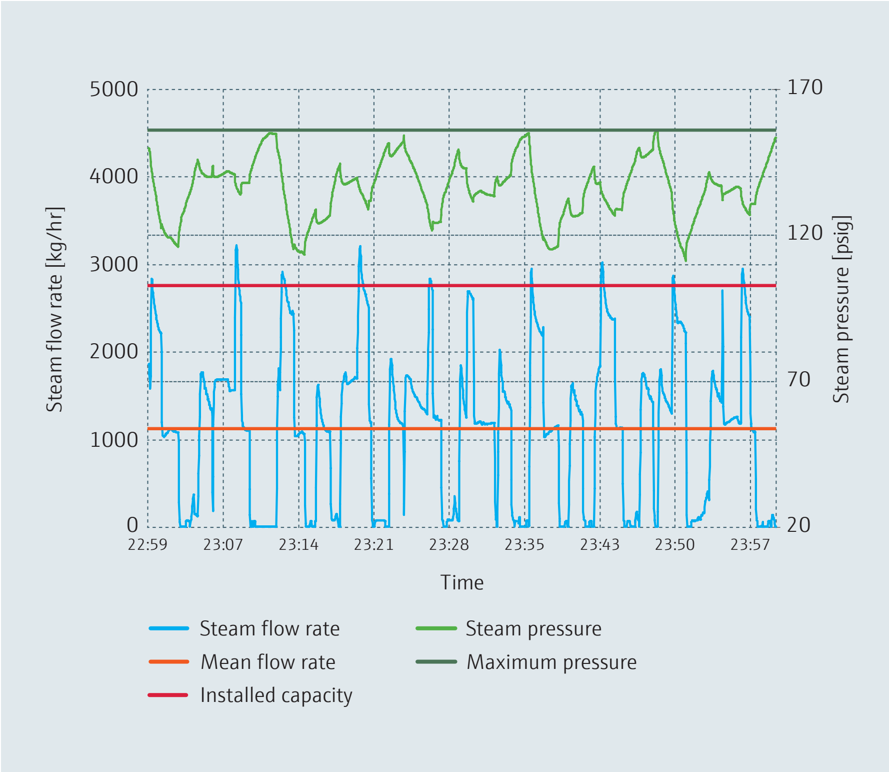


Figure 28: Process data with instantaneous demand profile.

Note that in the situation where steam demand is instantaneous, the meter set up can introduce significant errors into the measurements. A meter set to indicate flow rate with extended collection periods, for example, might cause significant rounding errors, in this instance, if the data points collected represent instantaneous rather than averaged amounts.

Due to the carry-over and extreme wetness of the steam, the data is again meaningless from the point of view of efficiency measurements. However, it does provide much useful information regarding the process, even when taken in isolation. The peak demands indicate that the user has a potential problem with priming and carry-over with consequent risks of low-water cutout and water hammer. The large pressure drops indicate large variations in both the steam flow rate and its quality.

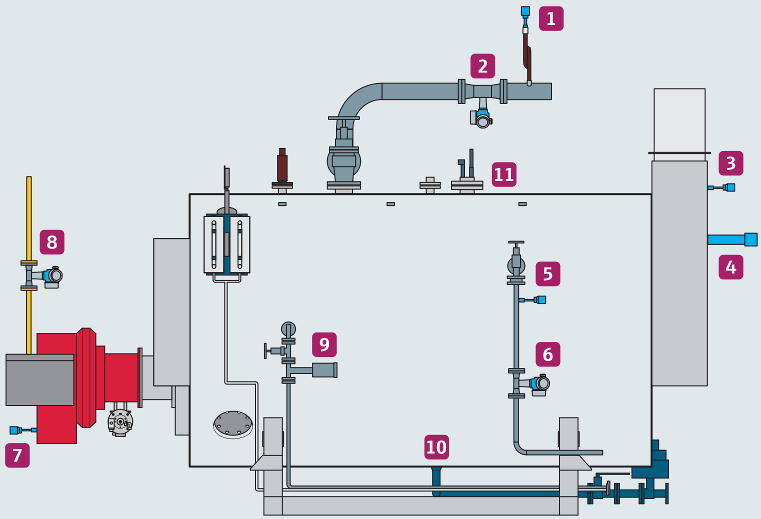
The data also shows that the boiler is being cyclically thermally stressed which – if combined with other causal factors – could be (and indeed has been) the cause of a dangerous incident.

Many users do not appreciate and therefore fail to use data gathered for efficiency purposes for monitoring with regard to operational safety. In the above case, for example, it would be very simple to link the pressure transmitter to an alarm or control point to give an additional, remote warning/control of boiler pressure.

Boiler management and control

Typical instrumentation for boiler management

Pressure and level instruments are used for the primary control of a boiler and indeed the boiler can be operated with this minimum, albeit sub-optimally. For optimal control of the boiler, flow, temperature and analytical instruments are also required and in addition further pressure and level instruments (Figure 29). The following text briefly describes the types and functions of the various instruments.



1. Pressure transmitter
2. Steam flowmeter
3. Exhaust temperature meter
4. Oxygen content measurement of exhaust gas
5. Feedwater temperature
6. Feedwater meter
7. Combustion air temperature measurement
8. Fuel meter for liquids or gas
9. Conductivity probe (TDS blow-down)
10. Bottom blow-down (sludge)
11. Pressure transmitter
Temperature sensor
Continuous level measurement
Level switches

Figure 29: Boiler control with various instruments.

Steam flowmeter (Fig. 29/2)

- When fitted to the steam main of a boiler they can be used for troubleshooting in conjunction with a pressure transmitter, as well as for proportional dosing of chemicals (if required) into the steam header.
- When combined with a feedwater meter blowdown calculations are possible.
- When combined with a feedwater and fuel meter direct efficiency calculations are possible.
- A steam meter can also be used in conjunction with the level controls and a feedwater meter in a three-element anticipatory control loop to optimize the boiler operation. This can reduce the pressure fluctuations and also reduce the negative effects of priming.
- Outside the boiler house steam meters are often used for process control. Care must be taken to adequately compensate volumetric meters when fitted after a PRV. Otherwise typical errors of greater than 20% (temperature only compensation) or approximately 4% (pressure only compensation) in the mass flow may occur due to local superheating. A pressure transmitter fitted to a steam meter downstream of a PRV can also be used to condition monitor the PRV.

Pressure transmitter (Fig. 29/1)

- A pressure transmitter should be fitted to the primary steam meter to ensure that it is correctly compensated. If the crown valve on the boiler has only been partially opened the steam downstream may be superheated.
  Note: superheated steam as a result of a partially opened crown valve is an unwanted occurrence.
- A pressure controller and separate limiter are always fitted directly to the boiler and linked to the burner for control and safety purposes. These are, however, normally supplied and fitted by either the burner or boiler manufacturer.
- Knowledge of pressure in the distribution system is useful for diagnostics such as the correct and safe operation of a PRV.

Fuel meters (Fig. 29/8)

- These are used in the calculation of total fuel consumptions and costs, as well as for determination of direct and indirect efficiencies.
- Fuel meters should also be used for calculating specific fuel consumption, e.g. Nm^3 gas per tonne of steam or cost/unit of product.

Feedwater meter, with temperature compensation (Fig. 29/6)

- To determine blowdown and direct efficiencies when combined with a steam flowmeter.
- For proportional dosing of chemicals into the boiler. This reduces under or over dosing and can help minimize corrosion inside the boiler, excess chemical consumption or excess blowdown.
- A feedwater meter can also be used in conjunction with the level controls and a steam flowmeter in a three-element control loop to optimize the boiler operation.
- Certain types of meter such as electromagnetic flowmeters possess the capability to output additional information such as the conductivity of the feedwater. This could be beneficially used to warn of a change in the conductivity which would indicate a problem with either the primary water treatment equipment or contamination of the condensate.

Feedwater temperature transmitter (Fig. 29/5)

- For a correct compensation of a feedwater meter.
- The temperature of the feedwater is important. Too low a temperature may result in thermal shock and the formation of cracks in the pressure vessel. This indicates an excessive oxygen content which in turn causes corrosion and a causal factor in catastrophic failures.

Incoming combustion air temperature measurement (Fig. 29/7)

This is used in the calculation of indirect efficiencies.

Oxygen content of the exhaust gases (Fig. 29/4)

This equipment is used to control the burner trim and can also be used for indirect efficiency calculations. It is unlikely that a standalone device which is not used as part of the burner control will be cost effective. The oxygen content is required for calculation of indirect efficiencies.

Exhaust temperature transmitter (Fig. 29/3)

- A boiler's exhaust temperature is a function of the boiler design, firing rate and fluid temperature. If the exhaust temperature rises independently of these factors it indicates that the heat transfer inside the boiler has deteriorated. A rapid increase in exhaust temperature indicates a burner-side problem. A slow increase indicates scale formation on the water side which can result in expensive repairs or in extreme cases a catastrophic failure.

- A measurement of the exhaust temperature is also required for the calculation of indirect efficiencies. A 20 °C change in exhaust temperature approximates to a 1% change in the indirect efficiency of the boiler.

Contamination of condensate

This is site specific, dependant on the processes for which the steam is being used. Contamination of condensate can cause foaming, carry-over, internal damage and catastrophic failure of a boiler. This should be discussed with the end user and appropriate monitoring equipment fitted as well as a means of disposing of unwanted contaminated condensate. Common measurements on condensate lines include conductivity, turbidity and pH.

Boiler level controls

Many boilers in Europe are fitted with level controls (Fig. 29/11) which comprise a capacitance probe for pump control and a capacitance and/or a conductivity probe for the limiting devices. These are fitted with protection tubes, and in general they are reliable and work well. However, they all rely on the conductivity of the water inside the boiler to function correctly and there is a residual risk in situations where foaming occurs. This may be due to some failure of the total dissolved solids (TDS) controller or unwanted contamination.

Alternative types of probe such as guided radar, vibrating fork, positive displacement and differential pressure are available at reasonable cost. By employing instruments with different principles of operation, technological diversity is brought into the installation and the risk posed by certain known and foreseeable events is reduced. It should however be noted that all controls possess vulnerabilities, and selection of appropriate controls should be made on a site-by-site basis with respect to (inter)national standards and norms.
Note: On sites where very pure water is used to feed the boiler, e.g. from reverse osmosis plants, there have been instances where the capacitance or conductivity metering systems have failed to operate correctly when the boiler is first filled with water as its conductivity is too low.

Hotwell temperature

The hotwell is a store of water for supply to the boiler. It should be maintained at a high temperature to minimize thermal shock and the oxygen content of the feedwater. Its temperature should be measured and compared to the feedwater temperature as a discrepancy may indicate a problem with the hotwell. The temperature measurement may also be used to control any steam injection to the hotwell.

Hotwell level

A hotwell is usually sized to hold enough water for the boiler(s) it supplies for them to steam at maximum forcing rate for an hour. Automatic level monitoring permits an orderly shutdown in the event of some upstream failure.

Total dissolved solids (TDS)

The feedwater to the boiler contains both impurities and chemicals. However, the steam exported is relatively pure water, consequently the impurities become concentrated inside the boiler. If the concentration of dissolved solids is too high, foaming will occur. Excessive foaming can cause carry-over of solids into the distribution system, or in severe cases result in a failure of certain type of level controls as described earlier. The TDS of the boiler is controlled by measuring it (Fig. 29/9) and blowing down the boiler if it is too high. This reduces the TDS inside the boiler to an allowable level. TDS blowdown systems are frequently sold as packages with all the necessary controllers, valve and the sensor contained within.

Along with the dissolved solids, water which has been heated up to the saturation pressure (and therefore has a high heat content) is also removed from the boiler. It is important to control the TDS effectively in order to minimize blowdown rates and the cost of blowdown.

The European boiler market

The European market is fragmented along the following historical lines: national borders, design codes, regulatory frameworks, economic spheres of influence and language. The UK forms one of these fragments. With the advancement of the European Union (EU) and its process of harmonization, some of these divisions are being reduced, e.g. the partial replacement of national design codes with pan-European versions. However, they are still significant and will remain so for the foreseeable future.

The UK market, in common with the rest of Europe, is very risk averse and conservative. MTHS boilers of the current, packaged form have been manufactured since the 1930's and can trace their lineage directly back to Trevithick's Cornish boiler of the early nineteenth century. The numerous catastrophic boiler failures leading up to the modern age has resulted in a very safety-conscious, highly regulated industry. This naturally results in a slow development process and a resistance to change.

Improvements to the design of the pressure parts of boilers over the last twenty-five years have been driven by improvements to the design codes with respect to safety and reliability rather than increases in thermal efficiency. The major advances in thermal efficiency during this time have been mainly due to the burner manufacturers with innovations such as fuel-air mapping and servo-controlled, fully modulating dampers. Improvements to the internal heat transfer and thus to thermal efficiencies have been minimal over the period. Current industry norms are around 80% for natural gas and 83% for diesel, indirect thermal efficiencies.

Environmental concerns and the large rises in fuel costs over the last decade, have led to a vastly increased interest in the market for higher fuel efficiency. These two issues have led some manufacturers in different parts of the world to independently pursue methods of increasing the amount of heat transferred in the convective passes of MTHS boilers thus improving efficiency. Typical gains are of the order of 1 to 2%, additional to the figures stated above. Although this is a relatively small amount, it is a very useful increase when the fuel consumption rates of large heavily used steam boilers are considered, e.g. a 10 MW (net) steam boiler burns around one tonne of fuel oil per hour at full load. Further work is currently being undertaken around the world to improve boiler efficiency and reduce the emissions of undesirable pollutants. In order to maximize gains however, there will need to be a greater emphasis on boiler control with attendant increases in metering and monitoring which will be discussed in more detail later on.

Regulatory structure

The following remarks relate in detail to the United Kingdom. That said, the regulation of steam boilers worldwide generally follows the following principles: The boilers are manufactured to a recognized standard with external oversight during the manufacturing process. Once installed and operational the boilers are inspected internally, usually on an annual basis coupled with a more in depth inspection requiring non-destructive testing at set intervals, often five to ten year intervals depending on the jurisdiction. The manning requirements for boiler operation are also regulated on a prescriptive or goal based approach depending on the jurisdiction.

Design and construction approvals, and permits to operate in the UK are regulated by the insurance industry with guidance from relevant government departments and statutory bodies. This is quite unlike in most of continental Europe, where approval for the construction and use of steam boilers is obtained through government agencies or other statutory bodies.

The system for design approval operated by the insurance industry is predicated on active authorization for all aspects of pressure parts and systems. No component may be introduced into a pressure vessel without approval, not only of the component, but also of the system containing that component. This also applies to retrospective alterations and repairs.

All steam boilers manufactured in the EU must conform to the Pressure Equipment Directive (PED). This is a general directive covering all pressure equipment and is a legal requirement. There are additionally design codes specifically for MTHS boilers such as the national UK standard PD 2790 and the pan-European EN 12953.

Although PD 2790, EN 12953 or other design codes are not legal requirements in the UK, they are to some practical extent mandatory as the insurance industry will not issue approvals unless conformance to a known standard has been satisfactorily demonstrated.

PD 2790 and EN 12953 are prescriptive standards relating to the structural integrity of the pressure vessel part of the shell. They are not performance standards and do not state any minimum efficiencies to be achieved. The approach of these standards is based on explicit statement of approval, e.g. specified grades of material for use in the varying parts of the shell. This has resulted in the anomalous situation of boiler tube still being manufactured in accordance with BS 3059:1990, even though that standard is withdrawn and replaced by BS EN 10217-1:2002. This is because PD

2790 explicitly permits the use of BS 3059, whereas BS EN 10217-1:2002 would require a formal derogation from the inspection body.

The use of alternative material grades, e.g. for boiler tubes, is permitted by PD 2790, however the manufacturer must satisfy the inspecting authority that the material will match or exceed the performance requirements of both the specified tube standard (BS 3059) and the shell construction standard. The process of demonstrating conformity of an alternative grade is both lengthy and costly as numerous additional tests over an extended period of time may be required. In practice this means that very few alternative materials have been adopted or used by UK manufacturers in boiler shells.

It should be noted that approval by the inspection authority is expensive. Design approval will typically cost in the region of £1000 per boiler. Each individual boiler must be individually approved; class approval is not an option. Any design change to an existing shell design must also be approved at extra cost to the manufacturer. Even minor changes to a shell such as the repositioning of a stabbing are classed as design changes requiring approval. The pressure parts are also physically inspected during production, also at the cost of the manufacturer.

Market structure

The retail market for MTHS boilers (<20 MW) in the UK is estimated at a stable 300 to 400 units per year with a total population of 5000 to 10,000 units. The market is thought to be static, slowly increasing for smaller boilers (<2 MW). Above this size the picture is one of gentle decline. The major cause of this is the ongoing shift of the manufacturing base out of the United Kingdom, although the market is difficult to assess precisely as industry-wide figures do not exist.

The market is considered to be strong in certain areas, for example: food and beverage manufacturing, pharmaceuticals and "state" contracts such as Private Finance Initiate (PFI) funded hospital modernization programs. The recent volatility in fuel costs whilst having little apparent affect on overall sales volumes have resulted in a move to higher specification and more efficient equipment.

Figure 30 presents the conservatively estimated MTHS boiler market by thermal output for 2007. Total sales are estimated at 300+ units, the balance being made up mainly by vertical shell boilers, generally less than 2 MW capacity. The market above 8 MW net thermal output is very variable in the context of the small numbers sold due to the few sites operating boilers of this output. Additionally, whilst the thermal output range 0.15 to 2 MW covers seven different models of boiler by

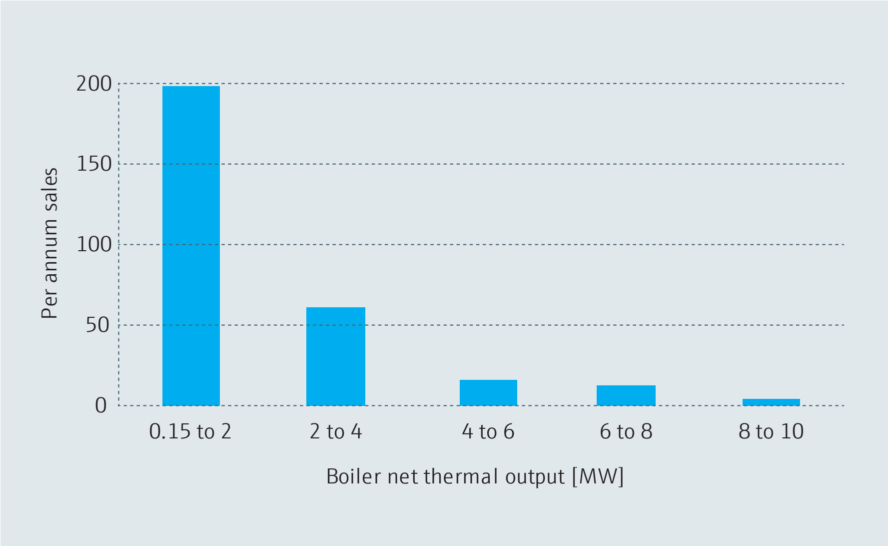


Figure 30: Estimated MTHS boiler market in the United Kingdom for 2007.

output, the range between 8 and 10 MW includes only two. It should also be noted that boiler sales do not exactly reflect the size distribution of the installed boiler population for the following reasons:

- The downsizing of installed plant due to reduced steam requirements is a long established trend.
- The stronger market sectors generally require small or mid-range boilers (2 to 6 MW).
- Large units are often being replaced by pairs of mid-range boilers caused by a policy shift by users to prefer pairs of smaller boilers rather than a single, large one.

The United Kingdom boiler population includes a large number of very old boilers with more than 20 years of operation. Whilst a number of these are maintained for standby purposes, many are still used as the prime steam generator. These are invariably less efficient than modern boilers fitted with the latest energy saving equipment. Additionally, many are also oversized for the current steam demand and will also suffer from poor quality and haphazard steam distribution systems – a consequence of changing requirements over the long life of the boiler. Therefore, there is much scope for reducing the fuel and other costs for many boiler operators.

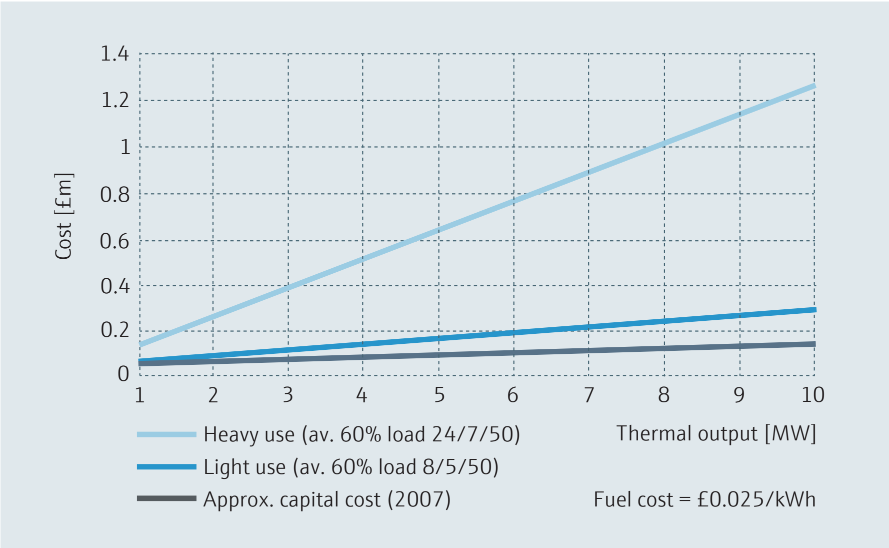


Figure 31: Fuel versus capital costs of industrial boilers.

When considering steam boiler costs, a lifetime cost approach should always be used. However the United Kingdom market is very sensitive to the capital cost aspect, often to the detriment of the lifetime cost. Figure 31 shows the capital cost of high specification boilers compared to the per annum fuel cost. The light-use curve represents a standard cost illustration that may be presented to a customer. The intensive use represents the maximum possible fuel cost of operating the boiler for a year. Although it would be very unusual to find an installation operating at maximum capacity 24/7, many intensive installations will operate at a significant fraction of this. It should be noted that these illustrations do not include other consumable costs such as water, electricity and chemicals which will have a significant effect on the overall operating cost and may easily exceed the capital cost of the boiler in a year.

The fuel to capital cost per annum multiple ranges from 2 to 11 for the small boilers and 5 to 25 for larger units. When considered over the standard 20 year lifetime illustration of a boiler the capital cost represents around 1% of the lifetime fuel cost (without compensating for inflation).

Although there are a small and growing number of operators who understand the benefits of optimising the product lifetime costs there are still too many who focus

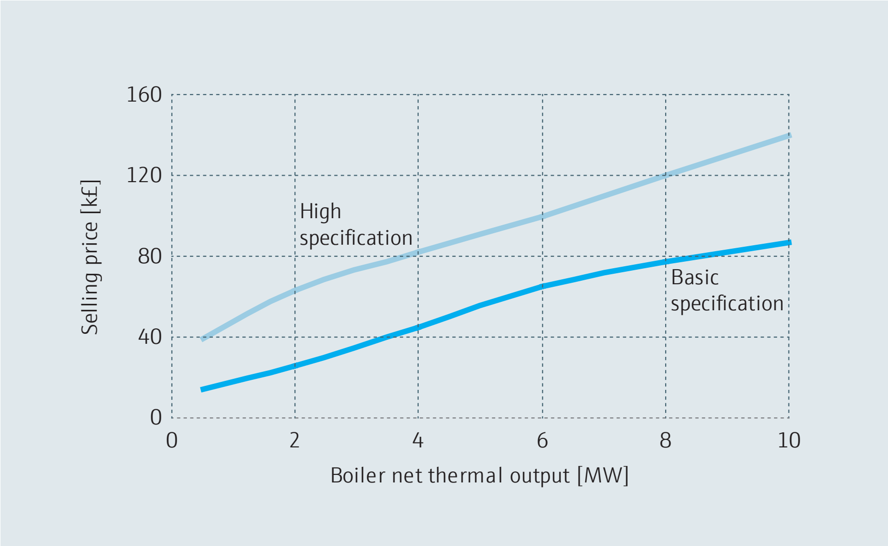


Figure 32: MTHS boiler cost variation (for 2007) due to specification enhancement.

on capital costs with short payback periods. This is especially so at the smaller end of the market when considering energy-saving enhancements to a basic boiler package. Boilers are often a distress purchase. Rather than being planned for operational gain, the purchase is initiated by the unreliability of the existing plant or impending (or failure to pass) 5-year non-destructive test. The smaller end of the market also often lacks, or is unable to access, capital for various reasons.

One difficulty occasionally experienced by all industrial boiler suppliers is where the purchase and installation of the boiler and ancillaries is undertaken by a company on a fixed price contract, but is not responsible for the boiler operation. It is thus in the interest of the purchasing company to reduce the specification of the boiler plant to maximize profit at the expense of the operator's running costs. Unfortunately this scenario occurs all too frequently.

Figure 32 presents the cost differential between a basic and high-specification horizontal steam boiler. The cost increase factor ranges from a maximum 2.6 for a small unit to 1.6 for a large one. Considering this in relation to fuel costs, a high specification gas-fired boiler would be expected to reduce fuel consumption by a minimum of 5% over a year. This equates to £8.5k for a small boiler rising to £40k for a large boiler over one year's operation based on the light-use pattern presented in Figure 32. The payback of the high specification package increment would thus be 36 months for a small boiler falling to 15 months for a large unit.

The 5% efficiency gain represents a minimum, taking the operating profile into account, this may easily increase to 10 or even more. As an example of how this is achieved, the heat flux to atmosphere during a purge cycle is in excess of 4.5% of the maximum continuous rating net thermal input to the boiler. This equates to a reduction in fuel consumption of about 1% if five purge cycles per hour are eliminated by the use of a modulating instead of a hi-lo burner – not unreasonable at low load conditions. Again, this analysis is simplistic as it ignores incidental savings due to reduced electricity consumption (if a variable speed drive is fitted), and the potential downstream process. Benefits of a higher average steam pressure and improved boiler response to variable demand.

Another way of looking at the boiler capital cost is by the cost per kilogramme of steam. This tends to decrease with increasing thermal output as larger boilers are often operated more intensively. It is thus easier to justify the extra capital required to purchase energy saving devices for a large boiler than a small one.

A number of companies supply boilers into the UK market. These may be ranked in order of estimated market share: indigenous manufacturers (these may also sell boilers imported from their overseas manufacturing operations), foreign manufacturers with UK operations and importers – mainly from the continent, but also from other parts of the world.

The UK market itself may be sub-divided into the following categories:

1) Private Finance Initiative (PFI) contracts: Essentially government contracts for hospitals and other facilities. These are frequently very price sensitive and thus low margin.
2) Intensive industrial users with an installed capacity usually higher than 3 MW who almost always purchase the three-pass wetback type.
3) Small industrial users who traditionally favor the reverse flame and vertical types.
4) Other, e.g. museums reverse operating steam engines.

Very small horizontal boilers are only available in the reverse flame form as it is not possible to manufacture three-pass wetback boilers due to space restrictions. The three-pass wetback and reverse flame ranges overlap in the net thermal input range from 0.8 to 3.3 MW. One consequence of the recent large increases in fuel prices has been that the three-pass wetback has taken market share from the reverse flame type due to its inherent greater efficiency.

One increasing trend, albeit of very small numbers, is purchase of containerized boiler houses. These offer several benefits to the customer: reduced on-site installation times, minimization of plant disruption, "all new" boiler installations offering maximum efficiency gains, the ability to move the boiler house quickly and with minimum disruption as well as increased residual values. They are often favored for new installations or where a complete new boiler house is required. Containerized boiler houses are also beneficial to the supplier in that they maximize the revenue earned whilst again minimising on-site time and cost. Installation is a significant – if site specific – cost of boiler plant, and is the area most open to unforeseen or miscalculated costs. It is not uncommon for the total package cost to be double or even treble the cost of the boiler itself.

Appendix

Nomenclature

All units are as indicated unless stated otherwise in the text.

Symbol	Meaning	SI unit
A	cross-sectional area	m^2
a	dimensionless constant	
C	dimensionless constant	
C_f	friction factor (4*f*)	
c_p	specific heat capacity	J/kg K
CSA	cross-sectional area	m^2
d	diameter	m
δp	differential pressure	N/m^2
δT_{LM}	log mean temperature difference	K
f	friction factor	
g	gravitational constant	m/s^2
G	gas mass velocity	kg/s m^2
GCV	gross calorific value	J/Nm^3
h	specific enthalpy	J/kg
H	enthalpy	J
k	thermal conductivity	W/m K
KE	kinetic energy	J
l	length	m
m	mass	kg
$\dot{m}$	mass flow rate	kg/s
M	molar mass	kg/kmol
NCV	net calorific value	J/Nm^3
Q	heat transfer	J
$\dot{Q}$	heat transfer rate	W

$\dot{Q}_m$	heat transfer rate per meter	W/m
P	pressure	Pa (Nm^{-2}) mbar bara (barabsolute) barg (bargauge)
PE	potential energy	J
r	radius	m
R	molar (universal) gas constant	kJ kg/mol K
t	thickness	m
T	temperature	K °C R
u	velocity	m/s
U	overall heat transfer coefficient	W/m^2 K
w	specific work	J/kg
W	work	J
X	characteristic length	m
z	height above datum	m

Greek symbols

α	local heat transfer coefficient	W/m^2 K
β	coefficient of expansion of a fluid	
ε	emissivity	
ϕ	effectiveness	
η	efficiency	%
μ	dynamic viscosity	kg/m s
ν	kinematic viscosity	m^2/s
σ	Stefan-Boltzman Constant	W/m^2 K^4
ρ	density	kg m^{-3}
χ	mole fraction	

General subscripts / superscripts

b	bulk temperature
f	film temperature fluid (e.g. h_f enthalpy in steam tables)
f_g	fluid to gas (e.g. h_{fg} enthalpy in steam tables)
g	gas (products of combustion) vapor/gas (e.g. h_g enthalpy in steam tables)
gr	gross
i	inlet
m	exponent
n	net/number
o	outlet
s	steel (wall material)
t	measured temperature
tc	thermocouple
sat	saturation point
w	wall temperature
∞	free stream condition

Dimensionless numbers

Dimensionless numbers are used extensively in many areas of engineering. They are ratios of physical quantities. They can be used in various ways, for example they can provide information regarding the properties of the system under question. Certain dimensionless groups can be related to each other empirically and used as design tools.

Grashof Number (Gr)

$$\mathrm{Gr} = \left(\frac{\beta\ \mathrm{g}\ \rho^2\ \mathrm{d}^3\ \Delta\mathrm{T}}{\mu^2}\right)$$

β	coefficient of expansion of a fluid
g	gravitational constant
ρ	density
d	diameter
ΔT	temperature difference (between the surface and the bulk of a fluid between local and ambient temperature)
μ	dynamic viscosity

The Grashof number is the ratio of buoyancy forces to viscous forces. It is used for systems involving natural convection. It is frequently used in calculations for heat transfer.

Nusselt Number (Nu)

$$\mathrm{Nu} = \left(\frac{\alpha\ \mathrm{d}}{\mathrm{k}}\right)$$

α	local heat transfer coefficient
d	diameter
k	thermal conductivity

The Nusselt number is used in heat transfer. It is the ratio of heat transfer by convection and conduction to conduction only. It can be related empirically, for example, to the Reynolds and Prandtl numbers (Nu = *fn* [Re, Pr]) for forced convection, or to the Rayleigh and Prandtl numbers (Nu = *fn* [Ra, Pr]) for natural convection.

Prandtl Number (Pr)

$$\mathrm{Pr} = \left(\frac{\mathrm{c_p}\ \mu}{\mathrm{k}}\right)$$

c_p	local heat transfer coefficient
μ	dynamic viscosity
k	thermal conductivity

The Prandtl number is the ratio of the rates of molecular momentum diffusivity to thermal diffusion. It is related only to the thermophysical properties of the fluid in question. It gives information on the mechanism by which heat is diffused through a fluid.

Reynolds Number (Re)

$$Re = \left(\frac{\rho\, d\, u}{\mu}\right)$$

ρ	density
d	diameter
u	velocity
μ	dynamic viscosity

The Reynolds number is the ratio of inertia to viscous forces. It is used in many aspects of fluid mechanics and heat transfer. It gives information on the flow characteristic, for example if the Reynolds number of a moving fluid is less than 2000 – then the flow is said to be laminar (smooth).

List of abbreviations

ASME	American Society of Mechanical Engineers
BS	British Standard
BLEVE	Boiling liquid expanding vapor explosion
CIP	Cleaning in place
EN	European Standard (Norm)
GCV	Gross calorific value
HHV	Higher heating value
LHV	Lower heating value
MCR	Maximum continuous rating
MTHS	Multi-tubular horizontal steam (boiler)
NCV	Net calorific value
PED	Pressure Equipment Directive
PRV	Pressure reducing valve
PD	Published Document
RO	Reverse osmosis
TDS	Total dissolved solids
WSE	Written Schemes of Examination

Glossary

Absolute pressure: Pressure measurement whose datum is a perfect vacuum. See also *gauge pressure*.

Adiabatic process: The adiabatic process occurs without the transfer of heat across the system boundary. The adiabatic flame temperature, for example, is that temperature which would be reached by the flame if there was no heat loss to the surroundings.

Blowdown pit: A blowdown pit consists of a covered pit partially filled with water which performs the same function as a *blowdown vessel*. They are more commonly found in older boiler houses as blowdown vessels are now preferred.

Blowdown vessel: Water which is blowdown from a boiler in the form of *dissolved solids* (see: TDS), *blowdown* or *bottom blowdown* is at elevated pressure and temperature. Both the temperature and pressure have to be dissipated safely before the blowdown can be safely disposed of. This is done in a blowdown vessel, i.e. a pressure vessel partially filled with water and possessing a large vent to atmosphere. When the blowdown enters the vessel some steam is flashed off and vented to atmosphere. The remainder mixes with the cold water and is discharged to drain. If blowdown rates are very high, the flash steam may be collected and reused. Blowdown vessels are rated for pressure and are matched to the boiler house for capacity. An alternative to a blowdown vessel is a blowdown pit.

Boiler horsepower (American): A measure of the useful output of a steam boiler. 1 boiler horsepower is equivalent to the generation of 15.6 kg steam from and at 100 °C in one hour (9.8 kW).

Bottom blowdown: As a steam boiler exports steam, it concentrates solids contained in the feedwater. Some of the concentrated solids sink to the bottom of the boiler. If allowed to build up they would interfere with its normal operation and may pose a hazard. They are normally removed on a daily basis by opening a valve located at the bottom of the boiler. The pressure inside the boiler then forces the solids out along with a quantity of boiler water. The blowdown is normally sent to a *blowdown vessel*.

Carryover: Droplets of liquid water, foam or solid particles are drawn up with the steam vapor into the distribution system.

Cavitation: A fluid close to or at its saturation temperature may contain bubbles of vapor. These frequently collapse of their own accord. The term for collapsing bubbles is “cavitation”. Cavitation is sometimes observed on the pressure side of a feed pump in boiler houses. If the feedwater is hot enough and there is not enough head, the water will be close to its saturation temperature and steam bubbles will be present. As the water passes through the pump, it moves from the low pressure side to the high pressure side. Due to the increase in pressure, the water is now unsaturated and the steam bubbles collapse instantaneously creating shock waves. These shock waves are capable of causing significant damage to the pump.

Condensate: Condensate forms inside the steam mains due to heat loss through the pipe walls. It is removed at frequent intervals using *steam traps* to prevent *water hammer* and maintain the quality of the steam. Indirect processes only utilize the latent heat of the steam so all the steam condenses back to the liquid phase. Condensate is normally returned to the boiler house for reuse. In some systems however the condensate is not recycled. This is frequently due to unavoidable contamination making the condensate unsuitable for reuse.

Critical point: The point at which the liquid, vapor and gaseous phases of a substance are in thermodynamic equilibrium. For water this occurs at a pressure of 221.2 bara and a temperature of 374.15 °C (647.30 K).

Crown valve: The main valve attached to a boiler through which steam passes through to the distribution system. It derives its name from its location on the top, or crown, of the boiler.

Deadleg: A steam line which is connected to the distribution system and maintained at pressure but which no longer serves any consumer.

Degrees of superheat: Superheated steam is commonly defined in terms of its pressure and degrees of superheat. The degrees of superheat is the temperature difference between the actual temperature of the steam and the saturation temperature at the steam pressure. Thus superheated steam at 10 bara and a final temperature of 199.9 °C would possess 20 degrees of superheat. (The saturation temperature being 179.9 °C).

Desuperheater: A piece of equipment used to reduce the temperature of superheated steam to its saturation temperature for the pressure at which it exists. Common examples are small turbines and Venturi water injection.

Direct efficiency: The ratio of useful heat exported from the boiler to the net or gross heat input. Comprehensive efficiency calculations may include the electrical energy consumption as well as the fuel consumption; sub-systems may also be accounted for.

Dry fry: The period during a low-water event/condition when the burner is still firing and the metal of the furnace exposed to steam rather than water. This causes an unwanted and potentially hazardous rise in the metal temperature.

Dryback furnace: A furnace in which *the reversal* cell is sealed using a refractory plug. No useful heat transfer to the *heat carrier* takes place through the refractory.

Dryness fraction: The fraction of wet steam which, if separated out, would consist of steam vapor only. A wet steam mixture comprising 95% by mass steam vapor and 5% by mass of saturated liquid would have a dryness fraction of 0.95.

Dry saturated steam: see *Saturated steam*.

Economizer: A heat exchanger used to recover waste heat from the exhaust gases of a boiler and transfer that heat into the feedwater.

Enthalpy: A term for the energy of a substance. The enthalpy of a substance is made up of the addition of its internal energy and the energy it possesses by virtue of its pressure and specific volume hence the distinct name for it. Enthalpy cannot be directly measured, however the change in enthalpy can be determined if the change in properties such as pressure and volume are known. Enthalpy is referenced to an arbitrary point. For water this is the *triple point* at 0.01 °C (273.16 K) and 0.006112 bara.

Entropy: A measure of the disorder of a system. As work is performed the entropy of a system tends to increase.

Fire-tube boiler: A boiler in which the furnace and heat transfer tubes are submerged by the water being boiled. The exhaust gases pass through the boiler inside the furnace and subsequent heat transfer surfaces, usually banks of tubes known as convective passes. The furnace and heat transfer surfaces are surrounded by a shell of sufficient strength to withstand the internal pressure – also known as “smoke-tube” boilers, boilers multi-tubular horizontal steam boilers (MTHS).

Flash steam (first type): Steam which is generated as a result of liquid water coming into contact with a surface which is hotter than the saturation temperature of the water. If the temperature of the hot surface is great enough, a flash steam explosion may result.

Flash steam (second type): Steam which is generated as a result of a pressure reduction. Liquid water at high pressure has a greater enthalpy than water at a lower pressure. If a saturated liquid at elevated pressure is suddenly reduced in pressure, the difference in enthalpy is given up by the liquid causing a small fraction of the liquid to boil off forming flash steam.

From and at 100 °C (F&A 100 °C): A measure of the useful heat output of a steam boiler – equivalent to the enthalpy of the change of phase of water at 100 °C and 1.013 bara. This measure is normally expressed in kg/hr. 1 kg/hr from and at 100 °C equates to 0.627 kW.

Gauge pressure: Pressure which is referenced to atmospheric pressure (i.e. 0 barg = atmospheric pressure at the time of measurement). Gauge pressure is the most common pressure measurement in steam systems. If required to be selected, the reference pressure used in instrumentation is frequently a fixed value chosen at the discretion of the end user, e.g. one standard atmosphere 1.01325 bara, or, more simply, 1.0 bara.

Gross Calorific Value (GCV): The total energy content per unit mass of a fuel.

Heat carrier: The working fluid of a boiler.

Hotwell: A tank of water maintained at around 80 °C which is used to supply water to the boiler. The water is maintained at temperature to reduce thermal shock to the boiler when it is injected via the feed pump and also to reduce the dissolved oxygen content of the feedwater.

Indirect efficiency: The determination of thermal performance by the assessment of the thermal losses and the measured thermal input or output.

Isenthalpic: A process in which there is no loss of *enthalpy*. In practice this is an idealized state. Therefore a device for which an isenthalpic efficiency is quoted is comparing its actual performance to that if it were perfectly efficient.

Isentropic: A process which takes place with constant *entropy*. In practice this is an idealized state. Therefore a device for which an isentropic efficiency is quoted is comparing its actual performance to that if it were perfectly efficient.

l-v (liquid–vapor) mixture: A substance which is composed of two phases in equilibrium, e.g. *wet steam*.

Latent heat: Heat which is added to or taken away from a fluid at constant temperature, which results in a change of phase of the fluid, e.g. wet steam condensing back into liquid water.

Low-water condition: Low-water conditions occur when the free surface of the water inside a steam boiler has dropped so that the heating surfaces are exposed to steam vapor rather than liquid water. The temperature of the gases on the fireside of the boiler can easily exceed 1000 °C and the surfaces exposed rapidly heat up. If they become hot enough, the steel loses its strength and is plastically deformed, i.e. pushed inwards by the pressure on the waterside. This situation is called a furnace collapse. In extreme cases it can cause a rupture in the pressure vessel resulting in an explosion.

Maximum continuous rating (MCR): The maximum capable output of a steam boiler under continuous operation, often expressed in terms of kg/hr from and at 100 °C.

Mollier chart or diagram: A Mollier chart is a graphical representation of the thermophysical properties of steam (Figure 33). The underlying data are the same as tabulated in *steam tables*. A Mollier chart plots the thermodynamic properties of a substance: entropy, enthalpy, temperature, pressure, specific volume and *dryness fraction*. If any two of the thermodynamic properties are known, then the others can be read from the diagram.

Net Calorific Value (NCV): The total energy content per unit mass of a fuel minus the energy content of the latent heat of evaporation of the water formed as a consequence of combustion.

Phase diagram: A diagram, chart or graph which presents the various states of a fluid divided into regions (Figure 34): solid, liquid, wet, vapor (superheated) and gaseous (supercritical). The boundaries between these regions are commonly termed “phase lines”.

Priming: A situation in which the steam demand is so great that the free surface of the water inside the boiler rises up locally beneath the crown valve. It is strongly associated with *carryover*. In severe instances a slug of water can detach and enter the distribution system. Priming can also be known as *swell*.

Pressure reducing valve (PRV): Steam is distributed at a higher pressure than processes require it. Its pressure is reduced close to the point requirement using a PRV to provide steam of the correct, controlled pressure for the downstream processes. The operation of a PRV may cause a small amount of *superheat* downstream of the valve on saturated steam systems.

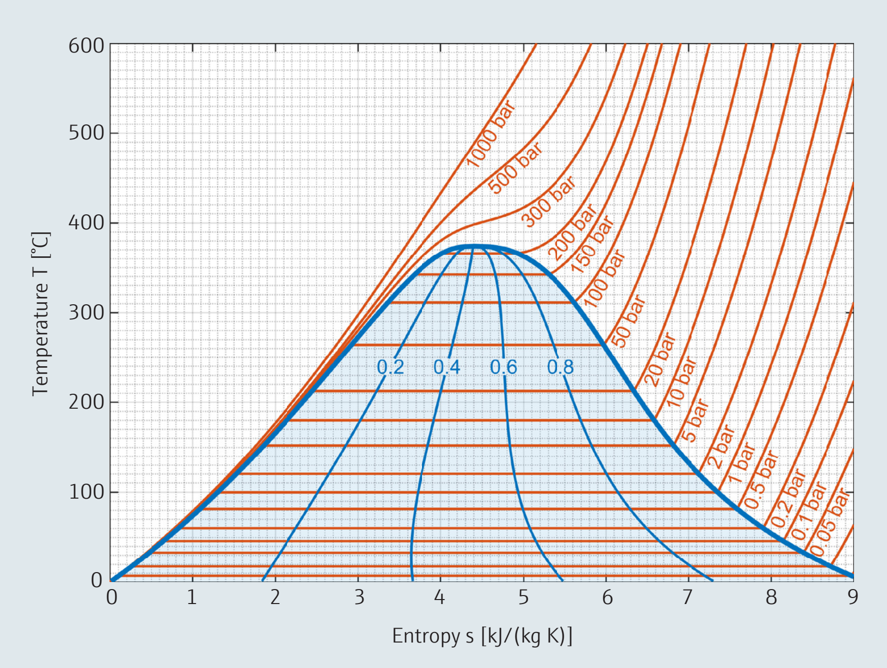


Figure 33: Mollier diagram (blue curves = dryness fraction).

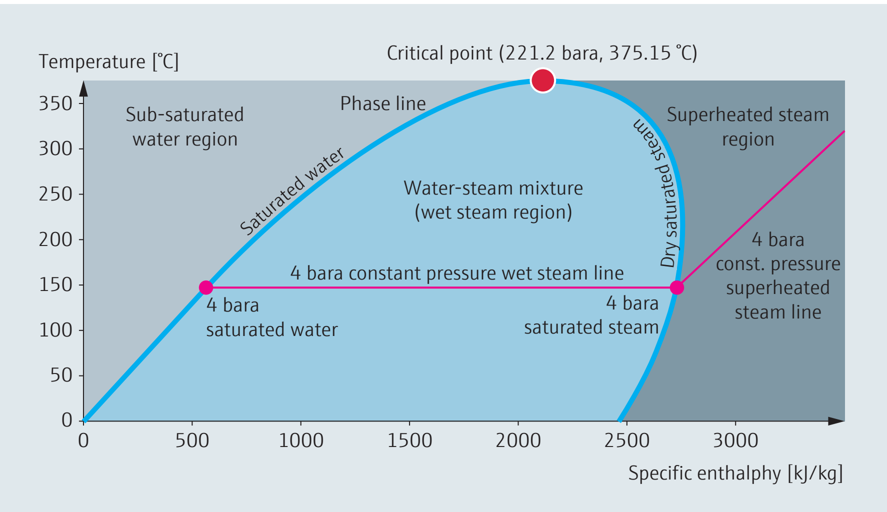


Figure 34: Simplified T-h phase diagram (Mollier diagram) with a 4 bar constant pressure line.

Reverse flame boiler: In this type of boiler the furnace is blank ended. The flame passes through the centre of the furnace. The hot gases impinge on the blank end of the furnace and in doing so are reversed in direction, passing back along the furnace wall prior to exiting the furnace.

Reversal cell: A *wetback* or *dryback* boiler is constructed with a chamber at the end of the furnace which causes the flow of combustion products to be reversed. These combustion products that pass through a bank of tubes in which further useful heat is transferred from the gases to the waterside.

Safety valve: A mechanical valve fitted to pressure vessels intended to prevent over-pressure events by venting fluid to atmosphere when its set point is reached. Safety valves are completely independent of other protection measures and are always fitted to steam boilers. They are also found downstream of pressure reducing valves in steam distribution systems.

Saturation point: The temperature and pressure at which a fluid cannot absorb or give up further heat without a phase change (see 4 bara example in Figure 34).

Saturated steam: Steam vapor which cannot absorb any additional heat without a consequent increase in temperature. Saturated steam is often called *dry saturated steam*. Saturated steam lies on the phase line between the wet steam and superheated steam regions of a *phase diagram* (see 4 bara example in Figure 34).

Saturated water: Liquid water which cannot absorb additional heat without a phase change occurring (see 4 bara example in Figure 34).

Sensible heat: Heat which is added to or taken away from a fluid which results in a change of temperature of the fluid. Water, for example, which is cooled from 50 °C to 30 °C has had sensible heat removed.

Smokebox: A chamber in which the burnt combustion gases are reversed in their direction of flow, normally not part of the pressure vessel.

Specific: A property of a substance which is defined per kilogramme of that substance, e.g. *specific heat capacity*.

Specific heat capacity: The amount of heat required to raise the temperature of 1 kg of a substance by 1 Kelvin [kJ/(kg K) or kJ/(kg °C)].

Specific energy consumption: The energy consumption of a process per unit of product or the units of steam produced per unit of fuel burnt (this is similar to miles per gallon [mpg] or liter per 100 km [l/100 km]).

Steam hammer: An overpressure event which is caused by the sudden opening or closing of a valve, for example. It can result in damage to the steam system. This phenomenon is explained in more detail on Page 63.

Steam tables: The thermophysical properties of water and steam have been formulated by the International Association for the Properties of Water and Steam IAPWS (http://www.iapws.org). These formulations are frequently reproduced in the form of tables known collectively as steam tables although they include all the various phases of water. Simplified versions referenced to both bara and barg are produced by many organizations and companies. Those interested in further reading on this subject are directed to the IAPWS website.

Steam trap: A device for separating condensate from steam and removing it from the distribution system. There are a number of different types commercially available.

Sub-cooling: Usually applied to condensate that has lost some heat resulting in a fluid temperature less than the saturation temperature. Sub-cooled condensate is frequently associated with several forms of *water hammer*.

Superheated steam: Steam vapor which is above the saturation line but below the critical point. For example, if heat is added to saturated steam under isobaric conditions its temperature will increase and the steam becomes superheated. Superheated steam is a vapor, not a gas. It is very commonly generated in boilers for power generation or less commonly process applications. Superheated steam may also be found in unexpected situations, e.g. in a notionally saturated steam distribution system due to a sudden reduction in pressure (see 4 bara example in Figure 34).

Supercritical steam: Steam, the pressure and temperature of which is above the *critical point*. Supercritical steam is a gas, not a vapor.

Swell: see *priming*.

Three-pass boiler: A boiler arranged with a furnace and two convective passes (Figure 9). The products of combustion pass through the furnace and their direction is reversed prior to entry to the first convective pass in a reversal cell. This is repeated prior to entry to the second convective pass (in a smokebox); hence the description “three pass”.

Total dissolved solids (TDS): As a steam boiler exports steam it concentrates the dissolved solids contained in the feedwater. Some of the concentrated solids remain dissolved. This concentration is often expressed in parts per million. Excessive TDS may cause foaming at the free surface of the water inside the boiler. This is undesirable as it promotes carryover and may negatively affect the operation of the level controls.

Total dissolved solids blowdown: The TDS naturally concentrates near the free surface of the water inside the boiler. It is controlled by opening a valve located just under the free surface. The pressure inside the boiler forces water along with the dissolved solids out of the boiler.

Vapor: Saturated and superheated steam are strictly a vapor, not a gas. A vapor is capable of being compressed back into the liquid phase under isothermal conditions.

Triple point: The temperature and pressure at which the solid, liquid and vapor phases of a substance are in thermodynamic equilibrium. For water the triple point is 273.16 K (0.01 °C) and 0.006112 bara.

Water hammer: There are several phenomena which are collectively termed "water hammer": plug flow water hammer, condensate-induced water hammer and column closure water hammer. These are explained in detail on Page 56 ff.

Water-tube boiler: A boiler which consists of two pressure vessels connected by tubes with one pressure vessel below the other. The lower vessel contains liquid water and is often known as a "mud drum". The upper vessel contains steam and is known as the "steam drum". The furnace is external to this arrangement. As the water passes up through the tubes it boils to form steam. This type of boiler is capable of producing very high pressure steam with a large amount of superheat.

Water wedged: A boiler that has been taken ouwt of service and completely filled with water to prevent corrosion of the internal surfaces of the pressure vessel.

Wetback furnace: A furnace in which the reversal cell is completely submerged under the surface of the water inside the boiler except for an access hatch. This type of furnace increases the available surface area for heat transfer.

Wet steam: A *liquid-vapor mixture* of steam, usually but not always with a high *dryness fraction* when found in a distribution system during normal operation. Condensate return systems however have a much lower dryness fraction as their purpose is to return condensate to the boiler house.

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