# Design and operation of modern steam systems




TM58: 2015

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# Design and operation of modern steam systems

## CIBSE TM58: 2015



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## Note from the publisher

This publication is primarily intended to provide guidance to those responsible for the design, installation, commissioning, operation and maintenance of building services. It is not intended to be exhaustive or definitive and it will be necessary for users of the guidance given to exercise their own professional judgement when deciding whether to abide by or depart from it.

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## Foreword

The image many people have of steam is that of steam trains or 19th century factory chimneys, leading them to conclude that steam is from a bygone era. But there is probably more steam generated globally today as there ever was, the difference is that the applications are changing.

Most steam generated today is used to produce electrical power, but this application is outside the scope of this manual. This text focuses on the use of steam at lower pressures and temperatures for a variety of uses in a building services environment.

Steam systems are no longer installed just to heat buildings. Instead, they play key roles in producing hot water, humidifying air, sterilisation and even cooling, all of which are relevant to building services engineers today.

To many, steam is a frightening and difficult fluid, so it can be overlooked in favour of what seem to be simpler alternatives. This is partly due to educational establishments taking the view that steam has had its day and need no longer be included in taught courses. This, of course, is not true. With some more knowledge and guidance, the benefits and flexibility that steam provides can be realised.

The aim of this technical manual is to provide some basic guidelines on how a modern steam system should be designed and commissioned. The operation of a steam and condensate system is also covered with respect to:

— energy efficiency

— training

— maintenance.

Regulatory changes are of course focusing attention on steam system safety. This manual introduces the reader to current regulations and guidelines, but these are of course fluid and subject to change so reference to current standards and regulations should always be made.

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# Design and operation of modern steam systems

## 1 Steam's use in building services today

### 1.1 Steam's unique properties that make it suitable for use in building services

It is true to say that steam is used less in building services today than it was 50 years ago, when steam heating systems were common. However, steam hasn't disappeared completely. Its unique properties still make it a valuable fluid in areas such as:

— distributing heat

— providing effective and safe humidification

— achieving efficient sterilisation

— using waste heat to provide chilled water.

The unique properties that make steam suitable for such applications are listed below:

— *Sterility:* steam itself is a sterile medium and is used extensively as part of the process of product sterilisation in healthcare and the food, beverage and pharmaceutical industries. This sterility is an important benefit of using steam to provide humidification in an air conditioning system.

— *High heat content:* a high heat content means that smaller pipes are required to transmit a given amount of heat compared with a water-based system.

— *High heat transfer value:* when steam gives up its heat, the resultant change of state from a gas to a liquid results in a very high heat transfer rate. With steam, the heat is transferred at a constant primary temperature, which means that there are no hot or cold spots in the heat exchanger. The high heat transfer rate means that heat exchangers can be smaller in size, saving valuable plantroom floor area.

— *Ease of distribution:* no pumps are required to move the steam around the pipework system. This avoids the need for complex system balancing. Control of flow (which in turn controls pressure, temperature or humidity, etc.) is achieved using only two port valves.

— *Flexibility:* a single steam supply to a plantroom can provide the means to heat, cool (absorption chillers) and humidify. Multi-fuel use can be accommodated easily. The flexibility of steam distribution systems means that the system can be modified without creating balancing problems. Steam storage enables energy to be stored to cope with peak loads, making system and equipment sizing a less critical issue.

#### 1.1.1 Distributing heat

Steam at the pressures used in building services applications has a condensable energy content of between 2000 and 2200 kJ/kg.

For every kg, approximately 2000 kJ of energy is released when the steam gives up its heat by condensing. The release of this energy, resulting from the change of state from gas to liquid, means that heat transfer coefficients (U values) are high, so heat exchangers with smaller heat transfer surface areas can be employed.

To distribute the same quantity of heat, a steam flow pipe may be similar in size to a medium temperature hot water (MTHW) or high temperature hot water (HTHW) pipe, but the pipe returning the condensed steam (condensate) will be smaller, typically by two pipe sizes.

Since the steam flow pipe contains a gas rather than a liquid, it will be lighter in weight, so reducing pipe support costs. Provision for expansion and the use of anchors and guides should, of course, be considered.

#### 1.1.2 Providing effective and safe humidification

Humidification means adding moisture to air. This can be done using water, but the water has to be introduced into the air in a form which is easily absorbed. This requires the water to be evaporated or atomised. Steam, in contrast, is already in an easily absorbed form (see Table 1.1). With appropriate design, all the steam injected will be absorbed, eliminating the risk of stagnant pools of water where bacteria can survive and grow. The steam is absorbed with no significant rise in air temperature. Although water

**Table 1.1** Comparison between steam and water for humidification

| Factor | Steam | Water |
|---|---|---|
| Cleanliness | Clean and sterile. Can be used in clean areas | Disinfection needed. Care must be taken to avoid formation of stagnant pools. Not suitable for clean areas |
| Efficiency | Easily absorbed | Carryover normal. Eliminator plates necessary |
| Heating | No change in air temperature | Reheat battery needed |
| Cooling | No cooling effect | Can provide evaporative cooling |
| Controllability | Easily controlled | Difficult to control |

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humidification using evaporative methods does have the advantage of cooling the air in summer, in winter it will add to the heating load.

The cleanliness and sterility associated with steam humidification makes it the preferred choice in most applications, particularly healthcare, pharmaceuticals and the food and beverage and electronics industries.

### 1.1.3 Achieving efficient sterilisation

Steam kills micro-organisms by coagulating and denaturing their cell protein. The presence of temperature, moisture and a rapid exchange of heat caused by the steam condensing all contribute to the sterilising process.

The temperatures, pressures and times required to achieve effective sterilisation depend on the process but typical values are:

— 121 °C (steam at 1.1 bar g) for 15 minutes (fluids in sealed containers)

— 134 °C (steam at 2.1 bar g) for 5 minutes (porous loads).

(*Note*: bar g represents gauge pressure, i.e. the absolute system pressure minus atmospheric pressure.)

### 1.1.4 Using waste heat to provide chilled water

The production of chilled water using heat is achieved through the use of absorption chillers (see CIBSE publication KS13: *Refrigeration* for more details of their operation). They use heat to re-concentrate the working fluid. This heat can come from any source, but waste heat in the form of steam could be utilised. There is generally more demand for flash steam during the winter, but harnessing it for use in absorbtion chillers would be an effective use during the summer months. Absorption chillers use steam at low pressures (1 bar g would be typical), such as would be available from a flash steam recovery system.

## 1.2 Overview of a typical steam and condensate system

### 1.2.1 The steam and condensate loop

The steam and condensate system is sometimes referred to as the steam and condensate loop due to the circular nature of the process. It extends from the site of steam generation, through the distribution system to the point of use after which the condensed water is collected and returned to the generator for reuse (see Figure 1.1).

Traditionally, most steam generators in building services applications have been shell and tube boilers. In more modern systems, shell boilers continue to dominate, but the use of smaller scale steam generators is growing. See Section 5.2 for more details. These smaller generators may be used simply to produce smaller quantities of steam or steam of a different quality, such as the 'clean steam' used for some healthcare and pharmaceutical applications. In combined heat and power (CHP) systems, a heat recovery steam generator (HRSG) is commonly used, in which hot exhaust gas, rather than a fossil fuel, is used to provide the heat source.

Steam is generated and distributed at high pressure and generally used at low pressure. Steam tables (see Table 1.2) show that higher pressure steam has a lower specific volume, meaning that smaller pipes can be used to distribute the same mass flow rate (similar to the high-tension system in electrical power distribution).

The steam is distributed by steel pipework, which must be correctly sized and regularly drained to remove any condensate.

To recap, the areas where steam is used in building services applications are in producing LTHW for heating, domestic hot water for ablutionary purposes, increasing humidity in air conditioning applications and providing sterilisation.

Within the steam-using equipment, when the steam condenses it gives up its enthalpy of evaporation,

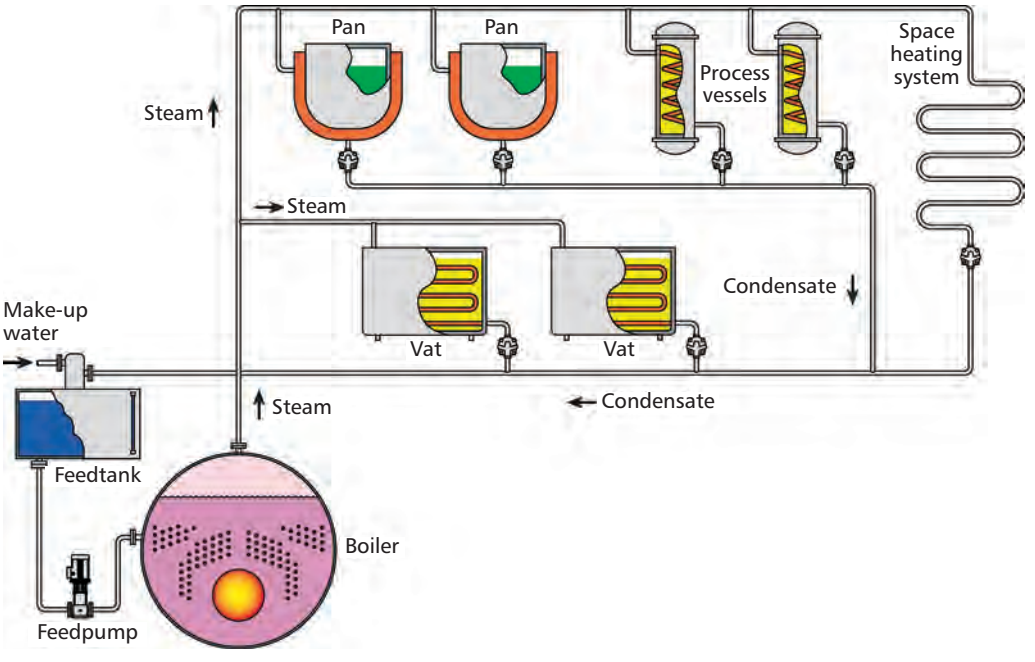



**Figure 1.1** The steam and condensate loop

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transferring heat to the secondary medium. Steam at low pressure has a higher enthalpy of evaporation, so less mass is required for a given amount of heat transfer.

A steam trap is needed to 'trap' the steam but release the condensate formed.

This condensate is a valuable source of both hot and treated water so is returned to the boiler feedtank for recycling, thereby saving water, energy and treatment costs. This condensate can flow back under gravity, but in most cases would need a mechanical or electrical pump.

The condensate pipework is more susceptible to corrosion than the steam pipework, so materials such as galvanised steel, copper or stainless steel are used.

The four steps outlined above – generation, distribution, usage and condensate return – comprise the steam and condensate loop (Figure 1.2).

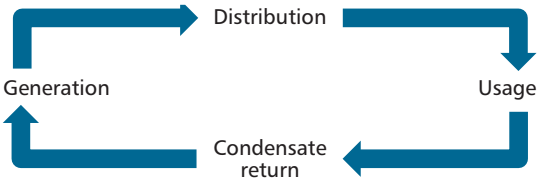



**Figure 1.2** Key areas of steam and condensate loop

### 1.2.2 Properties of steam

*Steam tables*

When a mass of water has been converted to 100% steam, it is referred to as dry saturated steam. Dry saturated steam is unique in having a fixed relationship between pressure and temperature. If the pressure is known, then so too is the saturation temperature, and vice versa.

The properties of dry saturated steam, and the condensate formed when steam transfers its heat and condenses back to water, are shown in steam tables (see Table 1.2 for an extract).

For each pressure, the table shows the following properties:

— Temperature °C: the temperature at which water boils at this pressure, the temperature of saturated steam and the temperature of the condensate formed when the steam condenses. Steam does not cool as it changes state to condensate, so a consistent temperature is maintained over the area of the heat transfer surface. Note also that because steam pressure and temperature are directly related, steam temperature can be accurately achieved and maintained by regulating the pressure to that corresponding to the temperature required.

— Enthalpy of water kJ/kg: the heat content of the boiling water above a datum of water at 0 °C. This is also the heat content of condensate formed when steam condenses.

— Enthalpy of evaporation kJ/kg: the heat required to convert boiling water at this pressure into steam at the same pressure. It is also the heat transferred to a heating surface when steam condenses. Note there is no change in temperature during evaporation or condensing. Evaporating 1 kg of water will produce 1 kg of steam and condensing this steam will produce 1 kg of condensate, all at the same temperature.

— Enthalpy of steam kJ/kg: the total heat energy content of the steam above water at 0 °C.

$$\text{enthalpy of steam} = \text{enthalpy of water} + \text{enthalpy of evaporation}$$

— Volume m³/kg: volume of the steam. Note how the volume changes with pressure, which will affect pipe sizes.

**Table 1.2** Extract from steam tables

| Pressure | Saturation temperature | Specific enthalpy of water, $h_f$ | Specific enthalpy of evaporation, $h_{fg}$ | Specific enthalpy of steam, $h_g$ | Specific volume of steam, $v_g$ |
|---|---|---|---|---|---|
| (bar gauge) | (°C) | (kJ/kg) | (kJ/kg) | (kJ/kg) | (m³/kg) |
| 0 | 100 | 419 | 2257 | 2676 | 1.67 |
| 1 | 120 | 506 | 2201 | 2707 | 0.88 |
| 2 | 134 | 562 | 2163 | 2726 | 0.60 |
| 3 | 144 | 605 | 2133 | 2739 | 0.46 |
| 4 | 152 | 641 | 2108 | 2749 | 0.37 |
| 5 | 159 | 671 | 2086 | 2757 | 0.31 |
| 6 | 165 | 698 | 2066 | 2763 | 0.27 |
| 7 | 171 | 722 | 2047 | 2769 | 0.24 |
| 8 | 175 | 743 | 2030 | 2774 | 0.21 |
| 9 | 180 | 763 | 2015 | 2778 | 0.19 |
| 10 | 184 | 782 | 2000 | 2781 | 0.18 |
| 11 | 188 | 799 | 1985 | 2784 | 0.16 |
| 12 | 192 | 815 | 1972 | 2787 | 0.15 |
| 13 | 195 | 831 | 1959 | 2789 | 0.14 |
| 14 | 198 | 845 | 1946 | 2792 | 0.13 |
| 15 | 201 | 859 | 1934 | 2793 | 0.12 |
| 16 | 204 | 872 | 1923 | 2795 | 0.12 |
| 17 | 207 | 885 | 1912 | 2796 | 0.11 |

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Steam does not require a circulating pump to cause it to flow in a pipe. The drop in pressure caused by the condensation of steam in a heat exchange process will cause it to flow. A steam system therefore does not require to be balanced in the way that a liquid system does.

*Temperature–enthalpy diagram*

A graph showing temperature vs. enthalpy is a useful way of visualising steam's properties (see Figure 1.3).

Note how the enthalpy of evaporation is higher at lower pressures.

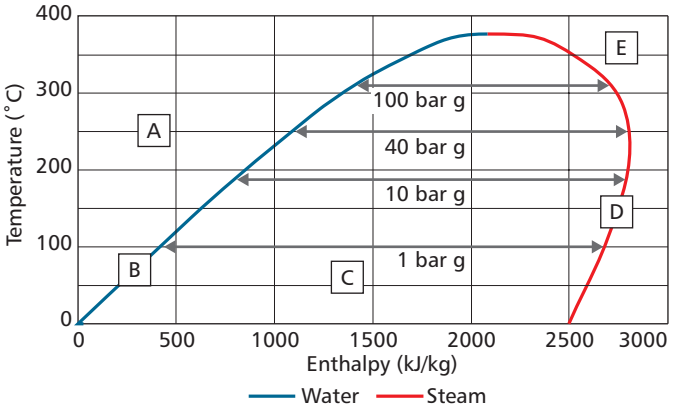



**Figure 1.3** Temperature vs enthalpy

### 1.2.3 Steam consumption

When steam condenses it is the enthalpy of evaporation (kJ/kg) that is transferred from the steam to the secondary medium across the heat transfer surface. This value is used to convert heat energy (kJ/s) to the flow rate of steam required to provide this energy.

**Example**

*An air heater is rated at 200 kW (200 kJ/s) when supplied with steam at 4 bar g.*

*How much steam does the heater require to satisfy this demand?*

*What will be the condensate flow rate leaving the heater?*

$$\text{Steam flow required} = \frac{\text{(energy required kJ/s)}}{\text{(enthalpy of evaporation } (h_{fg}) \text{ kJ/kg)}}$$

Enthalpy of evaporation at 4 bar g: 2108 kJ/kg

Therefore steam flow rate required:

200/2108 = 0.095 kg/s or 341 kg/h.

Condensate flow rate will be the same as the steam flow rate, 341 kg/h.

### 1.2.4 Flash steam generation

Reference to steam tables shows that the heat content (specific enthalpy of water) of condensate changes with pressure. When condensate at high pressure passes through a steam trap to an area at atmospheric pressure its heat content reduces to 419 kJ/kg. The difference in heat content is not lost but evaporates some of the low-pressure condensate (known as flashing) and produces flash steam, which ideally should be recovered and used elsewhere. The heat content, and hence usefulness, of flash steam is equal to that of live (boiler) steam.

**Example**

*1000 kg/h of condensate at 10 bar g passes through a steam trap to atmospheric pressure. How much flash steam will be produced?*

Enthalpy of water at 10 bar g: 782 kJ/kg

Enthalpy of water at 0 bar g: 419 kJ/kg

Heat available to produce flash steam at 0 bar g:

782 – 419 = 363 kJ/kg.

To evaporate 1 kg of condensate at 0 bar g into steam required 2257 kJ (enthalpy of evaporation from steam tables).

Therefore, 363 kJ will produce 363/2257 = 0.16 kg of flash steam per kg of condensate.

From 1000 kg/h of condensate, 160 kg/h of flash steam will be produced. The pipe after the steam trap will therefore be carrying 840 kg/h of condensate and 160 kg/h of flash steam. This two-phase flow needs to be taken into consideration when sizing condensate return pipes (see Section 5.10.2).

### 1.2.5 Classification of steam and steam quality

There are five distinct parts of the temperature–enthalpy graph shown in Figure 1.3, which relate to the properties of steam:

— Region A: in this region, the water has not yet reached saturation temperature (boiling point). It is referred to as sub-saturated water.

— Line B: line B is the saturated water line. The water has now reached boiling point and any addition of heat will cause some to convert from liquid (water) to a gas (steam).

— Region C: in this region there is a mixture of steam and water. Moving horizontally across the graph, the addition of enthalpy causes more of the water to convert to steam. Region C is the wet steam area.

— Line D: when line D is reached, all the water has been converted to steam. Line D is referred to as the dry saturated steam line.

— Region E: adding extra heat to dry saturated steam causes it to be superheated. In this region, the direct relationship between temperature and pressure is lost. Steam here is referred to as having $y$ degrees of superheat, where $y$ is the difference between steam temperature and saturation temperature at that pressure.

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*Wet steam*

Wet steam, as mentioned above, is a mixture of steam and water. It is difficult to produce dry saturated steam as, even in a boiler, the steam space above the water will contain a mixture of steam and water particles due to turbulence and splashing. The mixture will leave the boiler as wet steam.

Whenever steam loses heat, for example by way of radiation losses from pipes, that heat loss will be accompanied by an increase in water content – the steam will become wetter.

Wet steam can also result from steam picking up water from any condensate not removed from the distribution pipework.

If the water content of the mixture is 5% by mass, the steam is said to be 95% dry and has a dryness fraction of 0.95.

Due to the water content, the heat energy within wet steam is less than that within dry saturated steam. Importantly, in wet steam the enthalpy of evaporation – the normal usable energy content of steam – is reduced so the heat output of plant can be affected.

**Example**

*Steam at 7 bar g has a dryness fraction of 0.96. In the condensation process, how much heat energy is transferred compared with the case of dry saturated steam?*

Enthalpy of dry steam at 7 bar g: 2047 kJ/kg

Enthalpy of steam that is 96% dry:

$$2047 \times 0.96 = 1965 \text{ kJ/kg}.$$

Water particles in wet steam can also cause erosion of pipes and fittings. To dry the steam it is usual to fit a steam separator after the boiler crown valve. Separators may also be fitted before steam flowmeters and pressure reducing valves to protect them from wet steam damage.

Steam pipes must also be regularly drained to prevent the build up of condensate, which not only causes wet steam but also the more dangerous phenomenon of waterhammer (see Section 4.1.3).

## 1.3 Superheated steam

Superheated steam is rarely used for building services applications. For heating applications, superheated steam would have to lose its degree of superheat by cooling to reach saturation temperature, at which point it condenses and gives up its enthalpy of evaporation. The time required for cooling to take place will reduce the heat output of plant.

Where steam is used for sterilisation, superheat is also to be avoided. Moist heat is a prime requirement for effective sterilisation.

Superheated steam has particular properties which make it suitable for steam turbine applications. Consequently, one possible application in building services is within a CHP system, where steam is used to provide power (as opposed to being a means of transferring waste heat).

# 2 Calculating the cost of steam

## 2.1 Why calculate the cost of steam?

It is generally accepted that steam, in common with most utilities, is expensive and likely only to rise in price in future. Whereas most people would be able to quote the price they pay for other methods of producing heat (cost per kWh of electricity, therm of gas or litre of oil), the cost of 1 kg of steam is less well known, but it should be calculated.

Establishing the cost of steam has several advantages:

— the viability of any energy- and water-saving proposals can be readily established through a comparison with savings in steam costs

— with accurate metering installed, steam costs can be allocated to discrete buildings/departments, thereby introducing an incentive to save

— a benchmark comparison can be made with similar steam-producing facilities to determine generation efficiency.

The cost of steam is conventionally expressed in currency units per tonne (1000 kg). Costs can vary considerably, the main factors affecting them being the cost and type of fuel used and the parameters included in the cost of steam calculation.

## 2.2 Which costs to include and calculation methodology

The main components in the cost of steam are the raw materials involved in its production. These will be – in the order in which they are generally considered:

— fuel

— water

— water treatment consumables/chemicals.

Other factors that could be included are:

— the labour costs of staffing a boilerhouse

— maintenance costs

— costs of any disposal – water (condensate) disposal or ash removal from solid fuel boilers.

## 2.3 Calculating the cost of steam

### 2.3.1 Fuel

The principal cost in producing a kilogram or tonne of steam is the fuel used.

The quantity of fuel used will depend on its calorific value, the efficiency of the boiler, the incoming water temperature and the steam pressure required.

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**Example**

*Calculate the fuel cost of producing steam at 10 bar g from water at 80°C with a boiler efficiency of 90%. Fuel is natural gas at a cost of 0.03 currency units per kWh.*

The total energy in the steam is the energy required to heat the incoming water to saturation point plus the additional energy (enthalpy of evaporation) needed to convert this saturated water to dry saturated steam.

From steam tables, enthalpy of saturated water at 10 bar g: 782 kJ/kg

Enthalpy of incoming water:

4.19 (specific heat of water) × (80 – 0) = 335.2 kJ/kg

Additional heat energy to be added:

782 – 335.2 = 446.8 kJ/kg

Enthalpy of evaporation at 10 bar g: 2000 kJ/kg

Total energy to be added:

446.8 + 2000 = 2446.8 kJ/kg

With a boiler efficiency of 90%, energy to be added in fuel:

2446.8/0.9 = 2718.7 kJ/kg.

For 1000 kg (1 tonne), energy input required = 2718.7 MJ or 755 kWh.

At a cost of 0.03 currency units per kWh, total fuel cost is:

755 × 0.03 = 22.65 currency units per tonne.

### 2.3.2 Water

The cost of the initial filling of a boiler is ignored. In an ideal steam and condensate system, all of the steam supplied is condensed and all of this condensed water is returned to the boiler to be reused. However, in reality no steam system is ideal. In many, steam/water is lost through:

— steam leaks
— steam lost through leaking steam traps
— steam escaping from vent pipes
— water leaks from condensate system
— steam/water lost during boiler blowdown (the process of diluting boiler water contents to remove some of the concentrated solids)
— any steam used directly as part of a process (e.g. humidification).

For all of these reasons, a 100% condensate return rate is unachievable. Suitably treated make-up water will be required to make good for any shortfall. A water meter on the incoming supply to the boilerhouse is the simplest way to measure water consumption. An alternative is to calculate the percentage condensate return and use the difference to determine the make-up water required.

However, measuring the condensate return rate can be difficult if the condensate pipe contains a mixture of water and flash steam. An effective estimate of percentage condensate return can be determined by measuring the conductivity of the make-up water in terms of the total dissolved solids (TDS) fraction, the condensate return and the feedwater entering the boiler. This simple ratio calculation will give a reasonable estimate of percentage condensate return.

$$\%\ \text{rate of condensate return} = \frac{\text{make-up water TDS} - \text{feedwater TDS}}{\text{make-up water TDS} - \text{condensate TDS}} \times 100$$

**Example**

Readings with a conductivity meter indicate conductivity values as follows:

Make-up water: 300 ppm

Condensate return: 60 ppm

Boiler feedwater: 125 ppm

$$\%\ \text{rate of condensate return} = \frac{300 - 125}{300 - 60} \times 100 = 73\%$$

There may also be a component of water cost from any effluent disposal charges based on condensate being returned either to an effluent treatment plant or discharged into the public sewer system. It will be difficult to measure this, but the greater the amount of condensate that is returned and reused, the lower any effluent costs will be.

### 2.3.3 Water treatment consumables/ chemicals

Water treatment is a specialist subject meriting a document of its own. BS EN 12953-10 provides guidance.

It is only make-up water that requires treatment. Returned condensate has been formed as part of a distillation process so, in theory, should be pure and require no additional treatment. However, some processes may have the potential to contaminate condensate so it cannot always be assumed that no treatment of the condensate is required. An example of possible contamination would be a leaking heat exchanger where the product being heated could be drawn into the steam system. This could occur at shut-down when the steam condenses and a vacuum forms on the side that is normally at the higher pressure.

The composition of make-up water will vary. Some of the more commonly found contaminants, together with their effects and prevention, are listed in Table 2.1.

Treating the make-up water can be done by chemical or mechanical means, and sometimes by a combination of both. Each method will have its own cost, which should be translated into a cost per $m^3$ of water treated. This can then be added to the water purchase costs as it is only the incoming make-up water that will generally require treatment.

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**Example of total costs**

*Calculate the total cost involved in producing steam at 10 barg from water at 80°C with a boiler efficiency of 90%. Fuel is natural gas at a cost of 0.03 currency units per kWh. Water costs are 1 currency unit per m³, with treatment costs being 0.2 units per m³. Condensate return rate is estimated at 80%. Ignore water effluent charges.*

From the calculation in Section 2.3.1, fuel cost is 22.65 currency units per tonne.

With the condensate return rate of 80%, 20% make-up water is required. So, for 1 tonne steam, 200 kg (0.2 m³) of make-up water is needed.

Water cost is:

$0.2 \times 1 = 0.2$ units per tonne steam

Treatment cost is:

$0.2 \times 0.2 = 0.04$ units per tonne steam

Total steam cost:

22.65 (fuel) + 0.2 (water) + 0.04 (treatment) = 22.9 currency units per tonne.

**Table 2.1** Boiler water contaminants

| Contaminant | Effect | Prevention |
|---|---|---|
| Calcium carbonate<br>Calcium bicarbonate<br>Magnesium carbonate<br>Magnesium bicarbonate | Affect the 'hardness' of the water. These salts will form scale, which can reduce output in heat exchangers but, more seriously, cause overheating in steam boilers | Softening to convert scale-forming salts to soluble salts. Reduction in the dissolved solids level by demineralisation, reverse osmosis or de-alkalisation |
| Sodium chloride<br>Sodium carbonate | These are non-scale-forming salts, but they do add to the TDS count in the boiler. If the TDS level is not controlled, foaming occurs, leading to contamination of the steam system with boiler water and a wet steam supply | Automatic control of boiler TDS levels along with a properly managed boiler bottom blowdown regime will maintain TDS values at the right levels |
| Oxygen<br>Carbon dioxide<br>Other gaseous compounds | These contaminants are likely to cause corrosion in pipework and heat exchangers. They will also hinder heat transfer due to their insulating effect on heat transfer surfaces | Effective air removal through mechanical or chemical means. Mechanical removal involves raising the temperature of the stored feedwater to drive off oxygen and other gases. Install air vents in distribution system |

## 2.4 Comparison of steam costs vs cost of other heat transfer media

Section 1 listed the properties of steam and why steam is an effective means of transferring heat. But there are other fluids that can also be used for heat transfer purposes, the main ones being water and thermal oil. It is difficult to make an accurate assessment of the cost of using steam in comparison with these other options. However, there are some considerations that can assist the decision-making process.

Transferring heat is all about moving energy from source to point of use. In an inefficient system, some energy will be lost along the way. This loss would be the same whether steam, high temperature water or thermal oil was used. In all of these methods of distribuition, the aim must be to minimise heat loss with correct pipe sizing and effective insulation. But, with steam, there could be additonal losses to consider. These are the mass losses associated with generating $x$ kg of steam and returning $y$ kg of condensate, $x-y$ representing a mass (water) loss. This means that additional water has to be purchased, treated and heated.

It is difficult to completely seal a steam system as pressure drop is necessary to cause steam to flow. Dropping pressure can result in the formation of flash steam from condensate and this will be lost from vents unless it is captured for reuse. Section 8.4 covers flash steam formation and recovery in more detail.

All mass losses from the steam system can be minimised – leaks cured, more condensate returned, flash steam reused, boiler blowdown reduced – in order to make a comparison with other transfer methods more favourable.

Remember that no circulating pumps are required with steam, so electrical running costs should be lower.

Water is cheaper to buy and treat than thermal oil, so steam systems offer cost savings over thermal oil systems. Heat loss from the larger diameter return pipework associated with HTHW systems can also be greater than that from condensate pipework (which is at a lower temperature and has a smaller diameter).

Table 2.2 highlights the specific issues involved in the use of steam and makes a comparison with the other options mentioned. Some of the points listed will have cost implications, others will involve operational matters.

As can be seen, there are both favourable and unfavourable comparisons to be made between steam and the other options available to transfer heat. It is difficult to make a quantitative assessment of which is the most cost-effective option. The choice may be down to user preference, but a key factor could be whether there is an additional use for the steam other than transferring heat, such as humidification, sterilisation or product processing.

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**Table 2.2** Steam vs hot water vs high temperature oil comparison

| Steam | Hot water | High temperature oils |
|---|---|---|
| High heat content | Moderate heat content | Poor heat content |
| Latent heat approximately 2100 kJ/kg | Specific heat 4.19 kJ/kg·°C | Specific heat often 1.69–2.93 kJ/kg·°C |
| Inexpensive | Inexpensive | Expensive |
| Some water treatment costs | Only occasional dosing | |
| Good heat tranfer coefficients | Moderate coefficients | Relatively poor coefficients |
| High pressure required for high temperatures | High pressure needed for high temperatures | Low pressures only to get high temperatures |
| No circulating pumps required | Circulating pumps required | Circulating pumps required |
| Small pipes | Large pipes | Even larger pipes |
| Easy to control with two-way valves | More complex to control – three-way valves or differential pressure valves may be required | More complex to control – three-way valves or differential pressure valves may be required |
| Temperature breakdown is easy through a reducing valve | Temperature breakdown more difficult | Temperature breakdown more difficult |
| Steam traps required | No steam traps required | No steam traps required |
| Condensate to be handled | No condensate handling | No condensate handling |
| Flash steam available | No flash steam | No flash steam |
| Boiler blowdown necessary | No blowdown necessary | No blowdown necessary |
| Water treatment required to prevent corrosion | Less corrosion | Negligible corrosion |
| Reasonable pipework required | Searching medium (HTHW), welded or flanged joints usual | Very searching medium, welded or flanged joints usual |
| No fire risk | No fire risk | Fire risk |
| System very flexible | System less flexible | System inflexible |

# 3 Steam quality requirements

The quality of steam is determined by measuring the contaminants it contains. Steam from a conventional boiler might contain small quantities of boiler feedwater chemicals, traces of pipe scale and other corrosion deposits. It will contain water in suspension and, very possibly, air and other incondensable gases.

The process that requires the steam will dictate how free of contaminants it should be. For some processes, the steam must be free of water droplets. Typically, this would be for power generation, where superheated steam is common. Where steam is directly injected in food production, physical contaminants are obviously to be avoided.

For building services applications, the general requirements are less severe and, typically, the steam would be specified as being clean, dry, free of air and at the correct pressure.

## 3.1 Dryness

A conventional boiler (one without a superheater) can, at best, produce dry saturated steam. This would normally be referred to as being 100% dry, or having a dryness fraction of 1. In practice, because of turbulence and splashing in a boiler, the steam space above the water will contain a mixture of steam and water particles. The mixture leaving the boiler is known as wet steam. Wet steam can also result from steam picking up condensate particles in pipework. If the water content of the mixture is 5% by mass, the steam is said to be 95% dry and to have a dryness fraction of 0.95.

Due to the water content, the heat energy within wet steam is lower than that within dry saturated steam. Importantly, the enthalpy of evaporation, the normal usable energy content of steam, is lower, so heat output of plant can be affected.

**Example**

*Steam at 7 bar g has a dryness fraction of 0.96. In the condensation process, how much heat energy is transferred compared with that of dry saturated steam?*

Enthalpy of dry steam at 7 bar g: 2047 kJ/kg

Enthalpy of steam that is 96% dry:

$$2047 \times 0.96 = 1965 \text{ kJ/kg}.$$

Water particles in wet steam can also cause erosion of pipes and fittings. To dry the steam it is usual to fit a steam separator after the boiler crown valve. Separators may also be fitted before steam flowmeters and pressure reducing valves.

Steam pipes must also be drained regularly to prevent the build up of condensate as steam will pick up the water particles.

Wet steam formation can also be caused by boiler overloading, poor control of TDS levels or contamination of boiler feedwater. These factors should be addressed first but, as soon as there is heat loss from the steam system, additional water will be formed as the steam condenses.

To help dry the steam, it is recommended that a separator is fitted in the off-take from the boiler, as near as possible to the

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boiler. Separators work by forcing the steam to change direction rapidly. This results in the denser water particles being separated from the steam due to their inertia. They are then encouraged to gravitate to the bottom of the separator body, where they collect and drain away via a steam trap.

Steam can also be dried through the action of a pressure reducing valve. This drying effect can produce some superheated steam in the downstream pipework, something that must be avoided for applications such as healthcare sterilisers. The superheating effect is negligible and any superheated steam formed quickly disappears through heat loss from the pipework.

## 3.2 Non-condensable gases

These are gases that either come across with the steam from the boiler or enter the distribution system and boiler at shut-down. Air is the main contributor. It is a very effective insulator, so where steam is used to transfer heat, the presence of air will slow the heat transfer process.

The presence of air also disrupts the normal pressure–temperature relationship of dry saturated steam.

Air will contribute to pressure, but not to temperature. By measuring temperature and pressure, an estimation of the percentage of air in the steam/air mixture can be established. Dalton's law of partial pressures states that the total pressure in a mixture of gases is the sum of the partial pressure that each gas would exert if it occupied the total volume on its own. This is best illustrated by an example.

**Example**

*A steam/air mixture is at a pressure of 5 barg. The temperature is 150°C. What percentage of the mixture is air by volume?*

For Dalton's law, pressures must be expressed in bar *absolute* (bara = barg + 1)

150°C would normally represent a steam pressure of 3.8 barg (4.8 bara)

5 barg = 6 bara.

Percentage of air in mixture:

(6–4.8)/6 = 20%.

Air can also aggravate corrosion in the steam and condensate system. Carbon dioxide can be reabsorbed into any water formed, resulting in the formation of carbonic acid. This, coupled with the presence of oxygen, can result in corrosion, particularly in steam traps and condensate pipework. See Section 5.9.5 for guidance on air removal.

## 3.3 Cleanliness

Cleanliness can refer to the simple absence of particles, such as pipe scale, from the steam through to steam being so 'clean' that the condensate it produces could be safely injected into human beings with no adverse effects.

Cleanliness in a building services context means the installation of pipeline strainers that will trap particles before they can damage sensitive downstream devices, such as control valves, meters, steam traps, etc.

**Table 3.1** Clean steam applications

| Industry | Application | | | |
|---|---|---|---|---|
| | Humidification | Sterilisation | Sterilise in place (SIP) | Direct steam injection |
| Food and beverages | ✓ | ✓ | ✓ | ✓ |
| Healthcare | ✓ | ✓ | | |
| Pharamceuticals | ✓ | ✓ | ✓ | ✓ |
| Electronics | ✓ | | | |
| General building services | ✓ | | | |

Various designs are available but the most common type is the simple 'Y' pattern strainer, which can be fitted with an internal basket to offer differing degrees of filtration.

Where steam is used for culinary purposes, it is more appropriate to use a steam filter than a strainer. The principle is the same, but the degree of filtration is much finer. Typically, such filters would be fitted with 5 micron elements to remove 95% of particles larger than 2 microns.

Where even cleaner steam is required, again for culinary or medical purposes, a different type of steam generation system would be used. This would involve the use of a clean steam generator rather than a steam boiler. The feedwater would be specially softened potable water, deionised water or produced by reverse osmosis.

With both filtered and clean steam systems, it is important that contaminants are not introduced downstream. For this reason, pipework should be stainless steel tubing, or in some cases copper, where potable rather than the more ion-hungry forms of feedwater are used.

The most common applications for filtered and clean steam are for direct injection purposes rather than heat transfer. Table 3.1 shows some clean steam applications.

## 3.4 Steam quality standards

The standard for steam quality which is closest to being universally recognised is BS EN 285, a standard for steam quality for steam sterilisers. Guidance is also given in the UK Health Technical Memorandum (HTM) 2010: *Sterilization*.

# 4 Health and safety in steam and condensate systems

The design and operation of all steam and condensate systems must be carried out with safety in mind (see also Sections 7.3 and 9.4). It would be very unusual to find that there were no local regulations relating to their safe design and operation and these must always be adhered to. Some general issues that should be considered are outlined below.

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## 4.1 Safety issues particular to steam

The following issues will be addressed in detail in the subsections below:

— overheating (in boilers)

— overpressure

— waterhammer

— incorrect water treatment.

### 4.1.1 Overheating (in boilers)

There are two main causes of overheating in steam boilers. Both relate to the inability of the boiler to safely convert the heat energy available in the fuel into the energy required to convert water into steam. These causes are:

— excessive scale build-up on heat transfer surfaces

— low water levels in the boiler.

*Scale build-up*

When heated, some dissolved solids in feedwater are deposited as scale on the boiler furnace wall and on the water-side of fire tubes. These solids are typically sulphates and carbonates of magnesium and calcium. Their concentration will depend on the source and the percentage of make-up water used. A high percentage of condensate return can therefore minimise the effect of scale.

There are various methods available to convert these magnesium and calcium scales into something more manageable, such as the addition of chemicals, base exchange softening and demineralisation. Reverse osmosis, where dissolved solids are extracted from the water supply using a semi-permeable membrane, is gaining in popularity. Different methods have initial and running cost implications. Additional guidance on water treatment is available in BS 2486 *Recommendations for treatment of water for steam boilers and water heaters* and in BS EN 12953-10 *Shell boilers. Requirements for feedwater and boiler water quality*.

When boilers are stripped down for routine maintenance or for insurance inspection purposes an examination of all heat transfer surfaces on the water side should be carried out. This should be part of a written scheme of examination as required by the Pressure Systems Safety Regulations 2000.

*Low water levels*

If the heat transfer surfaces of the boiler furnace or tubes are exposed due to a lack of water, serious damage can result from overheating. Control of the water level in a steam boiler is therefore a crucial part of boiler operation.

The water level control will be either float switch or probe operated. Current practice is to install a level probe inside the boiler shell where true water level conditions can be measured. These probes will operate on conductance or capacitance principles, the former operating on an on–off basis and the latter continuous. On–off control would be appropriate for smaller boilers and those with a steady steam output.

The level control will operate a feedwater control valve or the feedwater pump. Control of the pump can be on–off or variable speed. Variable speed control provides more energy-efficient operation but any reduction in speed must still ensure that the pump is able to develop enough pressure to get water into the boiler.

Apart from normal level control, level alarms must also be fitted. These should be of high integrity, self-monitoring design to ensure operation in a fail-safe manner.

Level alarms should be of a two-stage design. Stage 1, referred to as 'first low', should switch off the fuel supply to the boiler and sound an alarm. If the water level is restored, the boiler would then go through its normal start-up sequence automatically. Stage 2, 'second low', is a more serious condition. This is triggered if the water level continues to fall below first low. In the second low scenario, the boiler is again shut down but in this case it is 'locked out' to prevent automatic start-up even if the water level rises. Restart has to be carried out manually, *after* the cause of the low water level has been established and rectified.

BS EN 12953-9 relates specifically to the requirements for boiler control systems of which level controls and alarms are a major part.

### 4.1.2 Overpressure

Steam boilers, pipework, valves, heat exchangers, etc. should be selected on the basis of the normal working pressure (NWP) of the steam system but also taking into account any allowable overpressure (see Section 5.7.4). Manufacturers' pressure limits must always be adhered to.

Steam will normally be generated and distributed at high pressure and used at low pressure because:

— a high generation pressure means a smaller boiler is required

— high distribution pressure means smaller pipes are required

— lower pressure at point of use means more enthalpy of evaporation ($h_{fg}$), and therefore less steam consumption (kg/h).

Pressure control valves should be used to reduce the steam pressure, ideally positioned close to the point of use. To protect boilers, pipework, fittings and steam equipment, safety valves should be fitted where there is any possibility of an overpressure situation arising. Advice on the sizing and selection of a safety valve should be sought from the manufacturer. Some key considerations are given below:

— The set pressure should be as far away from the normal operating pressure *as is safely possible* to prevent the safety valve from opening prematurely and wasting steam.

— The safety valve should be sized on the basis of a fault load – how much steam would it have to pass should upstream control equipment fail in the fully open position?

Safety valves, once set and installed, should not be tampered with for any reason. They must be able to discharge the required capacity of steam in a safe manner. Due consideration must therefore be given to the route and

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sizing of the discharge pipework. Again, manufacturers' advice on installation should be sought. Some general advice is given in Section 5.7.4.

### 4.1.3 Waterhammer

Waterhammer is caused when a slug of water travelling at high velocity hits a stationary object. The result is noise and, in extreme cases, a rupture of the pipework, valve, etc. Waterhammer in steam systems is caused by two different reasons. The cause is similar but the effect of one is potentially more severe than the other.

*Steam-flow waterhammer*

Water will gather at any low point in the steam distribution system if not properly drained. This water will be propelled forward with the steam flow (see Figure 4.1). Should the water fill the cross-sectional area of the pipe, then the velocity of the water 'slug' will be equivalent to that of the steam, which typically could be up to 40 m/s. If this slug of water, travelling at such velocity, hits a downstream obstruction, the kinetic energy it holds is dissipated as noise/vibration or component damage.

*Condensation-induced waterhammer*

This is the more dangerous type of waterhammer as the velocities involved are substantially greater that those seen in steam-flow waterhammer. The cause is the same – water accumulation in the steam pipework – but here, the water is not influenced by the steam velocity but by the creation of a vacuum caused by steam rapidly condensing (see Figure 4.2).

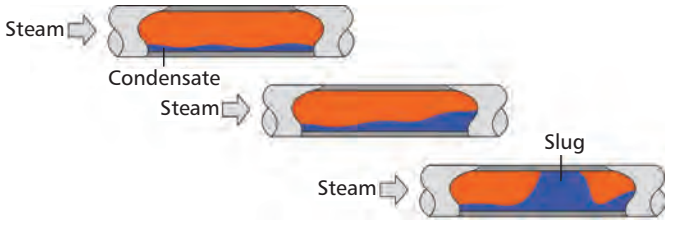



**Figure 4.1** Steam-flow waterhammer

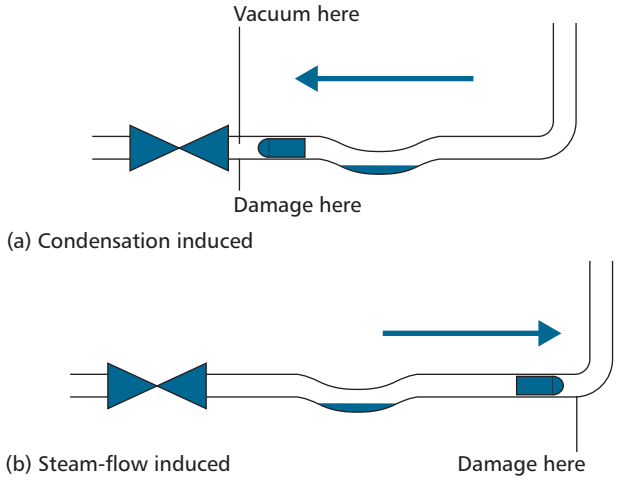



**Figure 4.2** Difference between condensation-induced and steam-flow waterhammer

This rapid condensing could be caused by steam entering a cold pipe or by some external source of cooling, so that the steam condenses almost instantaneously. The vacuum created induces the slug of water to flow towards it at velocities considerably in excess of 40 m/s.

Condensation-induced waterhammer is preventable by ensuring that any water formed is removed effectively and by preventing the rapid condensation of steam. The most likely time to experience this rapid condensation is at start-up. (See Section 6.3 for guidance on safe start-up.) It can also occur if the pipe or component is exposed to high heat loss (e.g. water sprayed onto the pipe).

### 4.1.4 Incorrect water treatment

Incorrect treatment of the boiler feedwater can cause problems with distribution pipework and heat exchangers (see Section 7) but the main impact is likely to be in terms of cost or loss of efficiency. As far as safety is concerned, one of the issues arising from incorrect water treatment – boiler overheating – has already been discussed.

Other safety-related issues, which are addressed below, are:

— foaming in the boiler resulting in excessive boiler water carryover

— corrosion in pipework resulting in steam/condensate leaks.

*Foaming*

If the TDS level is allowed to rise to a high value, foaming occurs in the boiler and some of this foam is carried by the steam into the distribution pipework. This foam is a mixture of water and boiler solids. It will form scale in pipework and heat exchangers, but a more dangerous factor is its contribution to waterhammer damage, which can be accentuated by the carried over boiler solids reducing the capacity of drain traps.

*Corrosion*

Incoming cold water will contain dissolved gasses, such as carbon dioxide and oxygen. These will be liberated as the water is heated. If this liberation occurs in the boiler, these gases will pass with the steam produced into the distribution pipework. Here, and in heat exchangers, valves and other devices, these gases can cause corrosion due to oxygen pitting or carbonic acid attack. Both of these forms of corrosion are more commonly seen in the condensate return system, where they can eventually result in leaks of hot water and flash steam – dangerous to anyone in the vicinity.

## 4.2 Burning/scalding

A *burn* from a steam system is likely to be caused by coming into contact with an exposed pipe, valve, etc. operating at steam temperature. As there is a direct link between pressure and temperature, the higher the steam pressure, the more severe the burn is likely to be. All exposed surfaces should be insulated, not just for personnel protection, but to reduce energy loss. A burn from a steam system will be similar to one from a MTHW or HTHW system, as they are operating at similar temperatures.

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A *scald* resulting from coming into direct contact with steam will be more severe than a burn. Steam contains a large amount of condensable energy. When it comes into contact with skin, this energy is transferred rapidly, causing severe skin damage. Even when it has condensed, the water produced will be at 100 °C and will exacerbate the injury.

Scalding can be caused by the inappropriate use of steam or by steam leaks. Steam leaks can occur in the same way as with any other piped fluid. However, because they are neither poisonous (like gas) nor make a mess (like oil), they are frequently left unrepaired. Leaks are dangerous and costly, so prevention (for example with the use of bellows-sealed control and isolation valves or with welded pipe connections) is better than cure.

## 4.3 Training

The use of modern automatic control systems does make steam systems and boilers safer pieces of equipment. However, anyone who has to design, operate or maintain a steam boiler or distribution system, or even just respond to an alarm, must be suitably trained. Training will depend on function, but among the factors to consider would be:

— how to design to prevent overheating, overpressure, waterhammer and incorrect water treatment

— how to operate the boiler controls

— how to start up a boiler and steam system safely

— how to shut down a boiler safely

— how to respond to alarms

— how and when to perform routine checks, and who the person responsible for these checks will be

— how to record test and performance results.

This list is by no means exhaustive. Additional training will be needed in specialist areas, such as maintenance. Manufacturers' training courses and manuals are good sources of such training material.

With safety in mind, it is vital that training improves competence levels and consideration should be given to how competence could be demonstrated. Health and safety training, like most other forms of training, should not be considered a one-off event. Regular retraining may be needed and some means of certification can help in the management of this process.

# 5 Design of steam and condensate systems

## 5.1 Calculating steam loads

When steam condenses, it loses its enthalpy of evaporation. The value of this is obtained from steam tables (see Table 1.2). Depending on the design and size of heat exchanger, even when the steam is condensed, some additional heat transfer can take place from the condensate to the secondary product. This is known as 'sub-cooling'. The heat transferred during sub-cooling is considerably less than that transferred during condensation.

The steam reaching the heat exchanger will seldom be 100% dry saturated. It will contain some water caused by heat loss in the distribution pipework. This 'wet' steam has a lower enthalpy of evaporation than dry saturated steam. However, it is common to ignore the reduction in enthalpy and the possible sub-cooling of condensate and simply to assume that heat transfer took place due to the condensing of dry saturated steam.

To establish steam flow rates, the following formula is used:

$$Q = \dot{m}\, h_{fg} \text{ or } \dot{m} = \frac{Q}{h_{fg}}$$

where:

$Q$ = heat transferred (kW)

$\dot{m}$ = steam mass flow rate (kg/s)

$h_{fg}$ = enthalpy of evaporation (kJ/kg).

This is assuming that the value of $Q$ (heat load) is known. If not, it should be calculated with reference to the secondary system conditions.

The calculation of steam consumption is best illustrated by examples, as shown in the following subsections.

### 5.1.1 Non-storage heat exchangers

A non-storage heat exchanger is one in which the secondary fluid (generally water) is pumped through the heat exchanger and around a secondary circuit.

The basic heat energy equation is:

$$Q = \dot{m}\, c_p\, \Delta T$$

where:

Q = heat transferred (kW)

$\dot{m}$ = mass flow rate of secondary fluid (kg/s)

$c_p$ = specific heat capacity of secondary fluid (kJ/kg·°C)

$\Delta T$ = temperature change in secondary fluid (°C).

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**Example**

*Calculate the mean steam load of a non-storage heat exchanger designed to operate at full load with steam at 2.8 bar g in the primary steam space.*

The secondary water flow and return temperatures are 82 °C and 71 °C respectively, at a pumped water flow rate of 7.2 kg/s. The specific heat capacity ($c_p$) for water is 4.19 kJ/kg·°C.

$$Q = \dot{m}\, c_p\, \Delta T$$

$$= 7.2 \times 4.19 \times (82 - 71)$$

$$= 332\ \text{kW}$$

$$\text{steam flow rate } (\dot{m}_s) = \frac{\text{load in kW}}{h_{fg} \text{ at operating pressure}}\ \text{kg/s}$$

Enthalpy of evaporation is determined from steam tables (see Table 5.1).

$$\text{steam flow rate} = \frac{332}{2139} = 0.155\ \text{kg/s or } 558\ \text{kg/h}$$

These two equations are commonly rolled together to give:

$$\dot{m}_s \text{ (steam flow rate)} = \frac{\dot{m}\, c_p\, \Delta T}{h_{fg}}$$

**Table 5.1** Steam table extract

| Pressure | Saturation temperature | Enthalpy (energy) in kJ/kg | | | Specific volume of dry saturated steam |
|---|---|---|---|---|---|
| | | Water | Evaporation | Steam | |
| (bar g) | (°C) | $h_f$ | $h_{fg}$ | $h_g$ | (m³/kg) |
| 2 | 134 | 562 | 2163 | 2725 | 0.603 |
| 2.8 | 142 | 596 | 2139 | 2735 | 0.489 |
| 3 | 144 | 605 | 2133 | 2738 | 0.461 |
| 4 | 152 | 641 | 2108 | 2794 | 0.374 |

### 5.1.2 Hot water storage heat exchangers

For storage heat exchangers, a similar calculation is used, with the secondary flow rate being the mass of water to be heated divided by the heat-up time (recovery period).

The mean rate at which steam is condensed during the heat-up or recovery period can be calculated using:

$$\dot{m}_s \text{ (steam flow rate)} = \frac{m\, c_p\, \Delta T}{h_{fg}\, t}$$

where:

$\dot{m}_s$ = steam flow rate (kg/h)

$m$ = mass of water heated (kg)

$c_p$ = specific heat capacity of water (kJ/kg·°C)

$\Delta T$ = temperature rise (°C)

$h_{fg}$ = enthalpy of evaporation of steam (kJ/kg)

$t$ = recovery time (hours).

**Example**

*Calculate the mean steam load of a storage heat exchanger with a capacity of 2272 litres (2272 kg), designed to raise the temperature of water from 10 °C to 60 °C in half an hour with steam at 2 bar g.*

$c_p$ for water: 4.19 kJ/kg·°C

$$\dot{m}_s \text{ (steam flow rate)} = \frac{m\, c_p\, \Delta T}{h_{fg}\, t}$$

| Pressure | Saturation temperature | Enthalpy (energy) in kJ/kg | | | Specific volume of dry saturated steam |
|---|---|---|---|---|---|
| | | Water | Evaporation | Steam | |
| (bar g) | (°C) | $h_f$ | $h_{fg}$ | $h_g$ | (m³/kg) |
| 2 | 134 | 562 | 2163 | 2725 | 0.603 |

where:

$m$ = 2272 kg

$\Delta T$ = 60 °C – 10 °C = 50 °C

$h_{fg}$ = 2163 kJ/kg

$t$ = 0.5 hours.

$$\dot{m}_s \text{ (steam flow rate)} = \frac{2272\ \text{kg} \times 4.19\ \text{kJ/kg·°C} \times 50\ °\text{C}}{2163\ \text{kJ/kg} \times 0.5\ \text{hours}}$$

$$= 440\ \text{kg/h}$$

If the recovery period was increased to 1 hour, the steam flow rate would be halved to 220 kg/h.

### 5.1.3 Air heater batteries

A similar approach is taken for air heater batteries. Here, the secondary flow rate is normally quoted in *volumetric* terms, so the formulae used have to be modified to take this into account.

$$\dot{m}_s \text{ (steam flow rate)} = \frac{3600\, \dot{v}\, c_p\, \Delta T}{h_{fg}}$$

where:

$\dot{m}_s$ = steam flow rate (kg/h)

$\dot{v}$ = volumetric flow rate of air being heated (m³/s)

$\Delta T$ = air temperature rise (°C)

$c_p$ = specific heat of air at constant pressure (1.3 kJ/m³·°C)

$h_{fg}$ = enthalpy of evaporation of steam in the coils (kJ/kg).

It must be noted that, as we are dealing in *volumetric* flow, the specific heat is quoted in volumetric terms. The constant 3600 gives the solution in kg/h rather than kg/s.

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**Example**

*Calculate the steam load on an air heater battery to raise the temperature of air flowing at 2.3 m³/s from 5 °C to 30 °C with steam at 3.0 bar g in the coils.*

The rating of the battery is unknown, but the steam consumption can be calculated using:

$$\dot{m}_s\,(\text{steam flow rate}) = \frac{3600\,\dot{v}\,c_p\,\Delta T}{h_{fg}}$$

| Pressure | Saturation temperature | Enthalpy (energy) in kJ/kg | | | Specific volume of dry saturated steam |
|---|---|---|---|---|---|
| | | Water | Evaporation | Steam | |
| (bar g) | (°C) | $h_f$ | $h_{fg}$ | $h_g$ | (m³/kg) |
| 3 | 144 | 605 | 2133 | 2738 | 0.461 |

where:

$\dot{m}_s$ = steam flow rate (kg/h)

$\dot{v}$ = air flow rate 2.3 m³/s

$\Delta T$ = air temperature rise 30 °C – 5 °C = 25 °C

$c_p$ = specific heat of air at constant pressure (1.3 kJ/m³ · °C)

$h_{fg}$ = enthalpy of evaporation of steam in the coils 2133 kJ/kg (from steam tables).

$$\dot{m}_s\,(\text{steam flow rate}) = \frac{3600 \times 2.3\,\text{m}^3/\text{s} \times 1.3\,\text{kJ/m}^3\cdot\text{°C} \times 25\,\text{°C}}{2133\,\text{kJ/kg}}$$

$$= 126\,\text{kg/h}$$

*Points to remember*

— The assumption above is that the steam is dry saturated. If it is not, $h_{fg}$ actual = $x\ h_{fg}$ max, where $x$ is steam dryness fraction and $h_{fg}$ max is dry saturated value.

— If the condensate is substantially sub-cooled, the additional heat transferred by this process should be considered. Heat from sub-cooling = condensate (steam) flow rate × 4.19 × °C of sub-cooling.

— The steam pressure should be the value at the heat exchanger inlet, not the main supply pressure.

— Consider the reasons for calculating steam loads. In most cases, a high degree of accuracy is not required.

## 5.2 Boiler selection, sizing and location

### 5.2.1 Boiler selection

The major factors affecting the selection and size of a steam boiler are the output required (kg/h) and the operating pressure. Other considerations would be physical size and number of boilers required to provide the total output needed.

*Number of boilers*

Once the peak steam load is determined, this should be matched to boiler output. How often this peak load occurs will affect the boiler size and number installed. A large boiler will operate very inefficiently if the load reduces below around 20% of maximum. A number of smaller boilers will provide better load matching, and hence higher efficiency, as well as ensuring that there is sufficient cover when the mandatory insurance inspections associated with steam boilers occur.

*Boiler pressure*

The choice of boiler pressure is dependent, first, on the pressure requirements of the load it serves and, second, on the standard pressure ranges available from the manufacturer. To reduce the size of distribution pipework, boilers should be run at high pressure. Any reduced pressure requirements at the point of use should be provided through a pressure reducing valve. A boiler should always be operated at the pressure for which it was designed. There is little to be gained by reducing the pressure setting. There may be a small reduction in heat loss, but any gain will be offset by the steam produced being wet, together with the danger of boiler water contents being drawn into the distribution system.

### 5.2.2 Types of steam boiler

A typical steam boilerhouse and associated plant is shown in Figure 5.1.

The steam boiler is a pressurised vessel containing hot water and steam at more than 100 °C, and its design and operation are covered by a number of standards and regulations.

The objectives of a boiler are:

— to release the energy in the fuel as efficiently as possible, transferring it into heat in the water and subsequently into steam

— to separate the steam from the water ready for export to the plant, where the energy can be transferred to the process.

A number of different steam boiler types have been developed to suit various steam applications. Two of the most popular, and the most common for building services applications, are packaged shell boilers and steam generators.

*Packaged shell boiler*

A typical shell boiler is shown in Figure 5.2.

These boilers are sometimes referred to as fire-tube boilers as the products of combustion pass through tubes which are fitted inside a shell. They may be further defined based on the number of 'passes' the flue gases undertake before exiting the boiler to the chimney. The illustration in Figure 5.3 shows a three-pass boiler, with the first pass being down through the furnace, the second a reversal through the central tubes and the third pass back through the outer tubes to the chimney.

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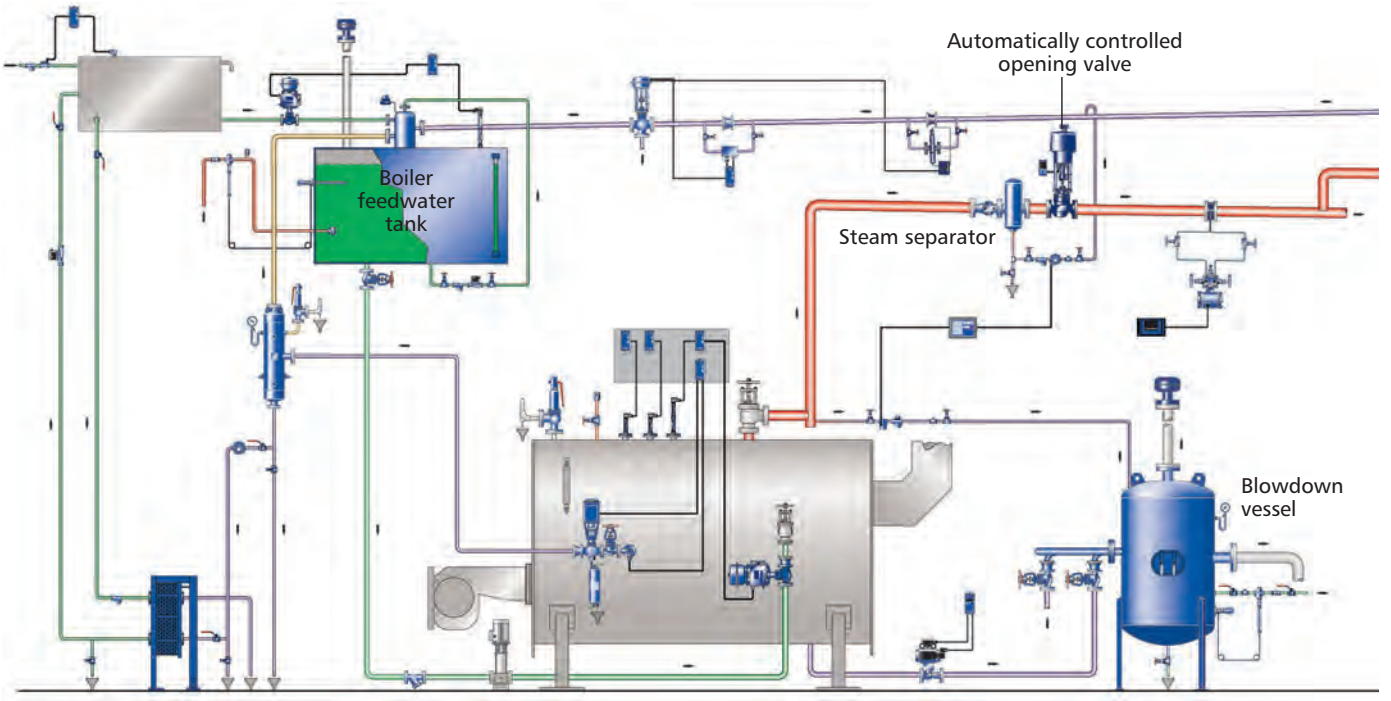



**Figure 5.1** Typical packaged shell boiler and associated plant

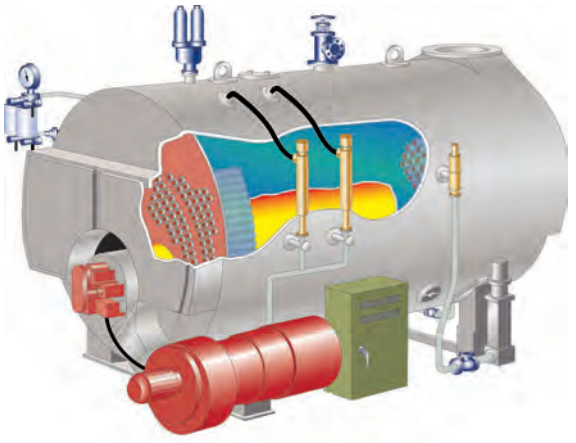


**Figure 5.2** Typical shell boiler

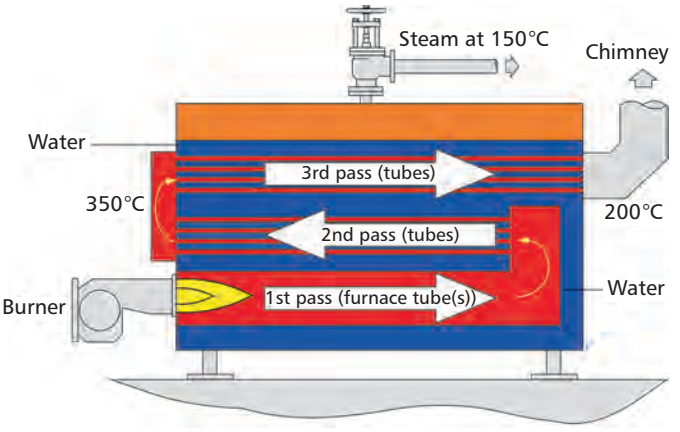



**Figure 5.3** Three-pass, wet back shell boiler

A packaged boiler is one that comes complete with the necessary firing equipment (burner, etc.), pressure and level controls, as well as feedwater pump and blowdown equipment. Packaged shell boilers have a number of advantages:

— the entire plant may be purchased as a complete package, only requiring to be secured to basic foundations and connected to water, electricity, fuel and steam systems before commissioning, minimising installation costs

— this package arrangement also means that it is simple to relocate a packaged shell boiler

— a shell boiler contains a substantial amount of water at saturation temperature, and hence has a substantial amount of stored energy, which can be called upon to cope with short-term, rapidly applied loads (the associated drawback is that, when the energy in the stored water is used, it may take some time before the reserve is built up again)

— the construction of a shell boiler is generally straightforward, simplifying its maintenance requirements

— shell boilers often have a single furnace tube and burner so control systems are relatively simple

— the majority of shell boilers operate at 17 bar or less. This relatively low pressure means that the associated ancillary equipment is readily available at competitive prices.

However, there are also disadvantages associated with packaged shell boilers:

— they take up a significant amount of boilerhouse space

— they are slow to start up

— their initial purchase cost is high.

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### *Steam generators*

A steam generator is an alternative to a shell boiler. Steam generators are generally preferred in situations where:

— the amount of steam required is too small to warrant a shell boiler, i.e. less than 1000 kg/h

— the process requiring steam operates on a day shift basis only, meaning that the plant would be started every morning and shut down every night

— the capital cost of a conventional shell boiler would adversely affect the economic viability of the process

— the level of expertise on site, as far as boilers are concerned, is not as high as would be required for a larger steam system.

To meet these specific demands, two types of steam generator have been developed, the coil boiler and the vertical tubeless packaged boiler. These pass water through the tubes rather than holding it in the shell.

### *Coil boiler*

These are a 'once through' type of water tube boiler and are referred to in some regulations as 'boilers with no discernible water level'. The principle of operation is shown in Figure 5.4.

Water supply to a coil boiler will usually be at 10–15% above the steaming rate to:

— ensure that all the water is not evaporated, thus preventing the production of superheated steam

— provide a vehicle for the feedwater TDS to be carried through. If this vehicle were not available, the salts in the feedwater would be deposited on the insides of the tubes and impair heat transfer, leading to overheating and eventually to tube failure. Clearly, a downstream separator is an essential component of this type of boiler to remove this contaminated water.

### *Vertical tubeless packaged steam boiler (often referred to as a steam generator)*

A boiler of this type is shown in Figure 5.5.

A low water content allows this boiler to be brought up to operating temperature very quickly, typically within 15 minutes. However, this small quantity of water means that only a small amount of energy is stored in the boiler. Consequently, it cannot readily cope with sudden and maintained changes in load. If the load change occurs faster than the boiler can respond, then the pressure inside the boiler will drop and ultimately the boiler will deliver very wet steam. This is aggravated by the small water surface area, which gives high steam release velocities. However, the path of the steam is vertical, up and away from the water surface, as opposed to horizontal, over the water surface (as in a shell boiler), and this reduces the effect.

Typical applications for coil boilers/steam generators include in laundries, where the demand is small and the rate of change in load is slow.

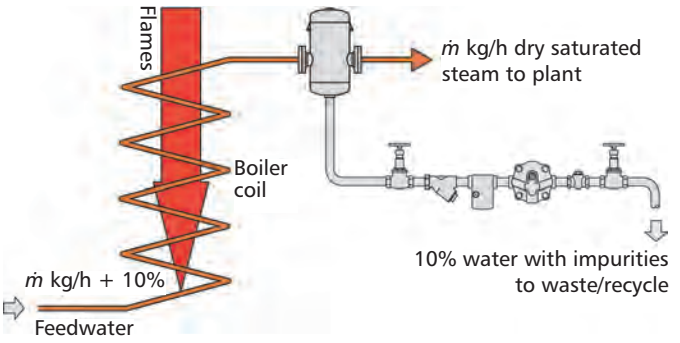



**Figure 5.4** Operating principle of a coil boiler

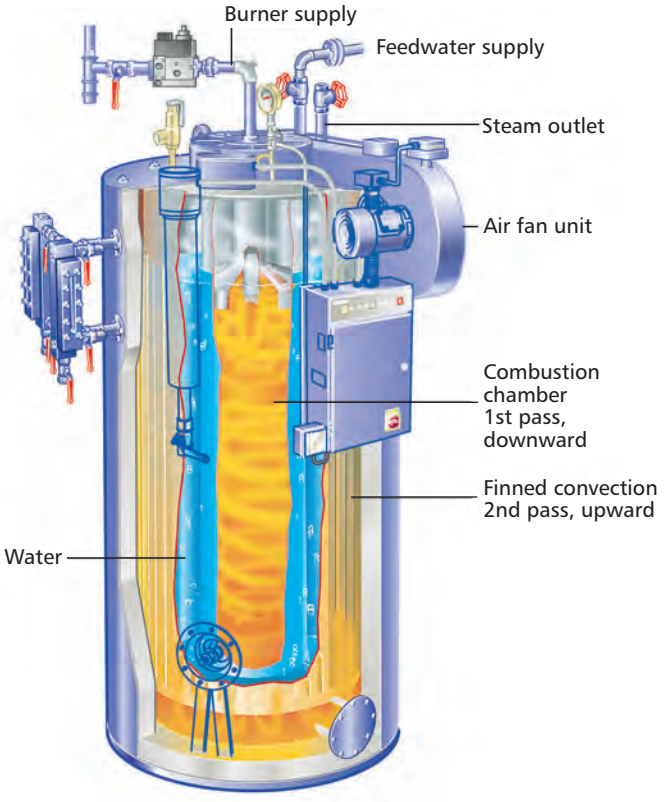



**Figure 5.5** Vertical tubeless packaged steam boiler

## 5.2.3 Boiler sizing

### *Boiler output*

Boilers that provide steam for building services purposes will most commonly be of the shell type, and will provide good quality, dry saturated steam. Coil boilers are also used, but these are mainly suited to smaller loads and where rapid start-up is required. They do not have the same steam space as a shell boiler so will tend to produce wetter steam than the conventional shell type. They overcome this drawback by having a separator installed immediately after the boiler outlet.

To determine the output needed, the load is divided by the enthalpy of evaporation at the operating pressure.

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**Example**

*A steam boiler operating at 7 bar g is required to serve a load of 500 kW. What steam output rate is needed?*

Enthalpy of evaporation at 7 bar g: 2047 kJ/kg (from steam tables)

Steam output:

500 kW/2047 kJ/kg = 0.244 kg/s or 879 kg/h.

*Boiler F&A rating*

Boiler manufacturers commonly use a 'from and at' (F&A) rating to specify boiler output. This is the output from the boiler when supplied with feedwater at 100 °C. In most cases, the inlet temperature will be below this, so additional energy has to be added, effectively reducing the boiler output. Graphs illustrating the output reduction, as shown in Figure 5.6, are readily available.

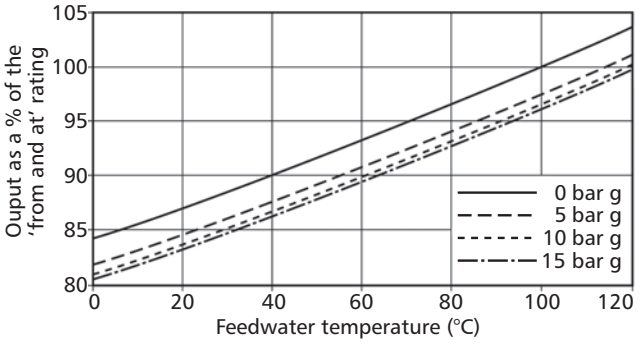



**Figure 5.6** Boiler output based on feedwater temperature

**Example**

*A boiler has an F&A rating of 2000 kg/h and operates at 15 bar g. The feedwater temperature is 68 °C. Using the graph in Figure 5.6, what will be the boiler evaporation rate?*

From Figure 5.6:

the percentage F&A rating ≈ 90%

Therefore:

actual output = 2000 kg/h × 90%

boiler evaporation rate = 1800 kg/h.

*kW rating*

Alternatively, a boiler may be rated in terms of kW output. This is subject to the same F&A reduction, as some of the power (kW) is needed to raise the water temperature to boiling point.

To establish the actual evaporation by mass, it is first necessary to know the temperature of the feedwater and the pressure of the steam produced, in order to determine how much energy is added to each kg of water.

$$\text{steam output (kg/h)} = \frac{\text{boiler rating (kW)} \times 3600}{\text{energy to be added (kJ/kg)}}$$

**Example**

*A boiler is rated at 3000 kW and operates at 10 bar g with a feedwater temperature of 50 °C. How much steam can be generated?*

Energy content of feedwater at 50 °C:

50 °C × 4.19 kJ/kg·°C = 209.5 kJ/kg

Energy content of steam at 10 bar g: 2781 kJ/kg (from steam tables)

Consequently, the boiler must provide:

2781 – 209.5 = 2751.5 kJ/kg

Steam output:

$$= \frac{3000\ \text{kJ/s} \times 3600}{2571.5\ \text{kJ/kg}}$$

= 4200 kg/h

*Dealing with peak loads*

In many circumstances it is impractical to size steam boilers to suit all peak loads. Steam is a compressible fluid, which means that the boiler and distribution system are capable of providing some storage capacity to cope with such peak loads. However, the modern steam boiler has significantly less storage capability than its predecessors as manufacturers now fill the shell with heat transfer surfaces rather than steam/water.

A symptom of an overloaded steam boiler is a drop in boiler pressure. This drop can cause level control problems as well as the production of poor quality (wet) steam.

Methods of overcoming this problem are, in order of cost:

— reschedule load to smooth out peaks

— install surplussing valves that will close down non-essential loads if the steam supply pressure drops

— fit a steam accumulator – a device which stores high (boiler) pressure water and releases it as steam when the pressure drops

— install a bigger or an additional boiler.

The selection and sizing of a boiler is a complex issue so manufacturers' guidance should always be sought.

### 5.2.4 Location of boilerhouse – centralised or decentralised boiler plant?

Modern steam boilers come in a wide range of sizes to suit both large and small applications. Generally, where more than one boiler is required to meet the demand, it becomes economically viable to house the boiler plant in a centralised location, as installation and operating costs can be significantly lower than with decentralised plant.

Neither centralised nor decentralised boiler plant is universally appropriate; the decision must be made based

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on the circumstances of each site. A number of factors will have a bearing on the choice, including:

— the size of the site and location of the steam users, e.g. whether the site is large and the steam-using plant widely scattered or whether the steam users are grouped

— the importance of the steam quality, which will be determined by the uses to which the steam will be put, e.g. process, space heating, direct contact with food or drink, autoclaves, etc. The decision may also be influenced by the proportion of steam to be used for each purpose and whether there will be seasonal variations, e.g. steam for space heating being switched off during summer months

— steam demand patterns, e.g. how often the boiler(s) will be shut down, whether there will be sudden high steam demands, how often the peak loads will occur and how they compare with the normal running load

— the choice, availability and cost of fuel, e.g. gas, oil or solid fuel, which needs storage space, and how accessible the boiler(s) will be for fuel deliveries or gas supplies

— security of fuel supply – whether dual-fired burners are necessary and, if so, how easily storage of fuel oil, for example, can be accommodated

— if electrically heated boilers are being considered, whether power is available at all locations

— whether there are any restrictions on the location and size of flues

— what staffing and skills are available

— what provisions are necessary if boilers are to be located near to populated areas and whether there are any environmental issues to consider

— how the site and demand for steam might develop in the future

— whether technologies such as combined heat and power (CHP) can be used

— financial restraints.

Traditionally, boiler plant was centralised to simplify fuel oil or coal storage and ash removal. In modern times, gas has become the preferred fuel and oil storage tanks and coal storage have been removed. However, more recently, driven by rising costs and climate change, the benefits of using biomass and waste products, such as wood chips and paper, make these increasingly attractive to boiler operators, so some type of storage facility might again become necessary.

The recent resurgence in co-generation and tri-generation will also see more organisations using CHP and generating steam from the exhaust gases. The use of waste steam in absorption chillers is also on the increase.

These, and other factors, make it more difficult for the design engineer to analyse and decide whether to opt for central or dispersed steam boiler plant. There are advantages and disadvantages of both arrangements and the main points for consideration are listed in Table 5.2.

## 5.3 Design of steam headers and off-takes

### 5.3.1 Multi-boiler installations

Where multiple boilers are installed on a load-sharing basis (as opposed to duty and standby), it is important that the boilers are connected in such a way as to ensure that the load is shared as required. This will generally be equal steam quantities from each boiler.

Consider the arrangement shown in Figure 5.7 where all four boilers are operating at the same pressure.

For steam to flow from point B to point A, the pressure at A must be lower than at B. Likewise, pressure at C must be greater than at B and A. Steam flow rate depends on pressure drop so, on the basis that all boilers are operating at the same pressure, boiler 4 must produce the most steam as the pressure drop from the boiler to point A is greater than from boiler 3 to B, etc.

Consequently, boiler 4 will handle most of the steam load and, in a typical four-boiler installation as shown, the difference in output between boiler 4 and boiler 1 can be as much as 15%.

If true balanced outputs are required from each boiler, the pressure drop from each boiler to the common output line must be the same. This is achieved using the arrangement shown in Figure 5.8.

### 5.3.2 Steam distribution headers and other off-takes

There is a common requirement to see more than one steam supply pipe exit from a boilerhouse or plantroom. A convenient way to ensure that the steam is distributed with minimal pressure loss and in a dry and clean condition is to use a main distribution header. A typical example is shown in Figure 5.9.

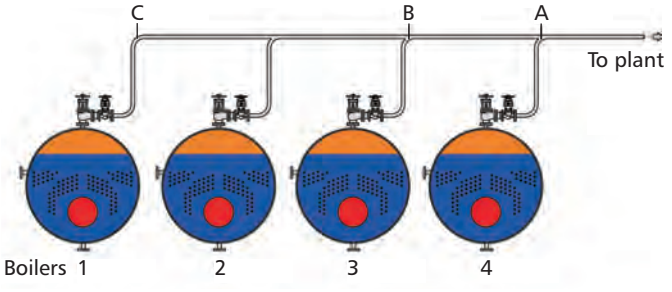



**Figure 5.7** Boilers in series

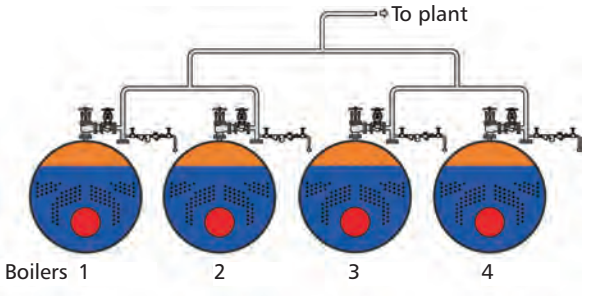



**Figure 5.8** Boilers in 'ladder' formation

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**Table 5.2** Boiler plant location comparison

| Centralised boiler plant | Positive = ✓ Negative = ✗ | Decentralised boiler plant | Positive = ✓ Negative = ✗ |
|---|---|---|---|
| Steam is provided from a single boilerhouse. Less flexible to upgrade in the future | ✗ | Steam is provided from a number of boilers, each located near to the using plant. Easier to upgrade in the future | ✓ |
| Can be sized more economically than dispersed plant. Applying a diversity factor for the whole site means that the centralised boiler plant does not usually have to cope with the sum of the connected loads | ✓ | Lower diversity factors for smaller areas may make it necessary to size each boiler for the total connected load | ✗ |
| Central plant may not always be operating at optimum loading and could therefore be less efficient. Less flexible for handling seasonal variations in steam demand. A boiler which is oversized for light loads will be less efficient | ✗ | Smaller boilers may operate at optimum load for longer periods. Flexible enough to handle seasonal loads, e.g. boilers can be shut off when not required for space heating | ✓ |
| Good response to high sudden variations in steam demand due to high water volume | ✓ | Smaller storage capacity with reduced ability to handle sudden peak demands, can lead to water carryover and low water lock out | ✗ |
| Large financial investment for boiler plant and civil work | ✗ | Also a large investment. A number of smaller boilers and associated plant are likely to cost significantly more than a larger centralised boiler, but civil work will be less | ✗ / ✓ |
| Permits concentration of staffing in one location. Fewer boilers to operate, monitor and maintain | ✓ | Multiple boilerhouses required. Multiple boilers to monitor and maintain – modern boilers can be fitted with high integrity safety controls and monitored from a remote location | ✗ |
| One central storage area for solid or liquid fuels. Flexibility of choice of fuels and price negotiation (fuel can be bought and stored at 'yesterday's' prices) | ✓ | Choice of fuels may be limited because of storage requirements. Alternatively, gas mains required at each location | ✗ |
| More practical to employ instrumentation, burner modulations and oxygen trim to maintain high efficiency | ✓ | Small boilers do not usually achieve the same efficiency as large boilers | ✗ |
| Easier to justify fitting an economiser to recover heat from the boiler exhaust gases | ✓ | More difficult to justify the cost of economisers to recover heat from flue gases | ✗ |
| Easier to justify automatic level control, automatic blowdown and heat recovery from blowdown | ✓ | More difficult to justify modulating feedwater controls. More difficult to maintain consistent steam quality with on/off controls. More difficult to justify automatic blowdown and heat recovery | ✗ |
| Steam from central boiler plant, distributed in carbon steel pipework, may not be compatible for direct use in food and drinks or hospital autoclaves | ✗ | Local 'clean' steam generators using factory steam as the energy source can be used where required | ✓ |
| Only one standby boiler required | ✓ | Multiple standby boilers may be required | ✗ |
| Steam can be produced using a waste heat boiler in conjunction with a CHP system | ✓ | More difficult to use waste heat for multiple boilers | ✗ |
| Long lengths of large size steam distribution pipework required, which needs draining and insulating, and longer lengths of condensate pipe. Higher heat losses. Condensate return may be prohibitive on a large site | ✗ | Short and smaller diameter steam and condensate pipes, lower heat losses, reduced maintenance, lower installation costs. Maximum potential for condensate return | ✓ |
| Needs more time to start up from cold | ✗ | Steam can be raised more quickly than in a large boiler | ✓ |
| Boiler operated at one pressure | ✗ | Boilers can be operated at different pressures if required, e.g. high pressure for local process plant | ✓ |
| Flues and vents from boiler feedtank and safety valves in one location | ✓ | Multiple flues and vents. Cleaner, lower maintenance electric boilers (without flues) can be used for small duties | ✗ |
| Future disposal of plant from a central boilerhouse, separate from other buildings, is easier and less disruptive | ✓ | Disposal of plant from boilerhouses located within other buildings, or adjacent to them, can be more difficult and disruptive | ✗ |

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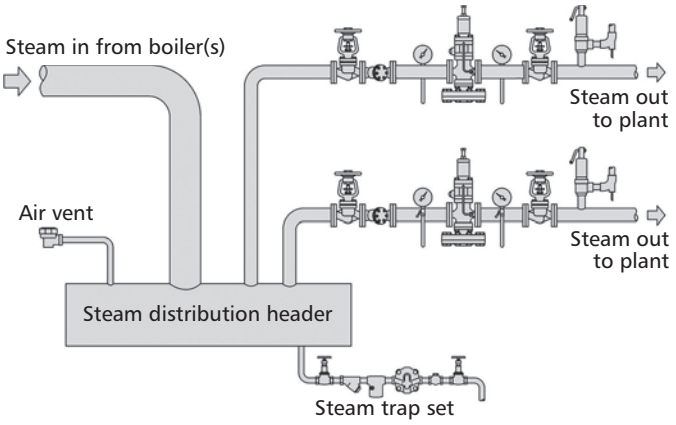



**Figure 5.9** Steam distribution header

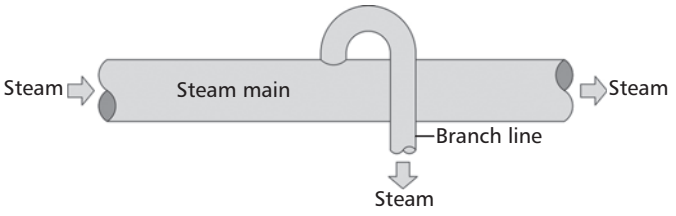



**Figure 5.10** Steam off-take

The important design considerations for such a header are:

— the header diameter should be calculated with a maximum steam velocity of 15 m/s under full load conditions. This will ensure that the header acts as a separator, with any moisture present able to drop out

— there must be provision for releasing any air and other non-condensable gases that could accumulate

— the header must be fitted with a drain trap set at the lowest point to ensure that any water (condensate) that gathers is continuously removed

— all of the steam off-takes should come from the top of the header and be fitted with individual isolation valves (and pressure controls if appropriate).

Whenever a steam off-take is required from a distribution system, it should come from the top of the pipe (Figure 5.10) to ensure that the steam is dry. All low points must be fitted with steam trap sets to allow condensate removal.

## 5.4 Steam flow metering

### 5.4.1 Why measure steam flow?

Measuring steam flow is essential for good housekeeping. It provides knowledge of steam usage and cost, which is vital to the efficient operation of a plant or building. The main benefits of using steam flow metering include:

— plant efficiency

— energy efficiency

— costing and custody transfer.

*Plant efficiency*

By analysing the relationship between steam flow and production, optimum working practices can be determined.

The flowmeter may also be used to:

— track steam demand and changing trends

— establish peak steam usage times

— identify sections or items of plant that are major steam users.

This may lead to changes in production methods to ensure economical steam usage. It can also reduce problems associated with peak loads on the boiler plant.

*Energy efficiency*

Steam flowmeters can be used to monitor the results of energy-saving schemes and to compare the efficiency of one piece of plant or building with another.

*Costing and custody transfer*

Steam flowmeters can measure steam usage (and thus steam cost) either centrally or at individual user points. Where steam is used in a production process, such as a hospital laundry or steriliser centre, it can be costed as a raw material thus allowing the true cost of production to be calculated.

### 5.4.2 Terminology

When discussing flow metering, the terms repeatability, uncertainty, accuracy and turndown are commonly used.

*Repeatability*

This term describes the ability of a flowmeter to indicate the same value for an identical flow rate on more than one occasion. Good repeatability is important where steam flow metering is required to monitor trends rather than accuracy.

*Uncertainty*

The term ‘uncertainty’ is now becoming more commonly used than accuracy. This is because accuracy cannot be established, as the true value can never be exactly known. However, ‘uncertainty’ can be estimated and an ISO standard exists offering guidance on this matter (BS EN ISO/IEC 17025).

*Accuracy*

This is a measure of a flowmeter’s performance when indicating a correct flow rate value against a ‘true’ value obtained by extensive calibration procedures.

The two methods used to express accuracy, which are detailed below, have very different meanings.

(1) Percentage of measured value or actual reading

For example, a flowmeter’s accuracy is given as ±3% of actual flow.

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At an indicated flow rate of 1000 kg/h, the 'uncertainty' of actual flow is between:

1000 kg/h – 3% = 970 kg/h

and

1000 kg/h + 3% = 1030 kg/h.

Similarly, at an indicated flow rate of 500 kg/h, the error is still ±3%, and the 'uncertainty' is between:

500 kg/h – 3% = 485 kg/h

and

500 kg/h + 3% = 515 kg/h.

(2) Percentage of full scale deflection (FSD)

A flowmeter's accuracy may also be given as ±3% of FSD, when the measurement error is expressed as a percentage of the maximum flow that the flowmeter can handle.

As in the previous case, the maximum flow = 1000 kg/h.

At an indicated flow rate of 1000 kg/h, the 'uncertainty' of actual flow is between:

1000 kg/h – 3% = 970 kg/h

and

1000 kg/h + 3% = 1030 kg/h.

At an indicated flow rate of 500 kg/h, the error is still ±30 kg/h, and the actual flow is between:

500 kg/h – 30 kg/h = 470 kg/h (an error of –6%)

and

500 kg/h + 30 kg/h = 530 kg/h (an error of +6%).

As the flow rate is reduced, the percentage error increases.

*Turndown*

'Turndown' or 'turndown ratio', 'effective range' or 'rangeability' are all terms used to describe the range of flow rates over which the flowmeter will work within the accuracy and repeatability values quoted:

$$\text{turndown} = \frac{\text{maximum flow}}{\text{minimum flow}}$$

Using the previous example, if the turndown of the flowmeter selected is given as 4:1, the accuracy can be met at a minimum flow rate of 1000 ÷ 4 = 250 kg/h.

When the steam flow rate is lower than this, the flowmeter cannot meet its specification, so large flow errors occur.

### 5.4.3 Types of steam flowmeter

Many types of flowmeter are available, the main ones used to measure steam flow being:

— orifice plate

— turbine (including shunt or bypass types)

— spring-loaded variable area

— vortex shedding.

Other less common types of meter that can be used on steam systems would include averaging pitot tube, coriolis and ultrasonic meters.

Each of these meter types has its own advantages and limitations. To ensure accurate and consistent performance from a steam flowmeter, it is essential to match the meter to the application.

*Orifice plate flowmeters*

Orifice plate flowmeters are the most commonly used means of measuring fluid flow. In simple terms, the pipeline fluid is passed through a restriction, and the pressure differential is measured across that restriction. The relationship between the velocity of fluid passing through the orifice is that it is proportional to the square root of the pressure loss across it.

To measure the differential pressure when the fluid is flowing, connections are made from the upstream and downstream pressure tappings to a secondary device known as a differential pressure (DP) cell (see Figure 5.11).

Because an orifice plate flowmeter operates within a limited turndown ratio of between 4:1 and 5:1, it is best suited to applications where flow rate changes are least severe. In a building services application, this is likely to restrict its use to the main boilerhouse.

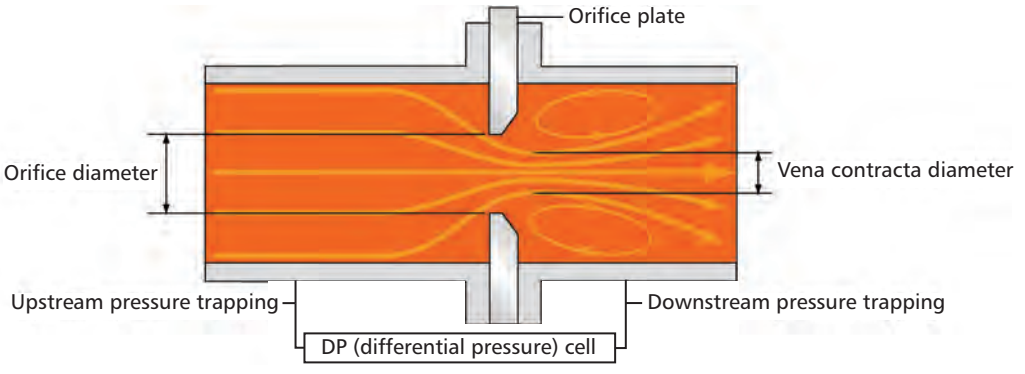



**Figure 5.11** Orifice plate flowmeter

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### *Turbine flowmeters*

The primary element consists of a multi-bladed rotor which is mounted at right angles to the flow and suspended in the fluid stream on a free-running bearing. Its speed of rotation is proportional to the volumetric flow rate. The speed may be determined using an electronic proximity switch mounted on the outside of the pipework, which counts the pulses, as shown in Figure 5.12.

In larger pipelines, to minimise cost, the turbine element can be installed in a pipework bypass, or the flowmeter body may even incorporate a bypass or shunt, as shown in Figure 5.13.

Turbine flowmeters are not ideal for the aggressive environment experienced inside a steam pipe, but their use in an insertion mode can make them an economic solution for larger pipes. They can operate over a turndown ratio of 10:1. They are not generally seen in building services applications today, but reference is made to them here as the shunt meter was commonly used in older systems.

### *Spring-loaded variable area flowmeters*

A spring-loaded variable area (SLVA) meter works on the principle that a flap (Figure 5.14) or cone (Figure 5.15) moves against a spring as the flow changes, hence exposing a greater or lesser area to the steam flow. Effectively these are variable orifice-type devices. Turndown ratio is significantly better than a simple orifice plate as it can be up to 100:1 in some models.

The flow rate is determined by measuring the position of the flap or the differential pressure across the cone.

The high turndown ratio achieved by SLVA meters makes them well suited to measuring steam flow at point of use, where more variations in flow rate are likely to be seen.

### *Vortex-shedding flowmeters*

These flowmeters exploit the fact that, when a non-streamlined or 'bluff' body is placed in a fluid flow, regular vortices are shed from the rear of the body (Figure 5.16). These vortices can be detected, counted and displayed. Over a range of flows, the rate of vortex shedding is proportional to the velocity, and this allows the flow rate to be measured.

A vortex meter offers reasonable turndown (25:1 max.), with no moving parts, little resistance to flow and, hence, minimum pressure drop. The disadvantages are their sensitivity to pipe vibration and the high velocities needed to achieve the higher turndown ratios specified.

Typical applications for vortex-shedding flowmeters would be steam measurement at both boiler and point of use locations. They are also commonly used to measure natural gas flow rates to boilers.

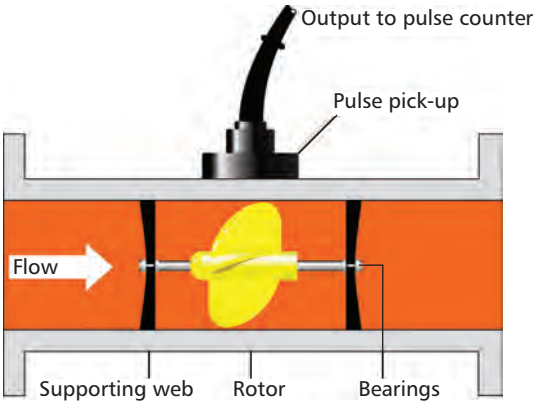



**Figure 5.12** Turbine flowmeter

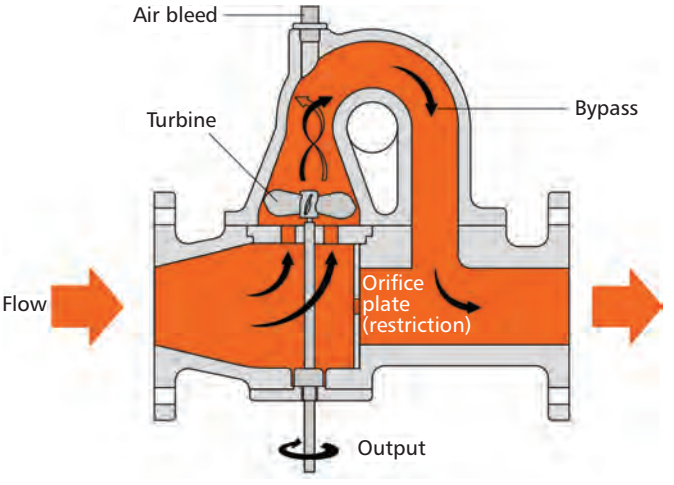



**Figure 5.13** Bypass or shunt turbine flowmeter

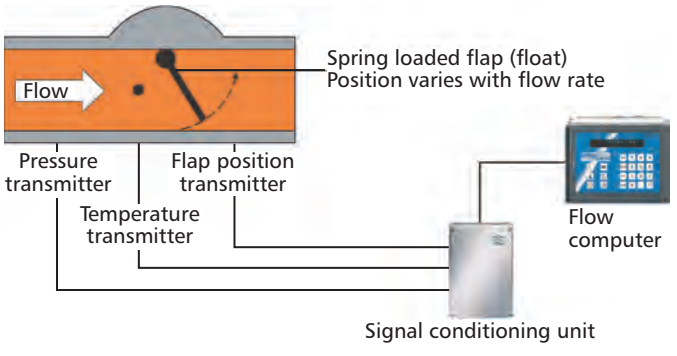



**Figure 5.14** SLVA flowmeter monitoring the position of the flap

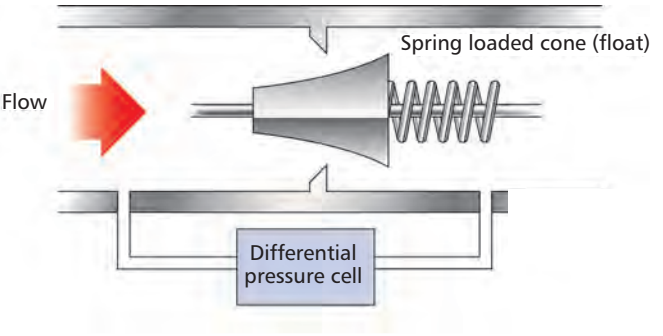



**Figure 5.15** SLVA flowmeter monitoring differential pressure

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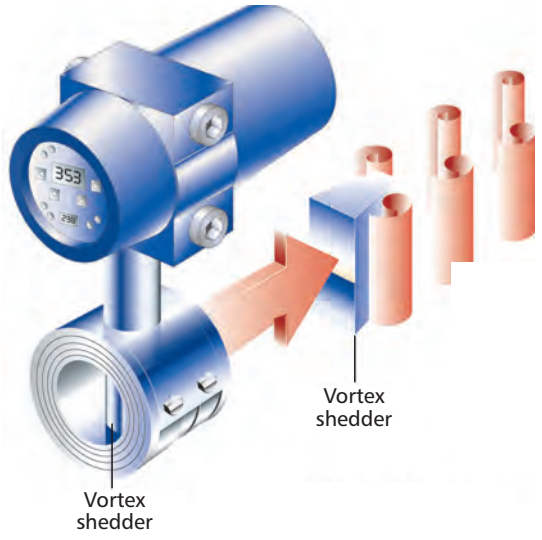



**Figure 5.16** Vortex shedding flowmeter

### 5.4.4 Installing a steam meter

*Orifice plate flowmeters*

Orifice plate flowmeters have some of the most exacting installation requirements, which are necessary to prevent turbulence in the flow through the orifice.

There is a requirement for straight pipework a minimum of five pipe diameters in length downstream of the orifice plate, in order to reduce the effects of disturbance caused by the pipework.

The amount of straight pipework required upstream of the orifice plate is, however, affected by a number of factors, including the nature and geometry of any preceding obstruction. Some example installations are shown in Figure 5.17.

*Turbine flowmeters*

The main considerations for the successful installation and operation of a turbine flowmeter are listed below:

— flow straighteners are essential

— wet steam can damage the turbine wheel and affect accuracy

— the fluid must be very clean (particle size not greater than 100 $\mu$m).

Figure 5.18 details the elements needed to dry (separator) and clean (strainer) the steam.

*Spring-loaded variable area flowmeters*

The requirements for clear lengths of upstream and downstream pipework are less exacting for SLVA meters than for any other type. This makes them well suited to situations where space in a plantroom is at a premium.

As with all meters, clean and dry steam will significantly improve the meter's performance and prolong its life. Figure 5.18 shows a recommended installation.

*Vortex-shedding flowmeters*

Like orifice plate meters, there are some critical requirements for a successful vortex meter installation. Careful installation is essential as a protruding gasket or weld beads can cause vortices to form, leading to inaccurate readings. Long, clear lengths of upstream pipework must be provided, as in the case of orifice plate flowmeters (see Figure 5.19).

## 5.5 Steam pipe sizing

### 5.5.1 Getting the size right

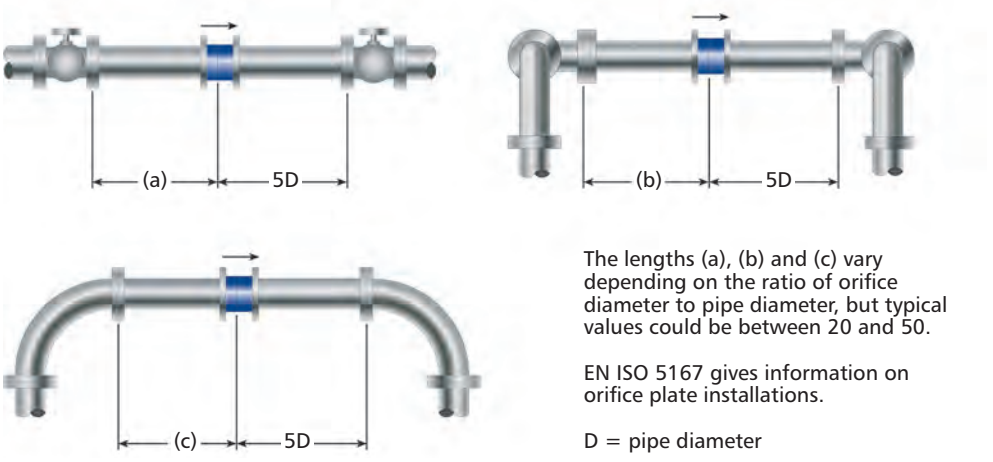
The aim of a steam distribution system should be to deliver steam that is:

— *at the right pressure*: steam pipes must be adequately sized

— *dry*: steam pipes need to be well laid out and drained

— *free of air* and other incondensable gases: steam pipes need to be vented of air

— *clean*: correct boiler feedwater treatment will minimise corrosion; strainers should be fitted.




The lengths (a), (b) and (c) vary depending on the ratio of orifice diameter to pipe diameter, but typical values could be between 20 and 50.

EN ISO 5167 gives information on orifice plate installations.

D = pipe diameter

**Figure 5.17** Orifice plate installations

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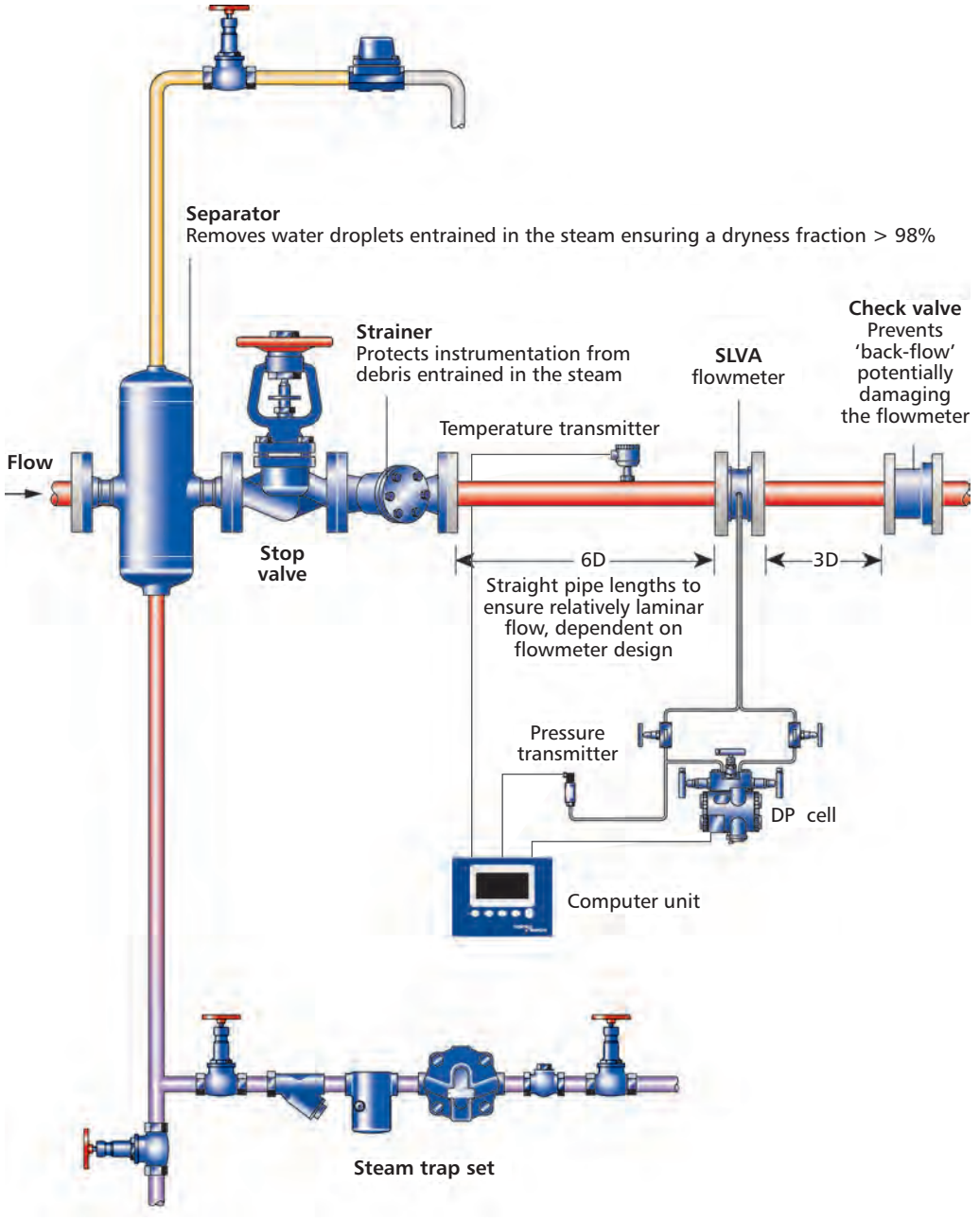
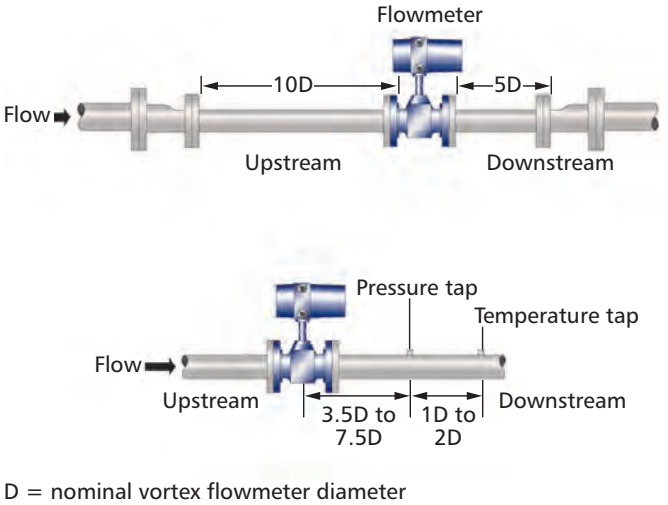



**Figure 5.18** Typical installation of an SLVA flowmeter




D = nominal vortex flowmeter diameter

**Figure 5.19** Vortex-shedding flowmeter – typical installations

Oversized pipework results in:

— pipes, valves, fittings, etc. being more expensive than necessary

— higher installation costs being incurred, including support work, insulation, etc.

— a greater volume of condensate forming due to the greater heat loss. This in turn means that either more steam trapping is required, or wet steam is delivered to the point of use.

Undersized pipework results in:

— lower pressure being available at the point of use

— a risk of steam starvation (a lack of steam at the point of use)

— a greater risk of erosion, waterhammer and noise due to the inherent increase in steam velocity.

To keep pipe sizes and costs as small as possible, steam should be distributed at high pressure to utilise high-pressure steam's lower specific volume (more kg per $m^3$).

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Steam should be used at low pressure because of its higher enthalpy of evaporation per kg, meaning that a lower mass flow rate of steam is needed. The temperature of the steam required at the point of use is achieved by controlling the pressure using a pressure reducing control valve or a temperature control valve. Steam pipes should be sized to carry steam at the required flow rate and with an acceptable velocity and pressure drop.

Two sizing methods are commonly employed:

— sizing based on the steam's velocity – this is quick and easy and the most common method of pipe sizing but it provides no guarantee of pressure at the using end; it is normally used for sections of pipes up to 50 m long

— sizing based on a specified pressure drop along the length being sized.

### 5.5.2 Sizing on velocity

*Lines under 50 m*

For pipe lengths (between take-off points) of up to 50 m, reasonable velocities for dry saturated steam are between 25 and 40 m/s. Higher velocities can lead to higher pressure drop and possible noise. The longer the length, the lower the velocity should be, and for pipes longer than 50 m, velocities should not exceed 25 m/s.

The quality (dryness) of the steam should also be considered when selecting an appropriate velocity. Due to the erosive nature of water droplets, pipework carrying wet steam should be sized on a target velocity of approximately 15 m/s. Table 5.3 can be used for sizing pipes by velocity.

*Note*: inside diameters of pipe other than Schedule 40 (e.g. DIN 2448 or EN 10255) will result in different steam

**Table 5.3** Saturated steam pipeline capacities in kg/h for different velocities (Schedule 40 pipe)

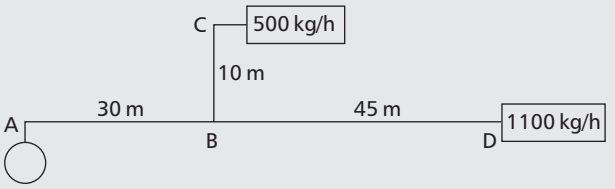
| Pressure | Velocity | Pipe size (nominal) (mm) | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | 15 | 20 | 25 | 32 | 40 | 50 | 65 | 80 | 100 | 125 | 150 |
| | | Actual inside pipe diameter (Schedule 40) (mm) | | | | | | | | | | |
| | | 15.80 | 20.93 | 26.64 | 35.04 | 40.90 | 52.50 | 62.10 | 77.92 | 102.26 | 128.20 | 154.05 |
| (bar g) | (m/s) | Pipeline capacity (kg/h) | | | | | | | | | | |
| 0.4 | 15 | 9 | 15 | 25 | 43 | 58 | 95 | 136 | 210 | 362 | 569 | 822 |
| | 25 | 14 | 25 | 41 | 71 | 97 | 159 | 227 | 350 | 603 | 948 | 1369 |
| | 40 | 23 | 40 | 66 | 113 | 154 | 254 | 363 | 561 | 965 | 1517 | 2191 |
| 0.7 | 15 | 10 | 18 | 29 | 51 | 69 | 114 | 163 | 251 | 433 | 681 | 983 |
| | 25 | 17 | 30 | 49 | 85 | 115 | 190 | 271 | 419 | 722 | 1135 | 1638 |
| | 40 | 28 | 48 | 78 | 136 | 185 | 304 | 434 | 671 | 1155 | 1815 | 2621 |
| 1 | 15 | 12 | 21 | 34 | 59 | 81 | 133 | 189 | 292 | 503 | 791 | 1142 |
| | 25 | 20 | 35 | 57 | 99 | 134 | 221 | 315 | 487 | 839 | 1319 | 1904 |
| | 40 | 32 | 56 | 91 | 158 | 215 | 354 | 505 | 779 | 1342 | 2110 | 3046 |
| 2 | 15 | 18 | 31 | 50 | 86 | 118 | 194 | 277 | 427 | 735 | 1156 | 1669 |
| | 25 | 29 | 51 | 83 | 144 | 196 | 323 | 461 | 712 | 1226 | 1927 | 2782 |
| | 40 | 47 | 82 | 133 | 230 | 314 | 517 | 737 | 1139 | 1961 | 3083 | 4451 |
| 3 | 15 | 23 | 40 | 65 | 113 | 154 | 254 | 362 | 559 | 962 | 1512 | 2183 |
| | 25 | 38 | 67 | 109 | 188 | 256 | 423 | 603 | 931 | 1603 | 2520 | 3639 |
| | 40 | 61 | 107 | 174 | 301 | 410 | 676 | 964 | 1490 | 2565 | 4032 | 5822 |
| 4 | 15 | 28 | 50 | 80 | 139 | 190 | 313 | 446 | 689 | 1186 | 1864 | 2691 |
| | 25 | 47 | 83 | 134 | 232 | 316 | 521 | 743 | 1148 | 1976 | 3106 | 4485 |
| | 40 | 75 | 132 | 215 | 371 | 506 | 833 | 1189 | 1836 | 3162 | 4970 | 7176 |
| 5 | 15 | 34 | 59 | 96 | 165 | 225 | 371 | 529 | 817 | 1408 | 2213 | 3195 |
| | 25 | 56 | 98 | 159 | 276 | 375 | 619 | 882 | 1362 | 2347 | 3688 | 5325 |
| | 40 | 90 | 157 | 255 | 441 | 601 | 990 | 1411 | 2180 | 3755 | 5901 | 8521 |
| 6 | 15 | 39 | 68 | 111 | 191 | 261 | 430 | 613 | 947 | 1631 | 2563 | 3700 |
| | 25 | 65 | 114 | 184 | 319 | 435 | 716 | 1022 | 1578 | 2718 | 4271 | 6167 |
| | 40 | 104 | 182 | 295 | 511 | 696 | 1146 | 1635 | 2525 | 4348 | 6834 | 9867 |
| 7 | 15 | 44 | 77 | 125 | 217 | 296 | 487 | 695 | 1073 | 1848 | 2904 | 4194 |
| | 25 | 74 | 129 | 209 | 362 | 493 | 812 | 1158 | 1788 | 3080 | 4841 | 6989 |
| | 40 | 118 | 206 | 334 | 579 | 788 | 1299 | 1853 | 2861 | 4928 | 7745 | 11 183 |
| 8 | 15 | 49 | 86 | 140 | 242 | 330 | 544 | 775 | 1198 | 2063 | 3242 | 4681 |
| | 25 | 82 | 144 | 233 | 404 | 550 | 906 | 1292 | 1996 | 3438 | 5403 | 7802 |
| | 40 | 131 | 230 | 373 | 646 | 880 | 1450 | 2068 | 3194 | 5501 | 8645 | 12 484 |
| 10 | 15 | 60 | 105 | 170 | 294 | 401 | 660 | 942 | 1455 | 2506 | 3938 | 5686 |
| | 25 | 100 | 175 | 283 | 490 | 668 | 1101 | 1570 | 2425 | 4176 | 6563 | 9477 |
| | 40 | 160 | 280 | 453 | 785 | 1069 | 1761 | 2512 | 3880 | 6682 | 10 502 | 15 164 |
| 14 | 15 | 80 | 141 | 228 | 394 | 537 | 886 | 1263 | 1951 | 3360 | 5281 | 7625 |
| | 25 | 134 | 235 | 380 | 567 | 896 | 1476 | 2105 | 3251 | 5600 | 8801 | 12 708 |
| | 40 | 214 | 375 | 608 | 1052 | 1433 | 2362 | 3368 | 5202 | 8960 | 14 082 | 20 333 |

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velocities but the overall effect for pipe sizing purposes in building services will be minimal.

**Example**

*In the arrangement below, find the appropriate size for pipe runs A–B, B–C and B–D.*




**Length A–B is a main distributor, 30 m long**

This section is to carry 1600 kg/h of steam at 7 bar g.

Table 5.3 shows that a DN65 pipe will carry 1600 kg/h of steam with an acceptable velocity of between 25 and 40 m/s.

**Length B–C can be considered as a branch line**

This section is to carry 500 kg/h of steam at approximately 7 bar g.

Sizing solely on velocity means that the pressure at point B is not known. Therefore consider the total length as being from point A to point C in order to determine which velocity would be appropriate. As the total length of 40 m from A to C is less than 50 m, section B–C is sized on an upper velocity limit of 40 m/s.

Table 5.3 shows that, for a pressure of 7 bar g, a DN32 pipe will carry 500 kg/h of steam with an acceptable velocity not exceeding 40 m/s.

**Length B–D can be considered as a branch line**

This section is to carry 1100 kg/h of steam at approximately 7 bar g.

The pressure at point B is unknown. Consider the total length as being from point A to point D. As the total length of 75 m from A to D is greater than 50 m, size section B to D on an upper velocity limit of 25 m/s to prevent undue pressure drop.

Table 5.3 shows that a DN65 pipe will carry 1100 kg/h of steam with a velocity of just below 25 m/s.

### 5.5.3 Sizing on pressure drop

*Lines longer than 50 m*

The longer the pipeline, the greater the pressure drop, even if the steam velocity is within acceptable limits. Longer pipelines should therefore be sized based on pressure drop. Graphs can be used to give satisfactory results. One such graph is shown in Figure 5.20.

Graphs such as this are adequate for quickly establishing pipe sizes or pressure drops. They are not sufficiently detailed to determine accurate or numerous calculations and, for these, pipe sizing software should be used.

**Example 1 (shown by the red line on Figure 5.20)**

*Calculate the pressure drop along a 150 m long DN100 pipeline supplied with dry saturated steam at 7 bar g, where the flow rate is 2000 kg/h.*

From the 7 bar g saturation line, project horizontally to intersect with the 2000 kg/h flow rate. Project upwards to intersect with 100 mm pipe diameter and then left to the pressure loss per 100 m axis.

The graph tells us we have a pressure drop of 0.2 bar for every 100 m, so for 150 m the pressure drop would be:

$$0.2 \times 1.5 = 0.3 \text{ bar.}$$

**Example 2 (shown by the blue line on Figure 5.20)**

*Determine the size of steam pipe needed to handle 500 kg/h dry saturated steam. The pipe is 50 m long. Pressure at the start is 10 bar g. Pressure drop is to be no more than 0.5 bar.*

A pressure drop of 0.5 bar in 50 m is equal to 1.0 bar per 100 m. Project right from this point.

From the 10 bar g saturation line, project horizontally to intersect with the 500 kg/h flow rate. Project upwards to intersect with the 1.0 bar per 100 m horizontal line to indicate that a DN40 pipe would be selected.

## 5.6 Steam pipe layout and drainage

Any water carried over from the boiler or formed by steam condensing due to heat loss must be removed from the pipework system. The rate of condensation due to heat loss will be highest at plant start-up, when the system is cold. Beyond this period, the rate is much lower.

Condensate forming on the inside of pipes will gravitate to the bottom of the pipes and will be swept along by the steam flow. Appropriately sized drain points at regular intervals, fitted with steam traps, will help to prevent droplets being re-entrained. This will minimise erosion of pipes, particularly at bends and valves. It is vital that every low point in the pipe is drained to prevent condensate collecting and leading to possible waterhammer damage. A few simple rules regarding layout and drainage will ensure a trouble-free system.

### 5.6.1 Pipe layout

Steam pipes should be installed with a fall of not less than 1:100 (1 m fall for every 100 m run), in the direction of the steam flow. This slope will ensure that gravity, as well as the flow of steam, will assist in moving the condensate towards drain points where it may be safely and effectively removed (see Figure 5.21). These drain points would typically be at 30 to 50 m intervals.

Where the contours of a site make it impractical to lay the pipe with a 1:100 fall in the direction of flow, the condensate must be encouraged to run downhill and against the steam flow. Good practice is to size the pipe on a low steam velocity of not more than 15 m/s, to run the line at a slope of no less than 1:40 and to install the drain points at not more than 15 m intervals (see Figure 5.22).

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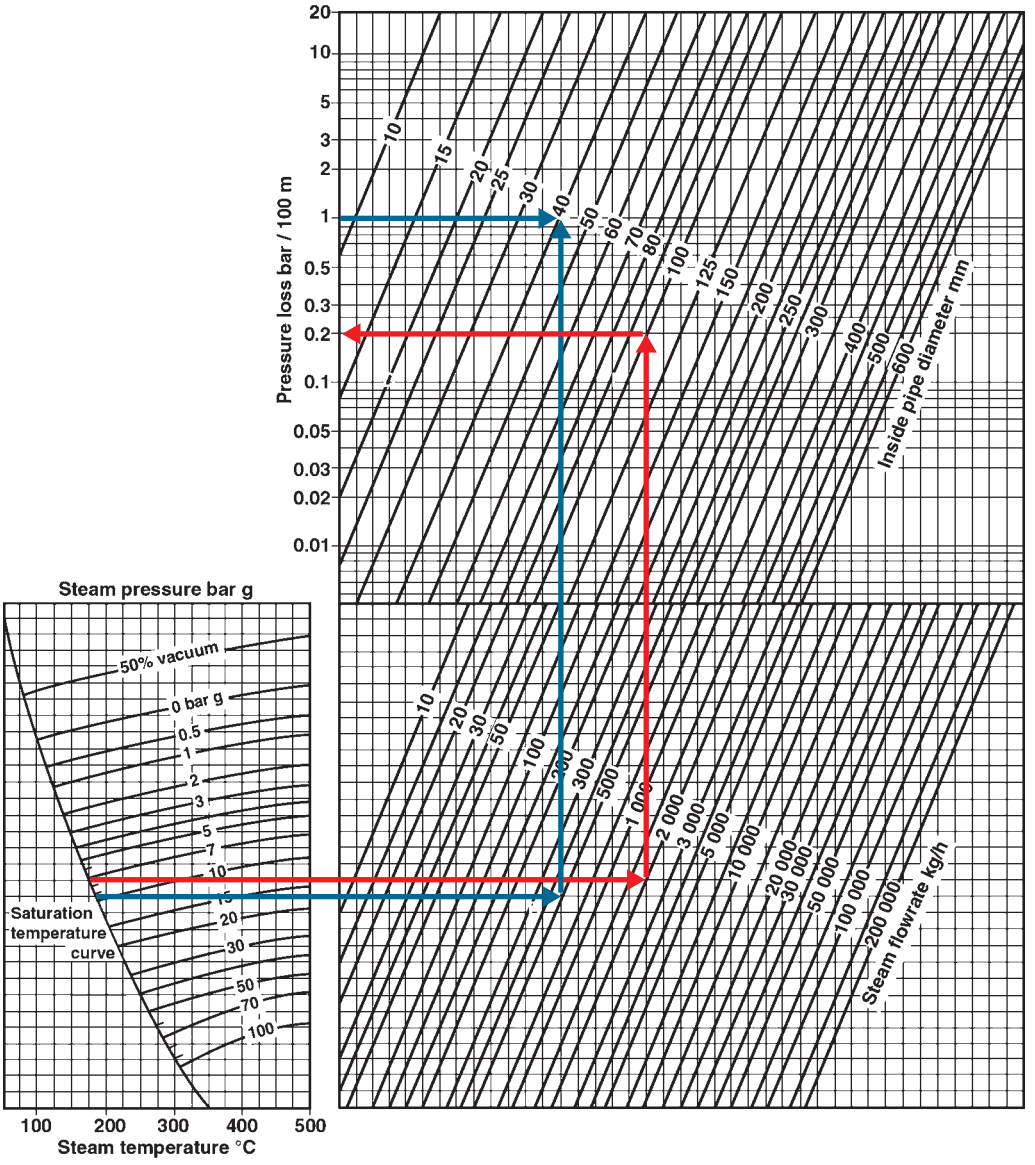



**Figure 5.20** Pipe sizing graph, using pressure drop

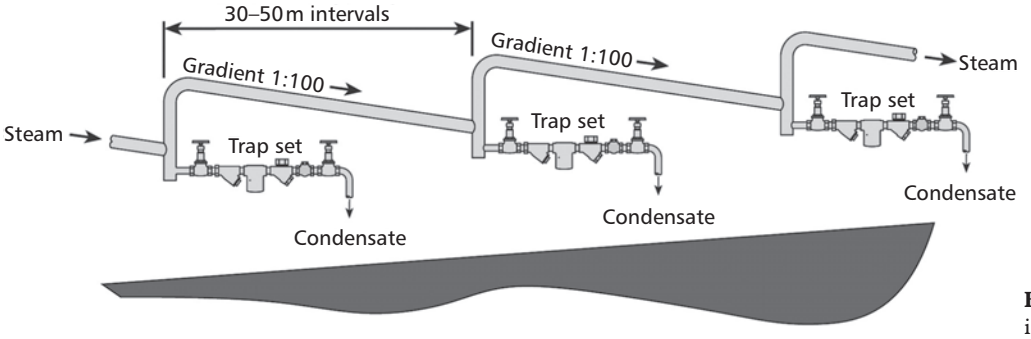



**Figure 5.21** Typical steam main installation

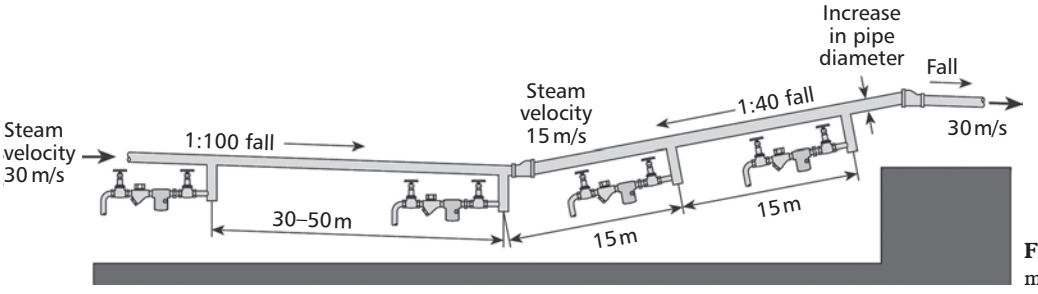



**Figure 5.22** Reverse gradient on steam main

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The objective is to prevent the condensate film on the bottom of the pipe increasing in thickness to the point where droplets can be picked up by the steam flow.

### 5.6.2 Drainage

Drain points should be fitted:

— at all low points where condensate could collect

— at intervals of between 30 and 50 m

— at terminal ends of pipe runs, where condensate will tend to be pushed

— immediately before manual or automatic valves that may be closed for periods of time.

A typical drainage point is shown in Figure 5.23.

The distance between the connection for the steam trap and the bottom of the pocket should be at least 25 to 30 mm for steam mains up to 100 mm and at least 50 mm for larger mains. This allows a space for any dirt and scale to settle. The bottom of the pocket may be fitted with a removable flange or blowdown valve for cleaning purposes. Recommended drain pocket dimensions are shown in Figure 5.24.

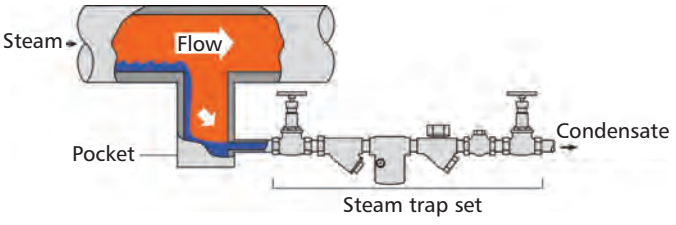



**Figure 5.23** Typical drainage point

| Mains diameter: $D$ | Pocket diameter: $d_1$ | Pocket depth: $d_2$ |
|---|---|---|
| Up to 100 mm nb | $d_1 = D$ | Minimum $d_2$ = 100 mm |
| 125–200 mm nb | $d_1$ = 100 mm | Minimum $d_2$ = 150 mm |
| 250 mm and above | $d_1 \geq D/2$ | Minimum $d_2 = D$ |

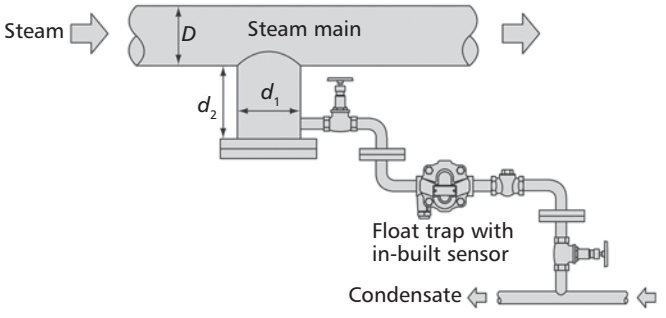



**Figure 5.24** Recommended drain pocket dimensions

### 5.6.3 Warm-up and running loads

Calculating warm-up and running loads enables the size of steam traps to be determined.

*Warm-up load*

The steam pipework needs to be brought up to operating temperature. The heat input required to do this can be determined by calculation, given the mass and specific heat of the pipework and fittings. From this, the steam condensing rate is determined:

$$\dot{m} = \frac{m\, c_p\, \Delta T}{T\, 3600\, h_{fg}} \text{ kg/s} \quad \text{or} \quad \frac{m\, c_p\, \Delta T}{T\, h_{fg}} \text{ kg/h}$$

where:

$\dot{m}$ = average steam condensing rate (kg/s)

$m$ = mass of pipe plus fittings (kg)

$c_p$ = specific heat capacity of pipeline material (kJ/kg·°C)

$\Delta T$ = temperature rise (steam temperature – ambient temperature) (°C)

$T$ = warm-up time (hours)

$h_{fg}$ = enthalpy of evaporation of steam (kJ/kg).

Alternatively, Table 5.4 may be used.

The table shows the amount of condensate generated when bringing 50 m of steam main up to working temperature – 50 m being the maximum recommended distance between trapping points.

The values shown are in kilograms. To determine the average condensing rate, the warm-up time must be considered. For example, if the warm-up process required 50 kg of steam and took 20 minutes, then the average condensing rate would be:

$$50 \times 60/20 = 150 \text{ kg/h}$$

*Running load*

Once the steam main is up to operating temperature, the rate of condensation is based on heat loss, which is a function of the ambient temperature, pipe size and the quality and thickness of the insulation.

The running load can be determined by calculation:

$$\dot{m} = \frac{U\, A\, \Delta T}{3600\, h_{fg}} \text{ kg/s} \quad \text{or} \quad \frac{U\, A\, \Delta T}{h_{fg}} \text{ kg/h}$$

where:

$\dot{m}$ = average steam condensing rate (kg/s)

$U$ = overall heat transfer coefficient (kW/m²·°C)

$A$ = area of heat transfer surface (m²)

$\Delta T$ = temperature difference (steam temperature – ambient temperature) (°C)

$h_{fg}$ = enthalpy of evaporation of steam (kJ/kg).

Alternatively, for quick approximations of running load, Table 5.5, which shows typical amounts, can be used.

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**Table 5.4** Amount of steam condensed (kg) to warm up 50 m of Schedule 40 pipe

| Steam pressure (bar g) | Steam main size (mm) | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|
| | 40 | 50 | 65 | 80 | 100 | 150 | 200 | 250 | 300 |
| 1 | 5 | 6 | 10 | 13 | 18 | 32 | 48 | 67 | 89 |
| 2 | 5 | 7 | 11 | 15 | 21 | 36 | 55 | 78 | 103 |
| 3 | 6 | 8 | 12 | 16 | 23 | 40 | 60 | 86 | 113 |
| 4 | 6 | 8 | 13 | 17 | 25 | 43 | 65 | 92 | 122 |
| 5 | 7 | 9 | 14 | 18 | 26 | 46 | 69 | 98 | 130 |
| 6 | 7 | 9 | 15 | 19 | 28 | 49 | 73 | 103 | 137 |
| 7 | 7 | 10 | 16 | 20 | 29 | 51 | 77 | 109 | 144 |
| 8 | 8 | 10 | 16 | 21 | 30 | 53 | 80 | 113 | 150 |
| 9 | 8 | 11 | 17 | 22 | 31 | 55 | 83 | 117 | 155 |
| 10 | 8 | 11 | 17 | 23 | 32 | 57 | 86 | 121 | 160 |
| 12 | 9 | 12 | 18 | 24 | 34 | 60 | 91 | 129 | 170 |
| 15 | 9 | 13 | 20 | 26 | 37 | 65 | 98 | 139 | 183 |
| 17 | 10 | 13 | 21 | 27 | 39 | 68 | 102 | 145 | 191 |
| 20 | 10 | 14 | 22 | 29 | 41 | 72 | 108 | 153 | 202 |

*Note*: figures are based on an indoor ambient temperature of 20 °C.

**Table 5.5** Condensing rate of steam (kg/h) in 50 m of Schedule 40 pipe at working temperature

| Steam pressure (bar g) | Steam main size (mm) | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|
| | 40 | 50 | 65 | 80 | 100 | 150 | 200 | 250 | 300 |
| 1 | 4 | 5 | 6 | 7 | 9 | 12 | 16 | 19 | 29 |
| 2 | 5 | 6 | 7 | 9 | 11 | 15 | 20 | 24 | 36 |
| 3 | 6 | 7 | 9 | 10 | 12 | 18 | 23 | 27 | 42 |
| 4 | 7 | 8 | 10 | 12 | 14 | 20 | 26 | 31 | 48 |
| 5 | 7 | 9 | 11 | 13 | 15 | 22 | 28 | 34 | 53 |
| 6 | 8 | 9 | 11 | 14 | 17 | 24 | 31 | 36 | 57 |
| 7 | 8 | 10 | 12 | 15 | 18 | 25 | 33 | 39 | 62 |
| 8 | 9 | 11 | 13 | 16 | 19 | 27 | 35 | 41 | 66 |
| 9 | 9 | 11 | 14 | 17 | 20 | 29 | 37 | 44 | 70 |
| 10 | 10 | 12 | 15 | 18 | 21 | 30 | 39 | 46 | 74 |
| 12 | 11 | 13 | 16 | 20 | 23 | 33 | 42 | 50 | 81 |
| 15 | 12 | 15 | 18 | 22 | 26 | 37 | 48 | 56 | 92 |
| 17 | 13 | 15 | 19 | 24 | 28 | 39 | 51 | 60 | 98 |
| 20 | 14 | 17 | 21 | 26 | 30 | 43 | 55 | 65 | 108 |

*Note*: figures are based on an indoor ambient temperature of 20 °C and an insulation efficiency of 80%.

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### 5.6.4 Pipe fittings

*Pipe size reducers*

Pipe size reducers should be of the eccentric type to allow condensate to flow freely along the pipe to the next drain point (see Figure 5.25).

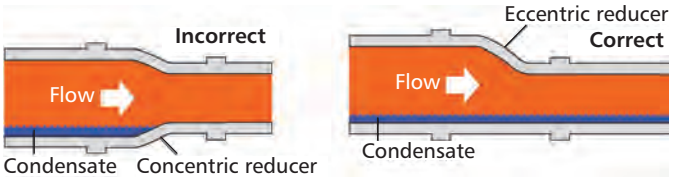



**Figure 5.25** Pipe size reducers

*Strainers*

Strainers in large diameter (above DN100) horizontal pipework should be fitted on their sides to prevent the build-up of condensate (see Figure 5.26).

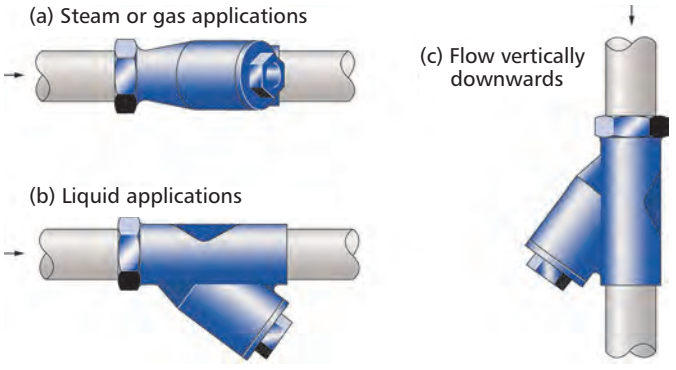



**Figure 5.26** Strainers

*Branch connections*

Branches should come from the top of mains, where the steam is driest (see Figure 5.27).

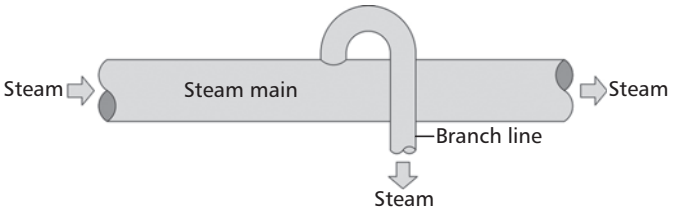



**Figure 5.27** Arrangement of branch connections

*Pipework supports*

The frequency of pipe supports will vary according to the bore of the pipe, the pipe material and whether the pipe is horizontal or vertical. It is recommended that specialist advice be sought when designing piping systems and their proper support.

*Allowance for expansion*

To ensure that no undue stresses are set up either in the pipework or the building structure, allowance must be made for any expansion caused by the system warming up. Expansion in long lengths of pipework can be taken up by a number of devices but their selection should be left to a specialist engineer, who can also advise on appropriate anchoring and guiding.

*Insulation of pipework*

Steam condenses as heat is lost by radiation from pipe and fittings to the surrounding air. This is a waste of energy as well as a cause of wet steam production. To minimise losses, pipes and fittings, such as flanges, valves and separators, should be insulated. The most economic thickness and types of insulation will depend on several factors:

— pipework temperature

— size of the pipework

— ambient air conditions, e.g. temperature, exposure to rain and wind speed.

## 5.7 Control of pressure

Due to the unique relationship between saturated steam's pressure and temperature, the temperature of the steam (which provides the driving force required to achieve heat transfer) can be controlled by varying the pressure. Pressure is easily controlled using the throttling action achieved with a two-port control valve.

### 5.7.1 Why reduce steam pressure?

The main reasons for controlling (reducing) steam pressure are:

— *to reduce steam consumption*: lower pressure steam has a higher enthalpy of evaporation, meaning that less (kg) is used for the same (kW) heat transfer

— *to reduce distribution pipe size*: high-pressure steam has a smaller volume for the same mass, hence acceptable velocities can be achieved in smaller diameter pipes – distribute at high pressure and reduce at point of use

— *to save energy*: reduced pressure means lower pipework temperatures, reducing heat loss. The amount of flash steam from drain traps is also reduced

— *to control temperature*: because steam pressure and temperature are related, control of pressure can be used to control temperature in some processes, such as hospital sterilisers and where a piece of equipment may have a design temperature limit (e.g. gaskets in a heat exchanger)

— *to save cost*: for the same heating duty, a heat exchanger designed to operate on low-pressure steam will probably be larger than one designed to be used on high-pressure steam, but the lower design specification may mean that costs are lower.

— *for safety purposes*: the safe working pressure of components in the downstream system may be lower than the steam generation or upstream pressure.

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**Table 5.6** Comparison of common pressure reducing valves

| Type | | Operation | Advantages | Disadvantages |
|---|---|---|---|---|
| Direct acting – bellows operated |  | Downstream pressure balanced through the use of a bellows, against a spring force produced by the adjustment spring. | Inexpensive<br>Small and compact<br>Easy to install<br>Fairly robust, being tolerant to imperfect steam conditions | Proportional control only<br>Wide proportional band<br>Variations in upstream pressure will affect downstream pressure |
| Direct acting – diaphragm operated |  | Downstream pressure balanced against the adjustment spring force using a rubber diaphragm. | Robust; will tolerate dirty and wet steam<br>Available in large sizes<br>Easy to set, adjust and maintain | More expensive than bellows-operated valves, but robust construction reduces lifetime cost<br>Large proportional band, leading to large changes in downstream pressure with changing load<br>Bulky |
| Pilot operated |  | Downstream pressure is sensed by a pilot valve, which then operates the main valve using a diaphragm or a piston. | Very accurate control due to narrow proportional band (typically less than 200 kPa)<br>Relatively unaffected by upstream pressure changes<br>Low hysteresis, allowing repeatable pressure control<br>Low cost compared to comparably accurate valves | More expensive than simple bellows-operated valves<br>Small internal clearances make them more susceptible to failure due to wet steam and dirt |
| Pneumatically actuated |  | Pressure is sensed electronically or pneumatically, sending a signal to a controller. This controller compares the signal to the set point and repositions the main control valve by varying the air (or water) pressure to a pneumatic actuator. | Can be linked to a proportional-integral-derivative (PID) controller to provide accurate and responsive control<br>Very accurate and flexible in terms of size and pressure range<br>Powerful enough to cope with high differential pressures<br>Set point can be remotely adjusted | Requires external power source (generally compressed air)<br>Expensive control package compared to self-acting systems described above |

### 5.7.2 Pressure control valve types

Pressure control valves (generally referred to as pressure reducing valves (PRV)) for steam will be two-port. Various types are available, characterised by the way they operate. Table 5.6 shows common types, with the advantages and disadvantages of each.

### 5.7.3 Pressure reducing valve stations

A PRV station would normally comprise the valve itself and some means of providing isolation and conditioning of the incoming steam. All pressure control valves, regardless of their type, will perform better and last longer if the steam supply is conditioned. This is achieved using a separator on the inlet to remove water droplets and a strainer to remove dirt particles. Some PRV station types are more sensitive than others to water and dirt, as Table 5.6 indicates. A PRV station will also include pressure gauges, to provide a means of setting and fault-finding. A typical PRV station is shown in Figure 5.28.

Where large changes in steam load are anticipated and where stand-by valves and large pressure reductions are required, two PRVs may be needed. These can be fitted in series or in parallel.

Valves in *parallel* (see Figure 5.29) would be used where:

— 100% stand-by is required

— the turndown ratio between the maximum and minimum flow rates is very high.

When demand is at maximum, both valves operate; when flow is reduced, the valve set at the lower pressure shuts off first, leaving the second valve to control flow. This is achieved by setting the valves at slightly different pressures. This method of operation prolongs valve life as a control valve operating with a small degree of opening over a long period can experience seat or disc erosion.

Valves would be fitted in *series* (see Figure 5.30) where there is a large turndown in pressure (for example, more than 10:1). An intermediate pressure should be selected so that

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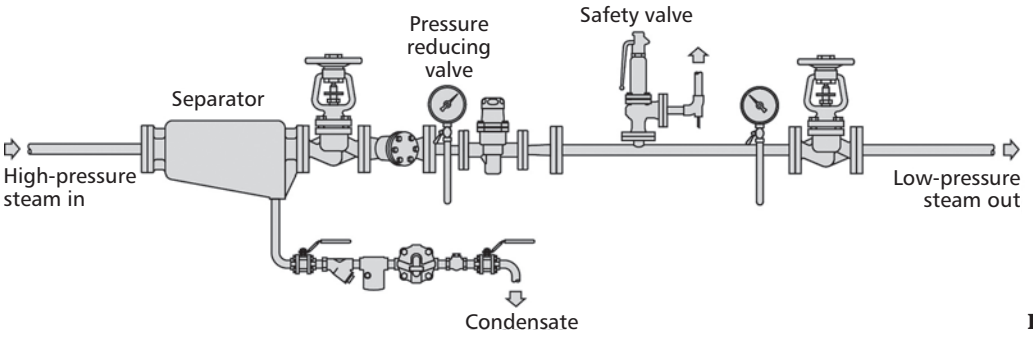



**Figure 5.28** Typical PRV station

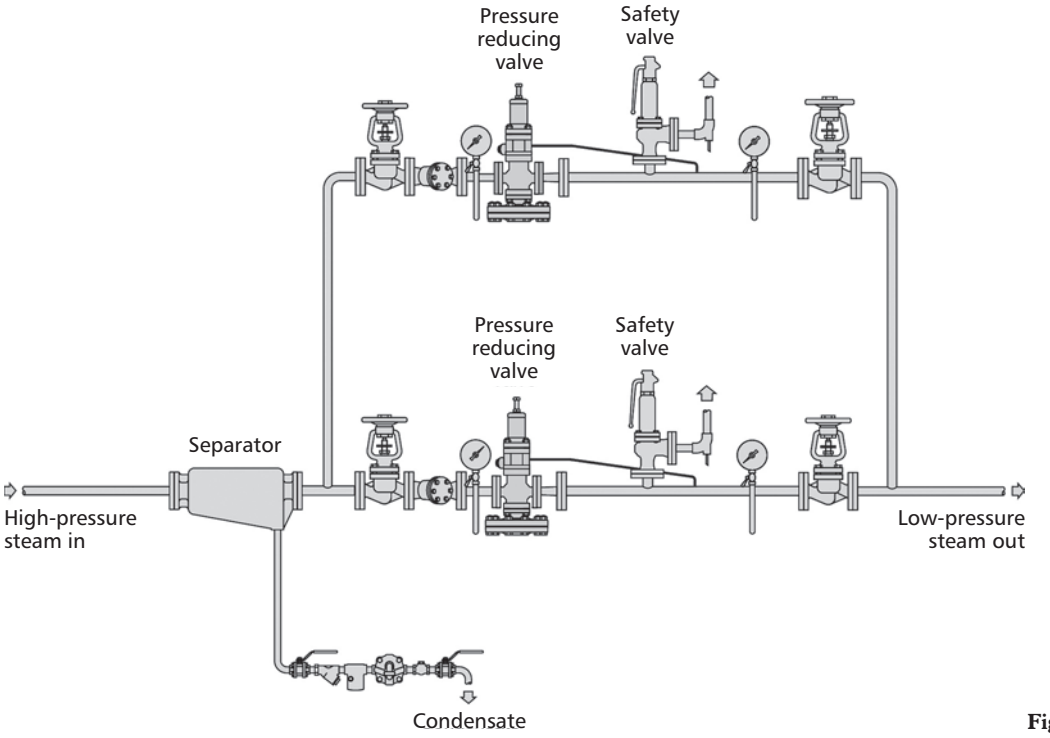



**Figure 5.29** Parallel PRV station

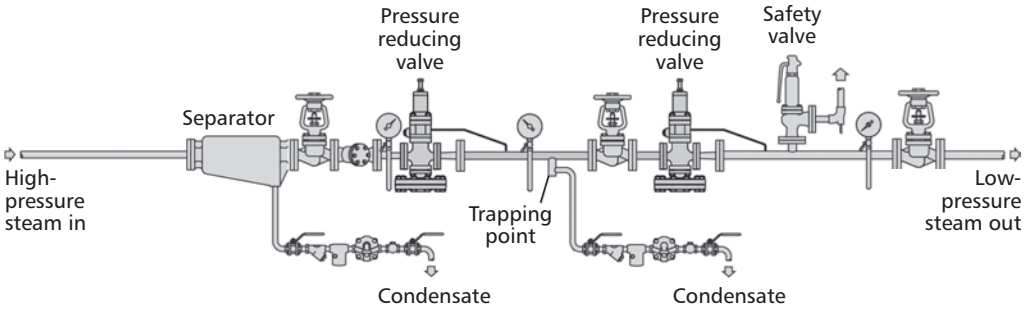



**Figure 5.30** Series PRV station

the pressure drop is shared between the two valves. Note the need for a trapping point between the two valves.

### 5.7.4 Safety valves

Valves to relieve pressure fall into two types, commonly referred to as *relief valves* and *safety valves*. Both provide the same functionality, but they operate slightly differently. A relief valve opens progressively as the pressure rises above a set point. They would commonly be used on incompressible fluid systems (e.g. liquids) where there is minimal energy storage. A compressible fluid, such as steam or gas, can contain a high degree of stored energy so it is important than any pressure excess is relieved quickly and safely. A steam safety valve therefore operates on the 'pop' principle, where it rapid lifts to the maximum degree of opening once a set pressure is reached.

Safety valves must be installed wherever the maximum allowable working pressure (MAWP) of a system or pressure-containing vessel is likely to be exceeded. In steam systems, safety valves are typically used for boiler overpressure protection and other applications, such as downstream of pressure reducing controls.

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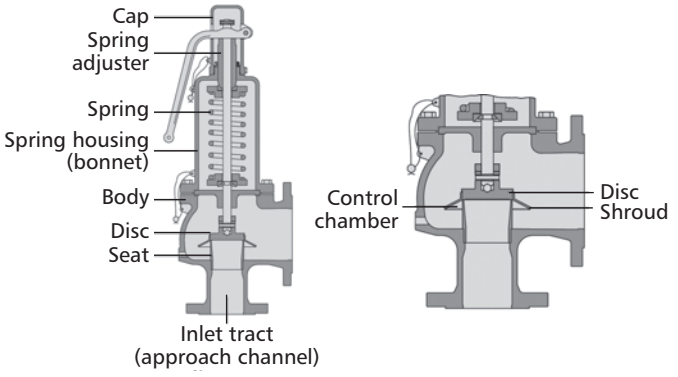



**Figure 5.31** Typical steam safety valve showing skirt or shroud

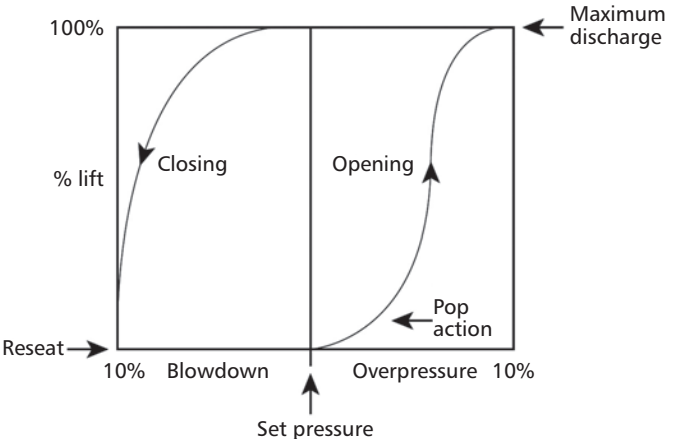



**Figure 5.32** Typical safety valve operation

An example of a safety valve used on a steam system is shown in Figure 5.31. It is the skirt or shroud arrangement around the seat which gives the rapid lifting effect as soon as the valve has started to open.

### *Operation*

Safety valves operate when the inlet static pressure rises above the safety valve set pressure. There will be an additional rise in pressure above set pressure before the safety valve will discharge its rated capacity. This additional pressure is called *overpressure* or $P_O$.

The allowable overpressure depends on the standards being followed and the particular application involved. For compressible fluids, such as steam, this is normally between 3% and 10% of set pressure.

Once normal operating conditions have been restored, the valve is required to close. The valve will not close until the pressure has dropped below the original set pressure. The difference between the set pressure and this reseating pressure is known as the *blowdown*, and it is usually specified as a percentage of the set pressure. For compressible fluids, the blowdown is usually less than 10%.

Figure 5.32 shows the operation of a typical safety valve.

### *Selection of set pressure*

A safety valve may appear to have a powerful spring with a large force holding it closed. However, this spring force is balancing the force created by the steam pressure. At just below set pressure the force holding the valve closed, and hence preventing unwanted steam escape, is very low. Many safety valves are seen to leak steam. One reason for this is that a set pressure which is too close to the system pressure has been selected.

The choice of set pressure should be made not on the basis of controlled pressure but by taking into account the pressure limitations of the product that the safety valve is protecting. The aim should be to select a set pressure which is as far away from the controlled pressure as is *safely* possible to ensure there is a reasonable spring force available to hold the valve closed.

In order to establish the set pressure correctly, the following terms require careful consideration:

— *normal working pressure* (NWP): the operating pressure of the system under full load conditions

— *maximum allowable working pressure* (MAWP): sometimes called the safe working pressure (SWP) or design pressure of the system, this is the maximum pressure existing at normal operating conditions (relative to the maximum operating temperature) of the system

— *maximum allowable accumulation pressure* (MAAP): the maximum pressure the system is allowed to reach in accordance with the specification of the system design standards – often expressed as a percentage of the MAWP.

For steam-using apparatus, the MAAP will often be 10% higher than the MAWP, but this is not always the case. If the MAWP is not readily available, the authority responsible for insuring the apparatus should be contacted. If the MAAP cannot be established, it must not be considered to be higher than the MAWP.

In terms of the valve itself, the following applies:

— The set pressure ($P_S$) is the pressure at which the safety valve starts to lift.

— The relieving pressure ($P_R$) is the pressure at which the full capacity of the safety valve is achieved – it is the sum of the set pressure ($P_S$) and the overpressure ($P_O$).

— To ensure a safe and effective set pressure, two concurrent conditions must be met:

  — the set pressure ($P_S$) must be less than or equal to the maximum allowable working pressure (MAWP), and

  — the relieving pressure ($P_R$) must be less than or equal to the maximum allowable accumulation pressure (MAAP) of the system.

Tables are available to make the selection of set pressure easier (see Table 5.7 for an example).

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**Table 5.7** Selection of safety valve set pressure

| Apparatus | | Safety valve overpressure | | | | |
|---|---|---|---|---|---|---|
| | | 5% | 10% | 15% | 20% | 25% |
| MAAP | 20% | MAWP | MAWP | MAWP | MAWP | 95% MAWP |
| | 15% | MAWP | MAWP | MAWP | 95% MAWP | 90% MAWP |
| | 10% | MAWP | MAWP | 95% MAWP | 90% MAWP | 85% MAWP |
| | 5% | MAWP | 95% MAWP | 90% MAWP | 85% MAWP | 80% MAWP |

**Example**

*A pressure reducing valve reduces steam pressure from 12 bar g to 5 bar g to serve a heat exchanger. The heat exchanger nameplate shows an NWP of 5 bar g, MAWP of 10 bar g and MAAP of 11 bar g. As the upstream steam pressure is above the limits of the heat exchanger, a safety valve is needed. One with a 10% overpressure is to be used. What should the set pressure be?*

The MAAP is 10% above MAWP. According to Table 5.7, the safety valve should be set at the MAWP, which is 10 bar g.

With a controlled pressure of 5 bar g, there is good margin to prevent nuisance leakage from the safety valve.

In some instances, a safety valve setting can be influenced by temperature limitations in the plant or process being protected. For example, a plate heat exchanger may have a safe working pressure (MAWP) of 10 bar g but the safety valve may need to be set at a pressure equivalent to 170 °C to protect gasket material.

### *Safety valve capacity*

A safety valve must be sized such that it can safely discharge the source of steam without causing the protected apparatus to exceed its maximum pressure limit.

The safety valve discharge capacity must be calculated on the basis of a 'fault load' – that is, an assumption that pressure control valves upstream fail in the fully open position. This could mean that the valve's required venting capacity is well above the normal running load.

It is also important that all sources of steam supply are taken into account. An example is a parallel PRV station or a PRV with a bypass. Should steam be able to flow through these parallel paths, the safety valve must be able to safely discharge this additional quantity. The fault load is calculated using the same formula applied in sizing a standard steam control valve (see Section 5.8):

$$\dot{m}_s = 12\, K_v\, P_1 \sqrt{1 - 5.67(0.42 - x)^2}$$

where:

$x$ = pressure drop ratio = $\dfrac{P_1 - P_2}{P_1}$

$\dot{m}_s$ = mass flow rate (kg/h)

$P_1$ = upstream pressure (bar a)

$P_2$ = downstream pressure (bar a)

$K_v$ = valve flow coefficient (actual)

but here $K_v$ is the $K_{vs}$ value for the upstream valve or valves. $K_{vs}$ is the maximum capacity with the valve fully open and would be the figure published in the manufacturer's specification. The downstream pressure ($P_2$) would be the safety valve set pressure, not the normal control pressure, as this is what the pressure would be if the safety valve opened.

Along with fault load, the other sizing factor for a safety valve is set pressure. Obviously, the greater the set pressure, the greater the valve's capacity. Knowing fault load and set pressure, standard manufacturers' tables can be used to determine the correct valve size.

### *Safety valve installation*

The correct installation of a safety valve is essential to ensure its safe operation. Always consult the manufacturer's instructions. The examples quoted below are for general guidance only.

#### *Dirt*

A safety valve with a damaged seat will leak steam. Seat damage is commonly caused by dirt and debris left in the pipework after installation. The pipework should be flushed or blown through to ensure that it is completely free of any foreign matter that may pass to the valve seat and cause damage, leading to seat leakage. Blow through must be carried out before installing the safety valve, which should be mounted where dirt, scale and debris cannot collect.

#### *Condensate build-up*

Leakage can also occur through a build-up of condensate on the upstream side of the valve seat. This can be prevented by installing the safety valve vertically upwards above the steam pipe, as shown in Figure 5.33.

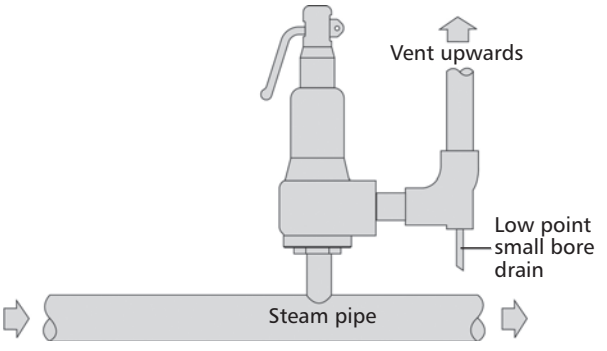



**Figure 5.33** Correct installation of a safety valve on a steam system

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*Inlet pipework*

Pressure drop in the inlet pipework must be minimised by keeping the pipework length as short as possible. The valve should be fitted at least 8 to 10 pipe diameters downstream from any converging or diverging 'Y' fitting or any bend. There should be no intervening valve or fitting (i.e. it should not be possible to isolate the safety valve). The inlet pipe connection should not be smaller than the valve.

Excessive pressure loss can lead to 'chatter', which may result in reduced capacity and damage to the seating faces and other parts of the valve. BS ISO 4126-9 recommends that the pressure drop be kept below 3% of the set pressure when discharging.

Any corners or bends in the inlet pipework must be suitably rounded. The standard BS ISO 4126-9 recommends that corners should have a radius of not less than one-quarter of the bore.

*Outlet pipework*

There are two possible types of discharge system – open and closed. An open system discharges directly into the atmosphere whereas a closed system discharges into a manifold along with other safety valves.

Discharge pipework for steam systems should ideally rise. Horizontal pipework should have a downward gradient of at least 1 in 100 away from the valve to ensure that any discharge will be self-draining. It is important to drain any rising discharge pipework so that downstream flooding (which can also encourage corrosion and leakage) cannot occur. Vertical rises will require separate drainage.

Many safety valves are provided with a body drain connection to ensure condensate removal. If this is not used or not provided, then a small bore drain should be fitted in close proximity to the valve outlet (see Figure 5.33).

Too high a back pressure on a safety valve outlet will affect the valve's performance.

BS EN ISO 4126-1 states that the pressure drop should be maintained below 10% of the set pressure. In order to achieve this, the discharge pipe can be sized using the equation:

$$d = \sqrt[5]{\frac{L_e \dot{m}^2 \frac{(V_g + 2)}{2}}{0.08P}}$$

where:

$d$ = pipe diameter (mm)

$L_e$ = equivalent length of pipe (m)

$\dot{m}$ = discharge capacity (kg/h)

$P$ = safety valve set pressure (bar g) × required % pressure drop

$V_g$ = specific volume of saturated steam at the set pressure ($P$) (m$^3$/kg).

The pressure ($P$) should be taken as the maximum allowable pressure drop according to the relevant standard. In the case of BS EN ISO 4126-1, this would be 10% of the set pressure and it is at this pressure that $V_g$ is taken.

A short discharge pipe and one with few bends and other restrictions will minimise any pressure drop. To make the calculation easier, many manufacturers offer sizing software for use in establishing whether a selected discharge pipe is large enough. The diameter of any outlet pipe should never be smaller than the safety valve outlet connection.

The outlet pipework should be directed to a safe point of discharge where there is no risk of injury to persons or damage to property in the event of the valve operating.

The outlet pipework should be adequately supported to ensure that no undue stress is placed on the safety valve, which could cause the valve to fail to operate properly or the set pressure to be affected.

## 5.8 Sizing control valves for steam

Whatever the purpose of the valve – controlling pressure, temperature, flow, humidity, etc. – the process of sizing it is the same.

Whereas two- and three-port valves are required for liquid distribution systems, steam control valves are only two-port.

Although it is true in most cases that a control valve will pass fluid at a rate corresponding to the differential pressure across it, flow through a steam valve is restricted due to a condition known as critical pressure drop. Increasing the differential pressure across a fully open valve increases the flow rate. This results in an increase in velocity through the valve seat. Bernoulli's principle is that increasing a fluid's speed results in a drop in pressure. A drop in steam pressure results in an increase in specific volume, which in turn means an even greater increase in velocity as more steam tries to pass through the valve. This drop in pressure and corresponding volume increase reaches the point at which the valve becomes 'choked' and increasing the differential pressure across the valve will not result in any mass flow increase. The pressure drop at which this choking occurs is known as *critical pressure drop* and, for saturated steam, occurs when the downstream pressure is 58% of the upstream absolute pressure.

Control valve manufacturers will be able to provide software to make the sizing process easier. Alternatively, formulae can be used or reference made to sizing charts.

The formula method or the sizing charts will enable a valve $K_v$ value to be obtained. $K_v$ represents a valve's capacity index and is expressed in two different ways:

— $K_{vr}$ is the required valve capacity, and would be the value obtained from the formula or sizing chart

— $K_{vs}$ is a measure of a valve's maximum capacity, and would be the figure quoted by the manufacturer.

Various formulae are used to determine valve size. One which gives good results is:

$$\dot{m}_s = 12\, K_v\, P_1 \sqrt{1 - 5.67(0.42 - x)^2}$$

where:

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$x$ = pressure drop ratio = $\dfrac{P_1 - P_2}{P_1}$

$\dot{m}_s$ = mass flow rate (kg/h)

$P_1$ = upstream pressure (bar a)

$P_2$ = downstream pressure (bar a)

$K_v$ = valve flow coefficient (actual).

If $P_2$ is less than the critical pressure, then the term within the bracket $(0.42 - x)$ becomes negative. This term is then taken as zero and the function within the square root sign becomes unity. The equation is simplified as:

$$\dot{m}_s = 12\ K_v\ P_1$$

A typical steam $K_v$ sizing chart is shown in Figure 5.34.

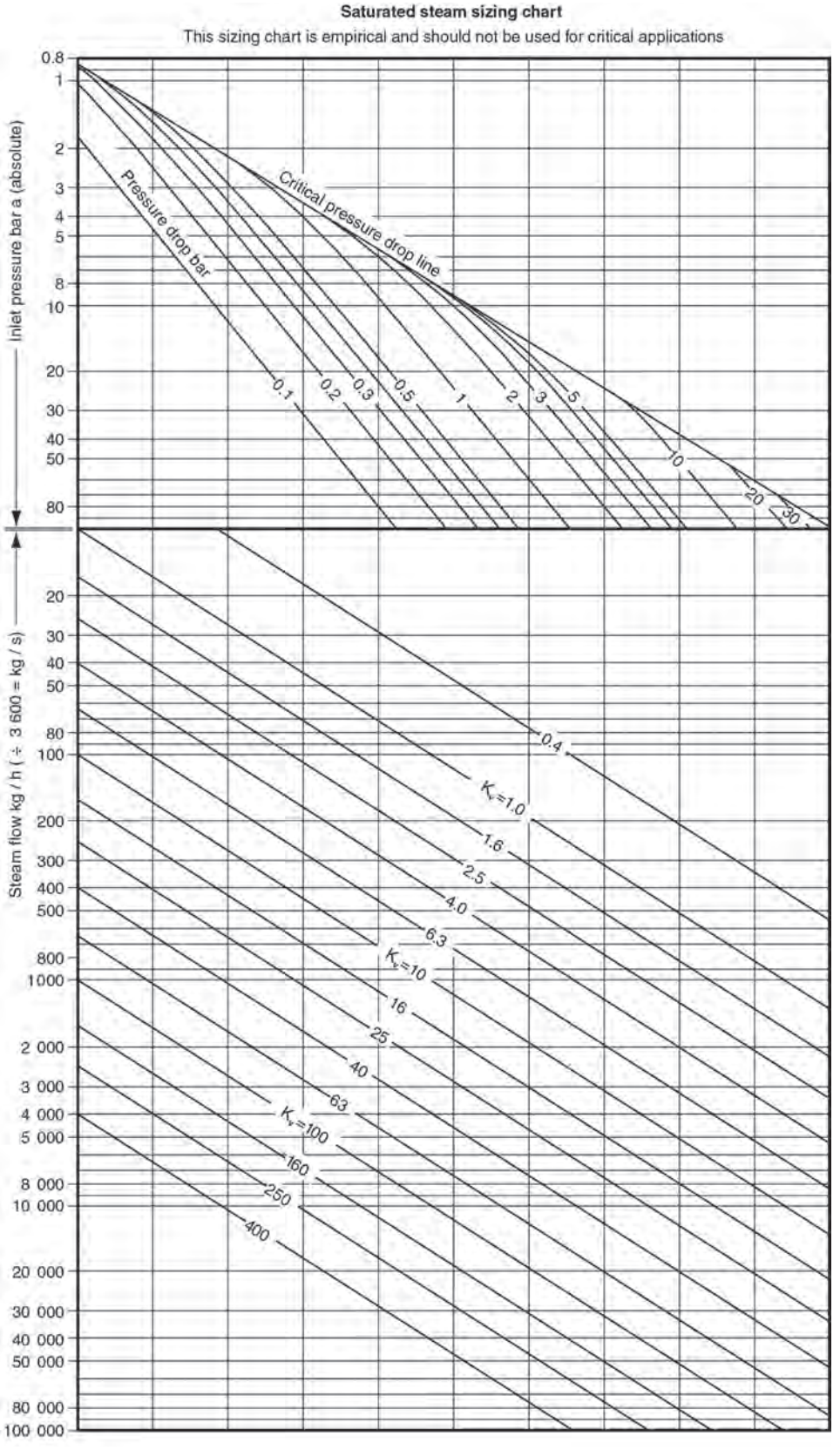



**Figure 5.34** Saturated steam valve sizing chart

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**Example**

*Determine the $K_{vr}$ of a valve to control the flow of 500 kg/h of saturated steam at an upstream pressure of 10 bar g and a pressure drop of 2 bar across the valve.*

**Using the formula**

The pressure drop ratio is:

$$x = \frac{11-9}{11} = 0.18$$

This is less than 0.42 so the steam flow must be 'sub-critical'.

The formula used is:

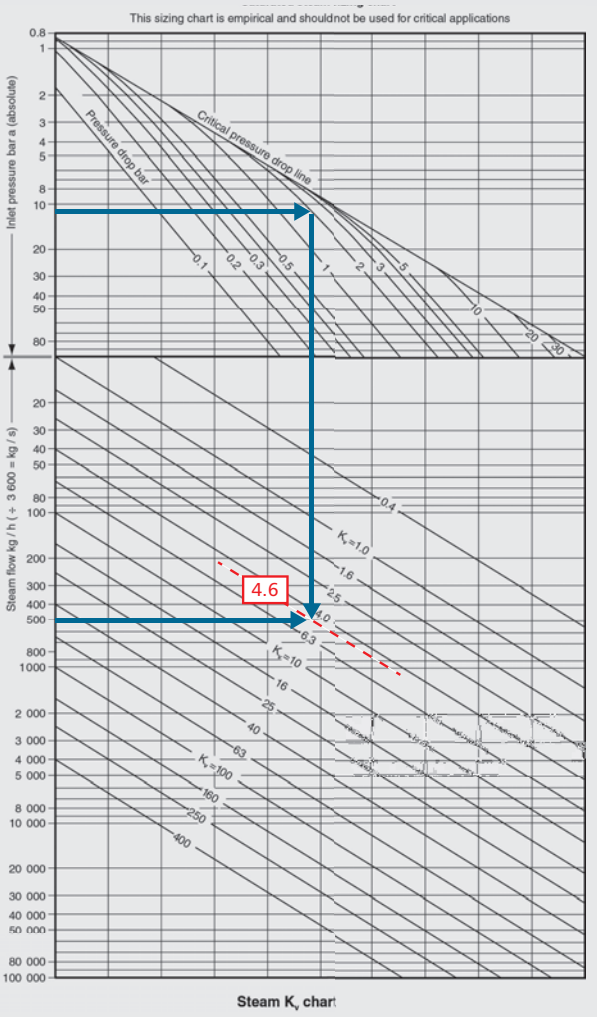
$$\dot{m}_s = 12\, K_v\, P_1 \sqrt{1 - 5.67(0.42 - x)^2}$$

Therefore:

$$K_{vr} = \frac{500}{12 \times 11\sqrt{1 - 5.67(0.42 - 0.18)^2}} = \frac{500}{108}$$
$$= 4.63$$

**Using the chart**

It is common to see valve $K_{vs}$ values following a recognised series. The lines on the chart in Figure 5.34 indicate this series. For this particular example, a valve with a $K_{vs}$ value of 6.3 would be chosen, being the next largest size available.




Formulae and sizing charts provide quick and easy ways of determining required valve sizes. However, the information they provide is limited. For applications where multiple valves are to be sized, where sizing is critical or where additional information, such as noise prediction, is required, then manufacturers' valve sizing software should be used.

## 5.9 Condensate and air removal

When steam condenses, condensate is formed. If this is allowed to build up, the heat transfer process is adversely affected and safety-related issues, such as heat exchanger failure and waterhammer damage, can result. Condensate removal is achieved through the use of steam traps, pumps or combined pump/trap systems.

Air is a very effective insulator and therefore a barrier to heat transfer. It also contains gases that can contribute to corrosion in steam and condensate systems. A steam system that is isolated for any period will fill with air, which must be removed at start-up. But air can also be introduced with the steam. Proper conditioning of the boiler feedwater can minimise the introduction of air in this way.

Condensate and air should be dealt with separately, but many types of steam trapping devices also have the capacity to remove air.

### 5.9.1 Why use steam traps?

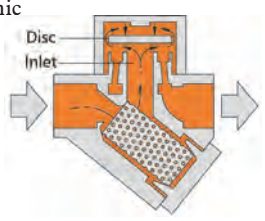
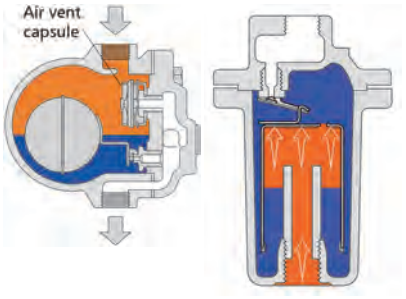
A steam trap's function is to 'trap' steam but release condensate and incondensable gases at the appropriate time. Without a steam trap, a heat exchanger would waste significant amounts of energy and temperature control would be a difficult process.

A steam trap is a passive device and will only allow condensate to flow through it if there is a positive pressure differential across it. A negative differential can result from back pressure created downstream of the trap, generally caused by the steam pressure being used to lift condensate into a high level return pipe, or by a reduction in pressure in the heat exchanger. It is extremely common to see heat exchange systems in building services fitted with a temperature control valve on the inlet. Sub-atmospheric conditions can therefore be created in the heat exchanger at light load periods, making condensate removal difficult.

When such conditions occur, using a steam trap on its own will not allow condensate to be removed. A pump (generally steam-operated) would be used. In such applications, the pressure in the heat exchanger will rise at higher loads and, under these circumstances, a steam trap will be needed to prevent live steam loss. To cater for both light and normal loads, a combination of a mechanical pump and a steam trap is therefore needed, either as two separate components in series or as a combined pump/trap product.

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**Table 5.8** Steam traps for building services applications

| Steam trap type | Features | Principle of operation |
|---|---|---|
| Thermodynamic   | Small, rugged and able to operate over a wide pressure range<br>Discharges condensate as soon as it forms | Hot condensate passing under the disc flashes off to steam. The increased velocity of this flash steam creates a low pressure, drawing the disc onto its seat.<br>A drop in pressure caused by the flash steam condensing above the disc causes the trap to open. |
| Mechanical   | Continuous discharge, automatically adjusting to pressure and flow rate changes<br>Large capacity, discharging condensate as soon as it forms<br>Can be damaged if frozen | Operates on the difference in density between water and steam. A float will rise in water, so opening the valve to allow the water to drain out.<br>An inverted bucket operates in the reverse way, with steam inside it causing it to float, and so close the valve.<br>An internal integral air vent, or a separate bypass, is used to remove air.<br>Float types are more common in building services applications. With the inclusion of an air vent, they are commonly referred to as float thermostatic traps. |
| Thermostatic  | Holds back condensate until it has cooled, so making use of some sensible heat and reducing flash steam formation<br>Suitable for applications where waterlogging can be tolerated.<br>Excellent air handling ability | Can be balanced pressure or bimetallic.<br>Balanced pressure is commonly used in building services applications. An internal capsule contains a liquid that vaporises as the surrounding condensate approaches steam temperature. This vaporisation causes the valve to close.<br>Condensate is then held back until it sub-cools. The vapour in the capsule condenses and the valve opens. By varying the filling, the operating temperature can be changed. |

### 5.9.2 Types of steam trap

A steam trap must be suitable for the maximum steam pressure within the space and should have sufficient capacity to discharge condensate at the correct rate, depending on the differential pressure across it.

The steam trap may have to discharge condensate at steam temperature (i.e. as soon as it forms in the steam space) or it may be required to discharge below steam temperature, giving up some of its 'sensible heat' in the process. The type of trap selected will determine the discharge temperature.

Three types of steam trap that are most widely used in heating, ventilating and air conditioning (HVAC) applications are detailed in Table 5.8.

### 5.9.3 Steam trap selection and sizing

The choice of type of steam trap should be made taking into account the characteristics of the steam plant to which it is connected. Table 5.9 shows the recommended traps for some common HVAC applications.

Steam traps are sized on the quantity of condensate to be removed and the differential pressure (trap inlet pressure less outlet pressure) across the trap. Account should be taken of any back pressure on the trap caused, for example, by any rise in the pipework. The differential pressure is commonly taken as the steam pressure at the plant inlet.

Traps must be sized for normal running and to cope with start-up conditions. During start-up, the steam condensation rate will be at its highest, while the pressure at the trap inlet will initially be very low. As the system comes up to temperature, the condensate flow rate decreases while the pressure increases.

To ensure that traps are suitable for the start-up conditions, they are sized on the normal running differential pressure and the normal running condensate flow rate multiplied by a factor, typically 2 or 3 unless the specific start-up load can be calculated (see Section 5.10.2).

**Table 5.9** Steam trap selection guide

| Application | First choice | Second choice |
|---|---|---|
| Steam mains | | |
| Horizontal runs | Thermodynamic | Float thermostatic |
| Terminal ends | Thermodynamic | Float thermostatic |
| Shut-down drain | Balanced pressure thermostatic | |
| Boiler header | Float thermostatic | Inverted bucket |
| Separator | Float thermostatic | Inverted bucket |
| Space heating equipment (heat exchangers, unit heaters, air heater batteries) | Float thermostatic | Inverted bucket |
| Canteen equipment | | |
| Boiling pans | Float thermostatic | Thermodynamic |
| Steaming ovens | Balanced pressure thermostatic | |
| Laundry equipment | | |
| Garment presses | Thermodynamic | Float thermostatic |
| Ironers and calenders | Float thermostatic | Inverted bucket |
| Tumble dryers | Float thermostatic | Inverted bucket |
| Autoclaves and sterilisers | Balanced pressure thermostatic | Float thermostatic |

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**Example**

*Select a steam trap type for a hot water storage calorifier requiring 300 kg/h of steam at 4.0 bar g. There will be a lift of 5 m in the condensate pipe after the trap.*

According to Table 5.9, a float thermostatic type steam trap is first choice for this application.

Using a factor of 3 on normal running load, the trap should be sized to pass 900 kg/h of condensate. The back pressure on the trap will be 0.5 bar. The differential pressure is therefore 4.0 – 0.5 = 3.5 bar.

A float thermostatic trap able to pass 900 kg/h with a differential pressure of 3.5 bar would be selected.

### 5.9.4 Steam trap installation

A typical steam trap station (see Figure 5.35) should include a means of isolating the trap upstream (and downstream if there is any possibility of back pressure) for maintenance purposes. A pipeline strainer (if not part of the trap itself) should be included to protect the trap from dirt. A check valve is required where back flow is a possibility. All of these items will be the same size as the trap connections. Note that the sizes may differ from the connection on the steam-using equipment.

Steam trap stations are now available as a single module with isolation valves, strainer trap and check valve included in a single housing, so reducing installation time and minimising the potential for leaking joints. Some models also incorporate a trap-checking device (see Figure 5.36).

Checking steam traps for correct operation is important. A trap that has failed open and hence is leaking steam is a costly item. A trap that has failed closed presents a different problem, with ineffective condensate removal resulting in reduced heat output and possible waterhammer damage.

An alternative to inbuilt testing is to conduct regular audits using experienced technicians. These audits can be supported by software programs, which are used to help manage the monitoring process (see Section 9.2).

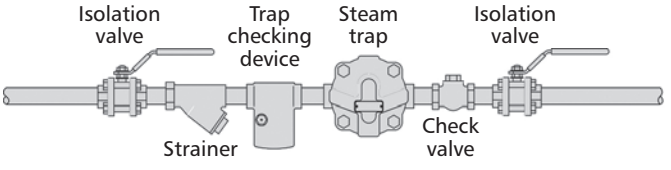



**Figure 5.35** Typical steam trap station

### 5.9.5 Air removal

The most appropriate place to remove air is before it gets into the steam boiler. Conditioning the feedwater using chemicals such as sodium sulphite or by raising the temperature of the water will minimise the quantity of air and other dissolved gasses that could enter the boiler with the feedwater. Atmospheric and pressurised deaerators are specialist types of storage vessels designed to remove air from stored feedwater.

At start-up, pipework and connected equipment will be full of air. Incoming steam will push the air within the pipework and plant to the furthest points. This air should be removed at these points as it can increase the time taken for the steam-using equipment to reach operating temperature, and can lead to cold spots, reduce heat output from plant and create corrosion. Automatic air vents, strategically located where air will be driven by the steam or will naturally collect, will remove any air and gases reaching them. These are automatic devices, operating on the same principle as a balanced pressure steam trap, releasing cooler air but holding back steam. The discharge from these air vents should be piped to a safe place.

Small amounts of air and other non-condensable gases may also be mixed with the steam during normal operation. This will reduce the steam temperature and will also be left behind when the steam condenses. Air vents will aid the removal of this steam/air mixture. Some types of steam trap, typically float thermostatic and balanced pressure, have the ability to discharge any air reaching them (see Figure 5.37).

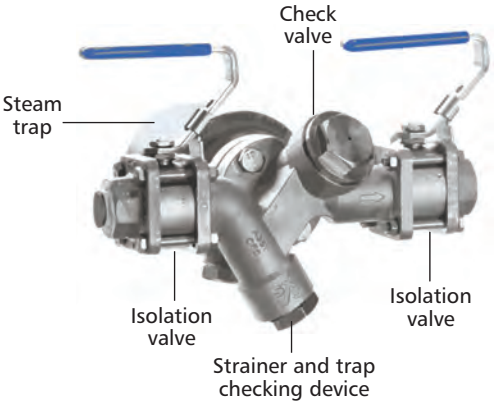



**Figure 5.36** Alternative integrated steam trap station

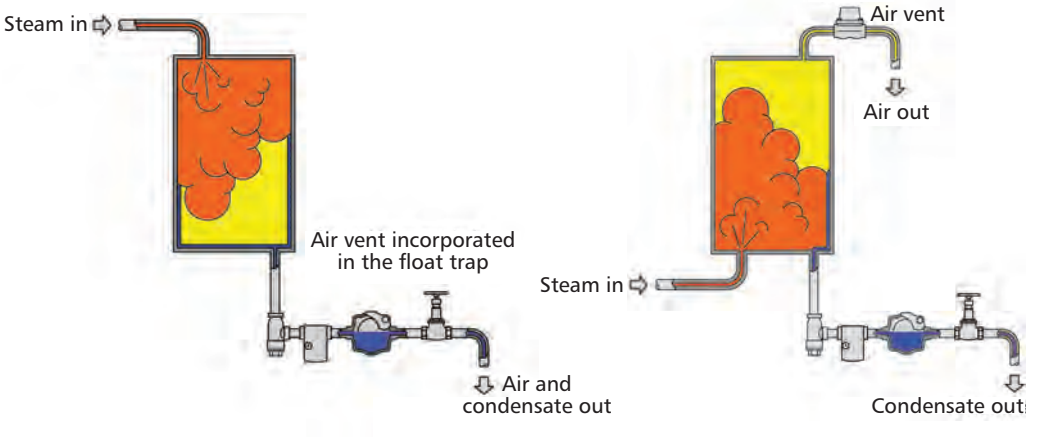



**Figure 5.37** Location of air vent

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## 5.10 Condensate handling

Condensate is a valuable commodity. See Section 8.2 for a full description of the benefits of returning as much condensate as possible. Condensate is returned using gravity or pumped systems or, commonly, a combination of both. The aim of all return systems is to return condensate to the boilerhouse with minimum losses so that it can be used to produce more steam.

### 5.10.1 Gravity and pumped systems

In a gravity return system, condensate falls from a high point to a lower one under the influence of gravity alone. There are few 100% gravity return systems. Steam pressure is often used to 'lift' condensate to a higher point. This lift prevents true gravity flow as well as creating additional back pressure on the steam trap, reducing trap capacity and increasing the likelihood of heat exchangers flooding with condensate, with a resultant reduction in output and potentially causing mechanical damage.

After the condensate has been lifted to a high point, gravity can be used to allow it to flow into the receiver of a local condensate pump set from which it can be pumped back to the boilerhouse. The slope of the pipe will determine whether gravity alone causes the condensate to flow at the required rate. A minimum slope of 1:70 (150 mm in 10 m) is recommended.

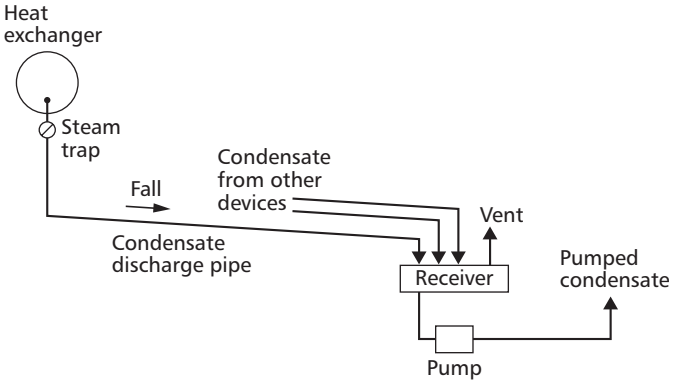



**Figure 5.38** Gravity and pumped condensate

Pipework connecting the trap outlet to the pump receiver will contain a mixture of condensate and flash steam. The amount of flash steam in the pipework will depend on the temperature of the condensate leaving the trap and the back pressure in the return system. Although the mass flow rate of flash steam may be small in comparison with the flow rate of condensate, the volume occupied by the flash steam is significant and this flash steam has a major impact on how the condensate return line should be sized. See the following example and Sections 1.2.4 and 8.4 for further information on flash steam generation and recovery.

Flash (or even 'live') steam in the return pipework can escape from the vent pipe on the pump receiver. This is a waste of energy and water so flash steam recovery systems should be used to capture and reuse this steam (see Section 8.4).

A pumped line will contain water only and should be sized on velocities more appropriate to water circulation systems. The pump used can be either of mechanical or electrical operation. Table 5.10 compares the advantages of each system.

**Example**

*Condensate discharges at steam temperature from a system operating at 10 bar g. The pressure in the return line is atmospheric (0 bar g). Determine the relative volume occupied by steam and water in the return pipework.*

$$\text{flash steam formed} = \frac{h_f1 - h_f2}{h_{fg}2}$$

where:

$h_f1$ = enthalpy of condensate at 10 bar g = 782 kJ/kg

$h_f2$ = enthalpy of condensate at 0 bar g = 419 kJ/kg

$h_{fg}2$ = enthalpy of evaporation at 0 bar g = 2257 kJ/kg

(see Section 1.2.4 for full explanation)

$$\% \text{ flash} = \frac{782 - 419}{2257} = 16\%$$

The return line therefore contains 84% water and 16% steam *by mass*.

**By volume**

Based on 1 kg of condensate being discharged.

Density of water at 0 bar g: 958 kg/m³

Volume occupied by 0.84 kg water:

$$\frac{0.84}{958} = 0.00088 \text{ m}^3$$

Specific volume of steam at 0 bar g: 1.673 m³/kg

Volume occupied by 0.16 kg steam:

$$0.16 \times 1.673 = 0.268 \text{ m}^3$$

Total volume:

$$0.268 + 0.00088 = 0.26888 \text{ m}^3$$

**By proportion**

Water occupies:

$$\frac{0.00088}{0.26888} = 0.3\%$$

Steam occupies:

$$\frac{0.268}{0.26888} = 99.7\%$$

The mixture in the return pipework is 99.7% steam and 0.3% water. It makes sense to size the pipework based on steam flow principles rather than water flow.

### 5.10.2 Sizing condensate return lines

The nature of the return line contents influences how the line should be sized. There are three main types of condensate return lines:

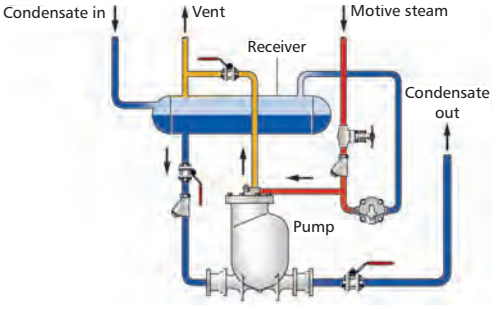
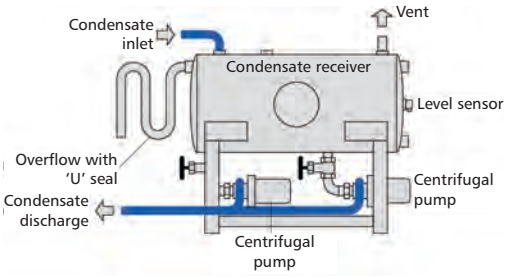
— *drain lines*, connecting the process to the steam trap (no flash steam present)

— *trap discharge lines*, connecting the steam trap to a gravity return system or local pump receiver (flash steam present here)

— *pump discharge lines* (fully flooded lines containing water only).

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**Table 5.10** Mechanical vs electrical condensate pump comparison

| Mechanical | Electrical |
|---|---|
|  |  |
| Handle high temperatures with no cavitation | Handle high loads |
| Ideal for hazardous, dirty and wet environments | Easier to monitor and install |
| Operate under vacuum conditions | Commonly accepted technology |
| Compact installation due to smaller receiver | Potentially smaller condensate return pipes |
| No issues with number of starts/stops | Able to pump against higher back pressures |
| No risk of high temperatures damaging seals | |

## *Drain lines*

The condensate flow rate in the equipment drain line has to be established before charts or tables that determine pipe size can be used. As well as handling condensate, the drain line will also discharge air, mainly under start-up conditions. The drain line will commonly be the same size as the steam trap fitted at the end of it. Steam traps are sized taking into account start-up and running loads. The condensate load can vary significantly (sometimes up to a factor of 3) between start-up and full running load conditions. Condensate formed under running conditions would be determined from the calculation of steam load (see Section 5.1). Start-up loads are more difficult to determine. The following can be used as guidance:

— *For steam pipe drainage applications*: assuming the pipe is drained at the recommended intervals (no more than 50 m) and has good insulation fitted, a reasonable estimate of condensate load is 1% of the steam flow rate. Doubling this for start-up conditions would be the basis for determining the drain line size.

— *For constant pressure processes*: examples of such processes are radiant panels and unit heaters where no temperature control is fitted, meaning that the application always runs at a constant pressure. For these processes, drain lines should be sized on twice the running load.

— *For temperature controlled applications*: most steam plant is temperature controlled, examples being air heater batteries, and storage and non-storage heat exchangers. The reductions in steam pressure caused by the temperature control valve require larger steam traps to be fitted. The size of the trap, and hence the drain line, would be based on three times the running load.

Apart from flow rate, the pipe size chosen will also depend on pressure drop and/or velocity. For drain lines to steam traps, because they contain water only, water sizing charts such as Table 5.11 can be used. A pressure drop of 100 Pa/m and a velocity of 1.5 m/s are recommended maximum values.

**Table 5.11** Pipe capacity/pressure drop chart for water

| Pressure drop | | Pipe size ∅ | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|
| | | 15 mm | 20 mm | 25 mm | 32 mm | 40 mm | 50 mm | 65 mm | 80 mm | 100 mm |
| (Pa/m) | (mbar/m) | Capacity (kg/h) | | | | | | | | |
| 90.0 | 0.900 | 173 | 403 | 745 | 1627 | 2488 | 4716 | 9612 | 14 940 | 30 240 |
| 92.5 | 0.925 | 176 | 407 | 756 | 1652 | 2524 | 4788 | 9756 | 15 156 | 30 672 |
| 95.0 | 0.950 | 176 | 414 | 767 | 1678 | 2560 | 4860 | 9900 | 15 372 | 31 104 |
| 97.5 | 0.975 | 180 | 421 | 778 | 1699 | 2596 | 4932 | 10 044 | 15 552 | 31 500 |
| 100.0 | 1.000 | 184 | 425 | 788 | 1724 | 2632 | 5004 | 10 152 | 15 768 | 31 932 |
| 120.0 | 1.200 | 202 | 472 | 871 | 1897 | 2898 | 5508 | 11 196 | 17 352 | 35 100 |
| 140.0 | 1.400 | 220 | 511 | 943 | 2059 | 3143 | 5976 | 12 132 | 18 792 | 38 160 |
| 160.0 | 1.600 | 234 | 547 | 1015 | 2210 | 3373 | 6408 | 12 996 | 20 160 | 40 680 |
| 180.0 | 1.800 | 252 | 583 | 1080 | 2354 | 3589 | 6804 | 13 824 | 21 420 | 43 200 |
| 200.0 | 2.000 | 266 | 619 | 1141 | 2488 | 3780 | 7200 | 14 580 | 22 644 | 45 720 |
| 220.0 | 2.200 | 281 | 652 | 1202 | 2617 | 3996 | 7560 | 15 336 | 23 760 | 47 880 |
| 240.0 | 2.400 | 288 | 680 | 1256 | 2740 | 4176 | 7920 | 16 056 | 24 876 | 50 400 |
| 260.0 | 2.600 | 306 | 713 | 1310 | 2855 | 4356 | 8244 | 16 740 | 25 920 | 52 200 |
| 280.0 | 2.800 | 317 | 742 | 1364 | 2970 | 4536 | 8568 | 17 388 | 26 928 | 54 360 |
| 300.0 | 3.000 | 331 | 767 | 1415 | 3078 | 4680 | 8892 | 18 000 | 27 900 | 56 160 |

Velocity: 0.15 m/s; 0.3 m/s; 0.5 m/s; 1.0 m/s; 1.5 m/s

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### *Trap discharge lines*

The discharge line from the steam trap will contain a mixture of condensate and flash steam. The flash steam will occupy the majority of the pipe so trap discharge lines should be sized based on flow rates of steam rather than water. The steam here will contain a lot of water so velocities associated with wet steam should be used, typical values being 15–20 m/s. Figure 5.39 is a useful chart for sizing trap discharge lines as it includes a calculation for the amount of flash steam formed. Examples of its use are given.

### *Pump discharge lines*

The discharge line from a pump will contain water only so water sizing charts can be used. The prerequisite for using the chart will again be knowledge of both the flow rate and the pressure drop. Flow rate will depend on the discharge rate from the pump. Pressure drop will depend on the pressure generated by the pump minus any static lift or back pressure in the condensate return system. The pressure drop concerned is that created by friction in the pipe. Before using the entire pump head available, a check should be made on the resulting velocity. Velocities below 1.5 m/s should minimise any noise and erosion.

Charts, such as that shown in Figure 5.39, are suitable for sizing all three types of condensate pipework, as illustrated by the following examples.

**Example A – Drain lines**

*Determine the size of drain line needed to handle a maximum condensate load of 6000 kg/h.*

As the drain line contains only water, the lower part of the chart can be ignored. From the condensate flow rate on the left-hand axis, go horizontally across the chart and read the pipe size on the right-hand side.

Here, the choice is between a 40 mm and a 50 mm pipe. As this is likely to be a short length of pipe, accurate sizing is not necessary. Either 40 or 50 mm could be chosen. The size of steam trap selected may be the deciding factor.

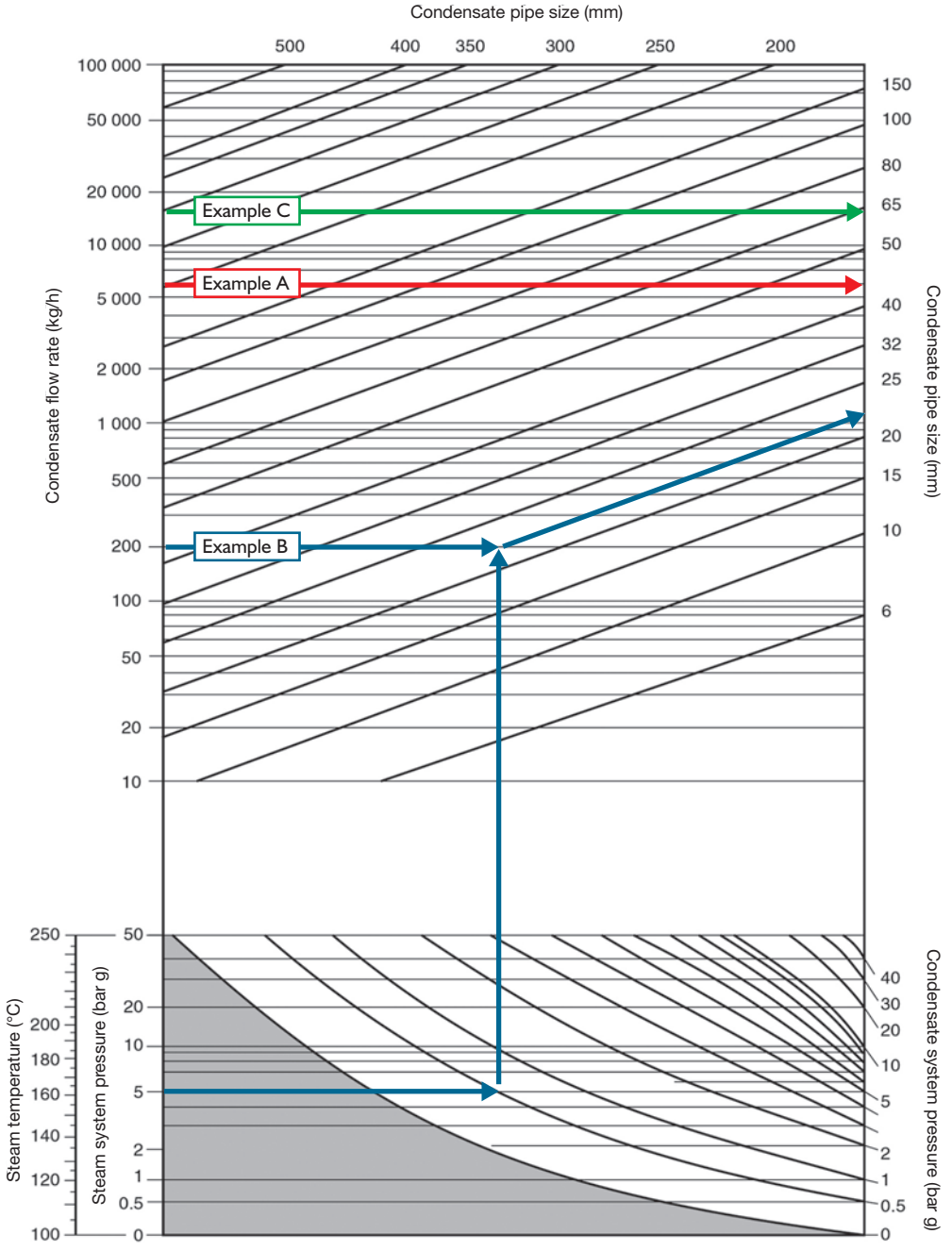



**Figure 5.39** Condensate pipe sizing chart

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**Example B – Trap discharge lines**

*Determine the size of discharge line needed to handle 200 kg/h of condensate discharging from a steam trap draining a 5 barg process. Condensate is lifted from the trap into an overhead return line, creating a total back pressure of 0.5 barg.*

This situation is complicated as the pipe contains a two-phase mixture of flash steam and condensate. The lower part of the chart is used to determine the influence of the flash steam. The upper part determines pipe size in a similar manner to Example A. In this case, the line follows the slope of the pipe size lines, showing a pipe size between 20 mm and 25 mm.

If the pipe is rising, it can be considered to be full of water. Consequently, the passage of flash steam is more difficult. A larger pipe will reduce the flash steam velocity and minimise noise. If the pipe is horizontal or falling, it will not be full of water and there will be more space available for the flash steam. In this case, the smaller pipe size can be chosen.

In Example B, the pipe is rising so 25 mm would be selected.

**Example C – Pump discharge line**

*A mechanical condensate pump has a maximum capacity of 4000 kg/h based on the back pressure that it pumps against and the operating steam pressure. Determine the size of discharge pipe needed.*

The discharge line from a mechanical or electrical pump will contain water only. Only the upper part of Figure 5.39 would be used and, as with the drain line to the trap, a horizontal line taken across to the appropriate pipe size.

The complexity with pumped lines lies in determining the condensate flow rate. The rate at which a pump discharges is *not* the same as the rate at which condensate flows into the receiver. The pump manufacturer's data should enable the discharge rate to be determined. For electrical pumps, a common figure would be 1.5 times the condensate inlet rate. The discharge time of a mechanical pump is typically 25% of the overall cycle time; because of this, a factor of 4 should be applied to the pump's maximum capacity in order to determine the discharge rate.

With Example C, the instantaneous discharge rate will be 4000 × 4 = 16000 kg/h. A horizontal line on the chart indicates that a 65 mm pipe would be appropriate. A smaller pipe may be suitable, particularly if the pump is able to pump against a much higher head than the return system requires. However, before selecting a smaller pipe, a check on the resulting velocity should be made.

*Common return lines*

A common return line is where two pipes connect together. The aim in sizing the common pipe is to keep the velocities equal. For this to happen, the cross-sectional area of the common pipe should be equal to the sum of the areas of the connecting pipes. See the example in Figure 5.40.

Common line 1:

$$= \sqrt{(15^2 + 20^2)} = \sqrt{(225 + 400)} = \sqrt{(625)}$$

$$= 25\,\text{mm}$$

Common line 2:

$$= \sqrt{(25^2 + 25^2)} = \sqrt{(625 + 625)} = \sqrt{(1250)}$$

$$= 35\,\text{mm (choose 40 mm pipe)}$$

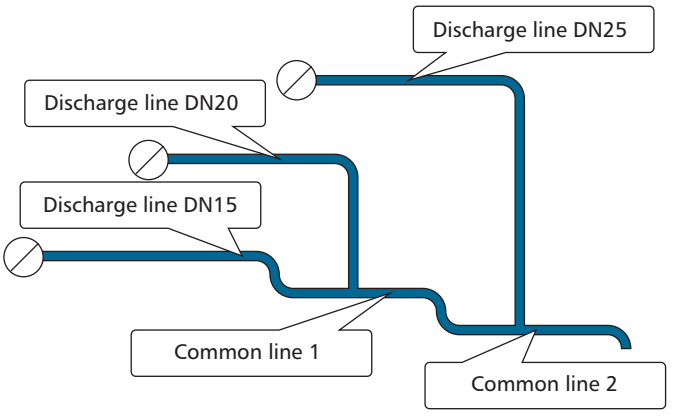



**Figure 5.40** Sizing common lines

# 6 System commissioning

Pipework, valves, fittings, etc. must be designed, constructed and installed in accordance with relevant standards. The umbrella standard relating to steam applications found in building services is the Pressure Equipment Regulations 1999 (as amended).

At the commissioning stage, it has to be assumed that the appropriate standards have been adopted in determining the suitability of the equipment installed and the installation itself. Nevertheless, prior to any commissioning, it is worthwhile conducting an additional initial check.

These are some of the items to consider:

— Does it appear that appropriate standards have been used in the selection of system components?

— Does the pipework installation look robust, with supports where you would expect to see them?

— Does the insulation appear to be in good condition?

— Is there provision for drainage at all low points and are these drains/traps able to readily release condensate?

— Are safety valves fitted where you would expect to see them?

— Are isolation valves fitted and accessible should an emergency shutdown be required?

## 6.1 Pressure testing

Individual system components will have undergone suitable pressure testing as part of their manufacture. Consequently, any pressure testing done as part of a commissioning routine is aimed at checking for system integrity (i.e. any leaks) rather than component strength.

Testing is done by raising the system pressure to a value above normal working pressure (NWP). Typically a factor of 1.5 or 2 times NWP would be used. For example, a steam system operating at 6 barg would be pressure tested to

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9 bar g. Although the condensate system operates at a lower pressure than the steam system, it is common to test both to the same value. This would ensure system integrity should the condensate system be pressurised by steam.

Test pressure could be generated using air or water. Although air is convenient, as it can be easily removed after the test, its use is uncommon. Air, being a compressible fluid, can store significant amounts of energy when under pressure. Consequently, pipework subjected to an air pressure test is theoretically more dangerous than pipework full of an incompressible fluid, such as water. Air leaks are also more difficult to detect.

Cold water is the test medium most commonly used. The process of testing is simple – a hand or electrically operated pump builds up pressure in the system (see Figure 6.1). It is easy to build up pressure as the water is incompressible.

There are some important points to consider as part of the pressure testing process:

— Provision must be made for air to be removed as the system is filled with water. Devices used for this purpose, if manual valves, should subsequently be closed during the test period.

— Any product whose internal structure could be damaged by being subjected to the test pressure should be suitably protected.

— Safety valves could lift if the pressure is raised. Where this is a possibility, the valves should be fitted with test gags (see Figure 6.2). These will hold the valve closed *but must be removed after the test*. An isolation valve at the safety valve inlet which would prevent the valve from lifting under test *must never be fitted*.

— What would be the impact of a water leak, should one occur?

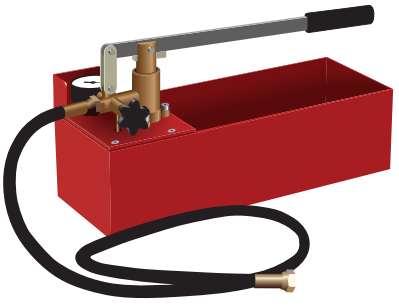


**Figure 6.1** Hand-operated pressure test pump

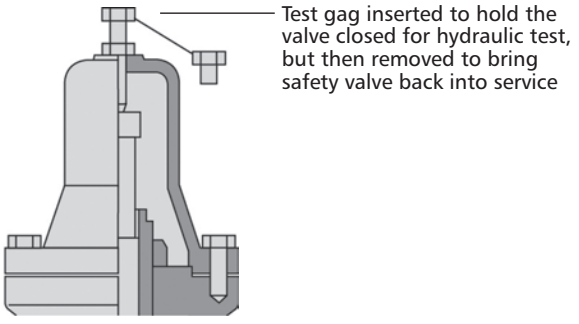



**Figure 6.2** Safety valve with test gag

— Should the system be tested in sections to facilitate inspection and to reduce the filling time?

— How long should the pressure be maintained? A period of 1 hour is a reasonable time.

— What provision has been made for draining the system down after the test?

— Is the test to be done on the basis of visible leaks or based on a decay in pressure. If the latter, how will the leaks that caused the pressure loss be identified?

— Consider whether air removal devices should be opened after the test to aid drain-down.

## 6.2 Flushing out pipework

Steam pipework will almost exclusively be of mild steel construction. This means that, even with new pipework, scale is likely to have formed on the internal surfaces. This scale can be dislodged when steam first flows through the pipe. Along with pipe scale, there is also the possibility of other foreign bodies being present on start-up. These could include casting sand, weld splatter, jointing material, welding rods, nuts and bolts, etc. All of these will be damaging to pipeline components and will need to be removed during the start-up process by flushing through the pipework.

Many components in a steam and condensate system will have pipeline strainers fitted upstream to protect them in normal service. Examples of such components would be control valves, meters and steam traps. These strainers can help with the initial flushing-through process but there is a real danger that, during this process, they can become overloaded and blocked completely.

Problems associated with strainer overload on start-up can be overcome by taking the following steps:

(1) Fitting amply sized drain points where debris can gather and be easily removed. A good example is the dirt/drain pocket illustrated in Figure 6.3. The drain connection for the steam trap comes off the side of the pocket, not the bottom. The flange on the bottom is easily removed for cleaning out. On smaller pipes, this flange may be replaced by a plug or a blowdown valve.

(2) Using a coarser screen in the strainer during commissioning, replacing it with a finer grade for normal operation. Fitting a coarser grade screen does mean that less protection is afforded to the valve, etc. during the flushing-through process.

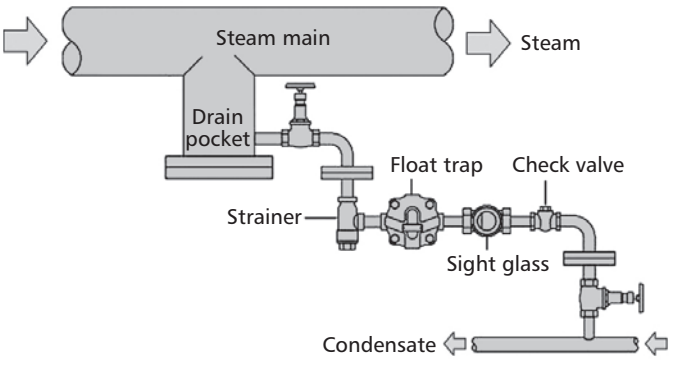



**Figure 6.3** Drain pocket

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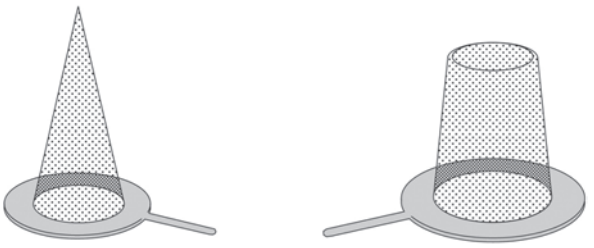


**Figure 6.4** Temporary strainers

(3) Fitting temporary strainers, such as the witches' hat design shown in Figure 6.4. The amount of dirt that a strainer can capture during commissioning should not be underestimated. Cleaning out the strainer or dirt pocket may need to be repeated more than once. The option of taking screens out of strainers (because they could become blocked!) should never be considered.

## 6.3 Start-up

Due to justifiable concerns over increasing fuel costs, more steam systems today are being started and stopped compared to the previous practice of keeping them 'ticking over'. A steam system is at its most dangerous under start-up conditions, but any potential danger can be avoided if some simple rules are applied:

— Follow the boiler manufacturer's start-up instructions. This will generally mean making sure that there is sufficient water in the boiler, the crown valve is closed and the whole boiler structure is warmed up in a progressive manner, dependent on its size.

— Have a properly documented start-up routine in place. This start-up routine should form part of the normal boiler operating instructions, guidance on which is available in the CEA/SAFed publication BG01 – *Guidance on safe operation of boilers*. Reference can also be made to other guidance documents and standards, such as Regulation 11 of the Pressure Systems Safety Regulations 2000, BS EN 12953-13 and the Pressure Equipment Regulations 1999 (as amended).

— Ensure that everyone involved in the start-up process is properly trained and understands the consequences of any actions they take.

— Only start 'unattended' boilers from cold if a risk assessment indicates it is safe to do so.

— Proper provision must have been made for pipe expansion. This does not simply mean the inclusion of expansion joints, etc. but also the proper anchoring and guiding of pipework to ensure that any expansion occurs where it is safe to do so. As an example of expansion, a 50 m long steam pipe operating normally at a pressure of 7 bar g will expand by 100 mm from cold. Figure 6.5 gives similar expansions for different pressures and lengths. This expansion must be catered for. There are various ways to do this and advice should be sought from specialist engineers and suppliers.

— Consider automating the start-up procedure.

— Make sure that any accumulated water is drained from the pipework system before any steam is admitted.

Assuming the boiler has the correct control equipment and is working properly, the time when the greatest risk is present is when the crown valve begins to open to admit steam to the distribution system. The pipework will be cold so any steam entering will quickly condense, causing a vacuum. If any water has accumulated at low points, this can be drawn into the vacuum at very high velocities causing, at best, noise and, in extreme cases, a rupturing of the pipe or valve. This is known as 'condensation-induced' waterhammer and can only occur if accumulated water is present. This accumulated water can also cause the more conventional steam-flow waterhammer, as it is pushed forward by the steam flow itself. Neither form of waterhammer can occur in the absence of accumulated water (see Section 4.1.3).

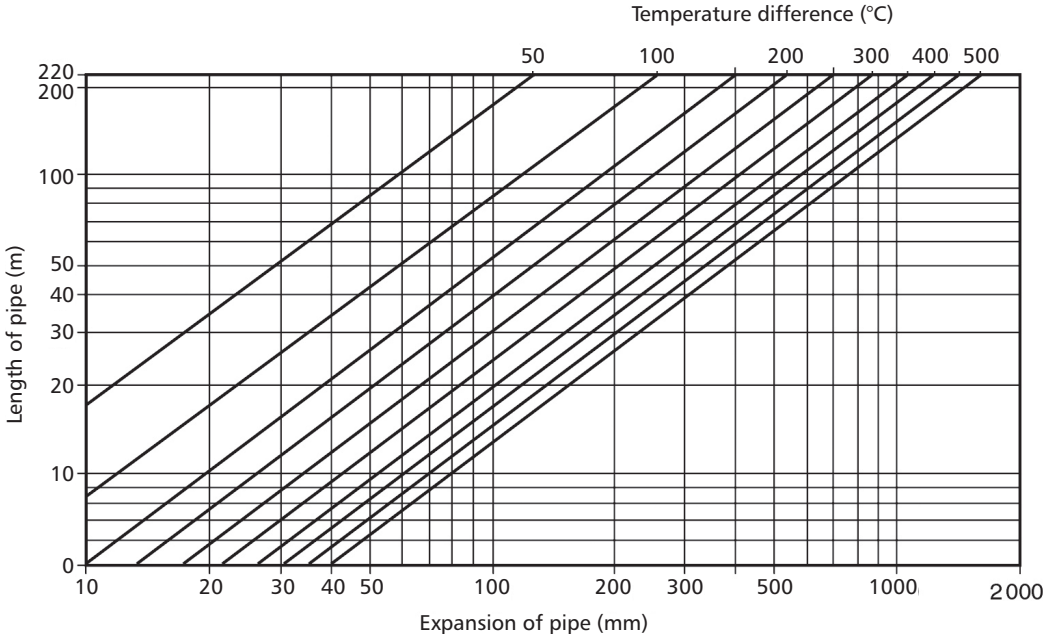



**Figure 6.5** Pipe expansion

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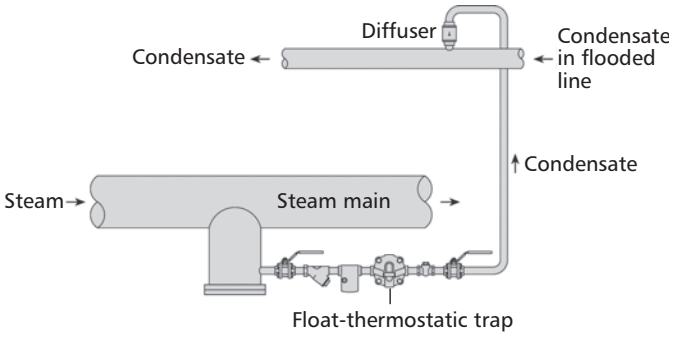



**Figure 6.6** Lift after steam trap

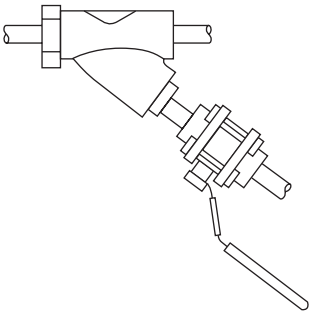


**Figure 6.7** Strainer with blowdown drain valve

At shut-down (see Section 6.4) the steam pressure will drop and in many cases will no longer provide the differential pressure needed across the steam trap for effective condensate removal. At start-up, a slow build-up of pressure is desirable, but it should be remembered that this slow build-up will affect the steam traps' ability to discharge condensate, so even if traps are correctly fitted at all low points, they may be less effective during start-up and shut-down processes. Figure 6.6 shows an example of where this situation can arise, when steam pressure is used to lift condensate.

Where such lifts occur frequently, and on large systems, consideration should be given to fitting a drain valve that can be opened manually at start-up and shut-down to drain condensate by gravity. Such a valve can be incorporated into a pipeline strainer (see Figure 6.7) so combining the need for manual drainage and screen clean out during start-up.

Steam should be admitted into a cold pipework system in a slow and controlled way. The time taken to open the main steam inlet valve will depend on the size of the connected system. For a small system, such as a hospital laundry, this may be done over a 15-minute period, but for a larger system the warm-up process could extend to several hours. The aim is to warm up the pipework slowly to prevent undue mechanical stress, to make sure that water is removed as effectively as possible under low-pressure conditions and to prevent high levels of vacuum occurring because of large quantities of steam condensing. Minimising vacuum will aid condensate removal but, more importantly, it will reduce the risk of condensation-induced waterhammer occurring.

An additional benefit of slowly bleeding steam into a cold pipework system is that the boiler will be able to run at its rated pressure rather than struggle to build pressure as it fills a large pipework system. While it is struggling to reach its rated pressure, the quality of the steam produced will be low, meaning that a mixture of water and steam will be released into the distribution system. This 'wet' steam can cause contamination of control and metering devices as well as adding to any water that may have already accumulated, so potentially increasing the risk of waterhammer.

Control of start-up can be automated using an actuated isolation valve. This would be opened over a period based on time, on pressure or on temperature. Timed control is the most common method, where a controller 'inches' the valve from closed to open over a fixed time period. This period will depend on the installation and expert guidance should be sought to determine an appropriate length of time.

In the arrangement illustrated in Figure 6.8, a separator is fitted upstream of the actuated isolation valve. Because the isolation valve is being opened slowly, it could be operating in an almost closed position for some time. In these circumstances, any steam that contains water droplets (wet steam) flowing through a partially open valve can cause mechanical damage to the plug or seat, such as grooves being cut in mating surfaces caused by the erosive effect of high-velocity steam. This is known as 'wiredrawing' and it can be prevented by ensuring that water droplets are separated from the steam, which is the function of the separator.

Opening a large valve to bleed steam slowly into the pipework may be difficult to control, so an alternative is to use a smaller valve in a bypass (Figure 6.9).

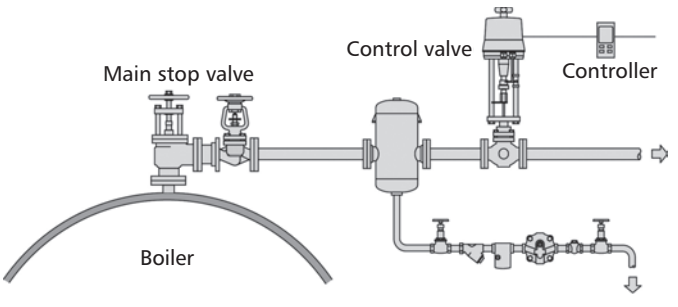



**Figure 6.8** Automatic start-up valve

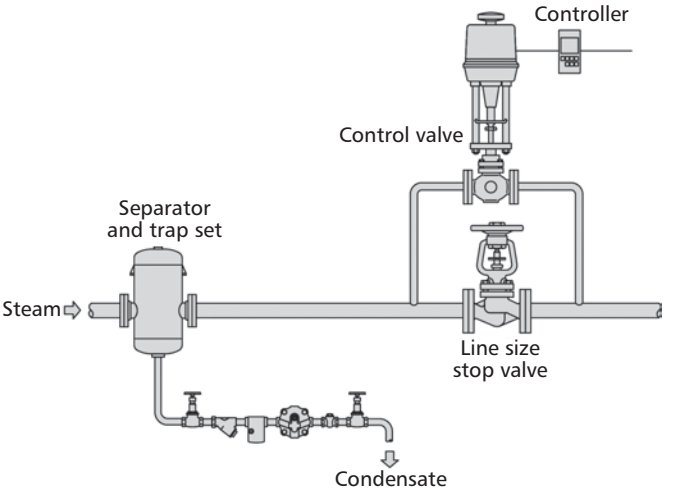



**Figure 6.9** Start-up valve in bypass

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This method is particularly useful if the main isolation valve is of the parallel slide format. A bypass valve will allow the pressure to equalise on both sides of the main valve, so making eventual opening easier.

## 6.4 Shut-down

Shutting a steam system down is a safer operation than starting up. However, the manner in which the system is shut down can influence how safely it can be started up again. When the steam is switched off, the pressure will drop. The ability of a steam trap to remove condensate will be reduced. As the steam condenses further, vacuum conditions can be created in the steam system. Under such conditions, steam traps, even if they discharge to drain, will be ineffective.

Air will find its way through any leak path, even if no steam leaks are evident. It is difficult to prevent air ingress and, indeed, it will be necessary if vacuum conditions are to be overcome and water allowed to drain out. This drainage may be done manually with drain valves upstream of steam traps or with an additional trap in parallel to the main one. This alternative type of trap would operate on the basis of temperature, opening when the temperature of the condensate falls. The condensate from this trap must discharge by gravity. Figure 6.10 shows the use of a liquid expansion trap.

Never be tempted to speed up the shut-down process by increasing heat loss from the pipe (such as by spraying with water). Apart from mechanical stress, this can create the serious problems associated with condensation-induced waterhammer mentioned earlier.

Check also that any steam equipment that is unable to operate under vacuum (a good example being a storage tank or vessel) is fitted with a vacuum breaker device to prevent implosion occurring. This will allow air into the steam system, so provision must be made for ensuring that this air is removed on start-up.

Ensure that check valves form part of any steam trap station where condensate is being lifted (see Figure 6.10). This will prevent the heat exchanger or drain point from refilling with condensate due to backflow when the steam pressure falls.

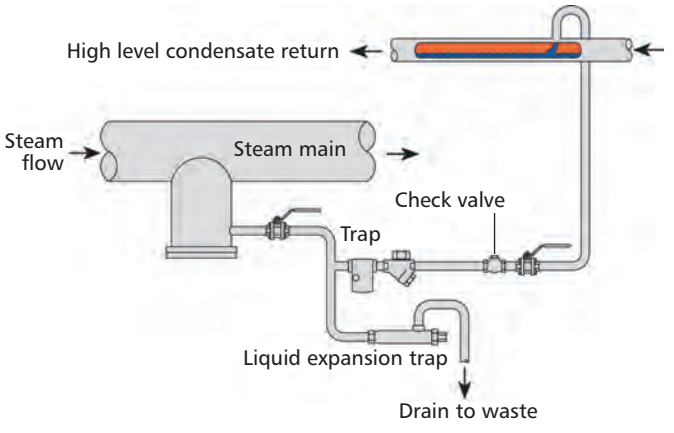



**Figure 6.10** Steam mains shut-down drainage

# 7 Operation of steam boiler plant

## 7.1 Feedwater conditioning

The correct conditioning of boiler feedwater is a specialist subject, with many alternative methods available depending on the condition of the raw water and how the steam is to be used. This memorandum can only cover the general principles involved in treating boiler feedwater. For more details and current best practice, additional guidance on water treatment is available in BS 2486 – *Recommendations for treatment of water for steam boilers and water heaters* and in BS EN 12953-10 *Shell boilers. Requirements for feedwater and boiler water quality*.

### 7.1.1 Effects of incorrect treatment

Steam boiler plant is required to operate safely, efficiently and with minimum maintenance over a long working life. The way in which the water supply to the boiler is 'conditioned' will have a significant effect on many of these factors.

As far as conditioning (or, as it is more commonly referred to, 'water treatment') is concerned, the main aims in the boiler are to prevent the formation of scale on heat transfer surfaces and to prevent internal corrosion. Any hardness in the feedwater which is not treated chemically can result in scale building up on heat transfer surfaces. This scale can reduce heat transfer efficiency and, in extreme cases, lead to tube or furnace failure due to overheating. In the event that hardness is not treated, the frequency of boiler internal cleaning would have to be considerably increased.

Feedwater will contain dissolved gases, such as oxygen and carbon dioxide. These can contribute to corrosion inside the boiler and in the associated steam and condensate system. These gases should be removed as far as possible before the water enters the boiler. The pH value of the boiler water also needs to be controlled. Too low a value can result in acidic attack. Too high a value can exaggerate foaming, leading to the production of wet steam. Sodium hydroxide is a major contributor to increased alkalinity but also an instigator of caustic embrittlement/cracking, leading to mechanical failure of stressed metallic components.

The importance of producing good quality steam is covered in Section 3. Correct water treatment and the proper operation of the steam boiler will both have a very positive effect on the quality of the steam produced, in particular its dryness and cleanliness.

Water inside a steam boiler is far from clean. It contains very high levels of TDS due to the evaporation process. A well-operated boiler, with good water conditioning, will ensure that only dry steam enters the distribution system, and that boiler water is excluded. Ingress of such water (known as boiler water carryover) has the effect of:

— adversely affecting the operation of sensitive devices, such as controls or meters

— potentially blocking condensate removal devices

— forming a barrier to energy exchange across heat transfer surfaces.

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**Table 7.1** Causes and cures for priming and foaming

| | Cause | Explanation | Cure |
|---|---|---|---|
| Priming | Boiler operated at below manufacturer's rated pressure | Low pressure means larger steam bubbles. | Ensure boiler runs at rated pressure by not reducing pressure setting or overloading |
| | Boiler overloaded so that it struggles to build pressure | The water level is closer to the outlet due to presence of a greater number of large bubbles | Fit devices that help maintain boiler pressure, such as accumulators or surplussing valves |
| | Boiler started with crown valve open | Large bubbles splash more when they break the water surface | Start boilers with steam outlet closed |
| | | Steam release velocity is increased | |
| | Water level in boiler is too high | Boiler operating at too low a pressure | As above |
| | | Level controls wrongly set | Better control of water level including resetting of level controls |
| | | | Addition of high level alarm |
| Foaming | Agitation inside the boiler | Compact design of modern boilers means there is less separation space between water and steam outlet | Do not increase the steam release rate by overloading the boiler |
| | | | Consider anti-foaming agents |
| | Water hardness dropped too low | Soft water due to high alkalinity will cause more foaming than harder water | Control hardness at right level with correct treatment |
| | Foreign substance introduced that creates foaming | Oils, fats, milk, etc. are all contaminants that increase the likelihood of foaming occurring | Prevent contaminants reaching the boiler by using contamination control equipment |
| | High levels of TDS | Highly contaminated water means steam bubbles are less likely to burst and hence pass through the steam outlet | Control levels of solids by means of automatic TDS control system |

The principal causes of boiler water carryover are *priming* and *foaming* (see Table 7.1).

Boiler water conditioning is done either external to the boiler – treating the feedwater before it enters – or inside the boiler itself.

### 7.1.2 External water treatment

External treatment is designed to:

— remove oxygen and other dissolved gases present in the make-up water

— remove hardness salts that could cause scale.

#### *Oxygen removal*

Raising the temperature of water reduces its ability to hold oxygen, as seen from Figure 7.1.

A very effective way of removing oxygen without using chemicals is simply to keep the feedtank temperature high. This is achieved by returning more condensate or using boiler steam to heat the tank contents. In theory, if water is at its saturation temperature (100°C at atmospheric pressure) it cannot contain dissolved gases. However, there may be limitations on the operation of the boiler feedpump that prevent a sufficiently high temperature being achieved.

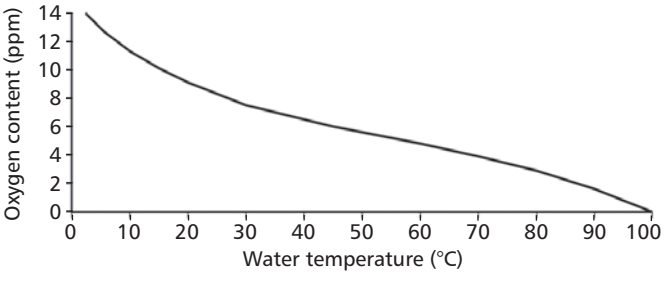



**Figure 7.1** Ability of water to hold oxygen

Where air removal is critical, an atmospheric feedtank could be replaced with a pressurised deaerator. This would be of pressure vessel construction as it is required to operate at pressures just above atmospheric (typically 0.2 to 0.5 bar g) and could reduce oxygen levels down to 20 parts per billion (ppb).

Oxygen removal or 'scavenging' can also be achieved using chemicals. Sodium sulphite is a compound commonly used for this application. Bear in mind that adding chemicals will increase the amount of boiler blowdown required.

#### *Scale prevention*

Hardness in the feedwater will cause scale in the boiler unless it is treated. Various methods of treatment are available and, as this is a specialist subject, expert advice should be obtained. BS 2486 *Recommendations for treatment of water for steam boilers and water heaters* offers guidance.

The type of treatment used will depend on the condition of the raw make-up water, the quantity of condensate returned and the requirements as far as steam quality is concerned; the latter depending on how the steam is to be used.

For the relatively low pressure associated with steam generation for building services purposes, the common types of external scale prevention treatment are base exchange softening and dealkalisation. Reverse osmosis is increasing in popularity as membrane technology develops.

#### *Base exchange*

The base exchange process involves changing the scale-forming calcium and magnesium ions into less harmful

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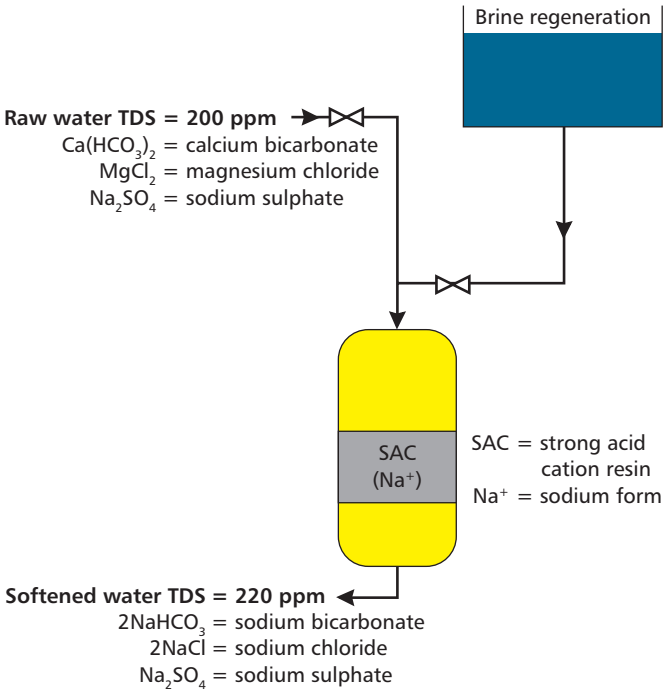



**Figure 7.2** Base exchange process

sodium ions (see Figure 7.2). Resin beads in the ion exchanger are activated by passing a dilute brine solution through the bed. The sodium chloride ensures that water leaving the exchanger is rich in sodium. Harmful calcium carbonate is transformed to less harmful sodium bicarbonate. Sodium salts stay in solution and do not form harmful scale on heat transfer surfaces.

Over time, the resin bed will need to be regenerated. This is achieved using an inexpensive sodium chloride (brine) solution, making base exchange a relatively cheap and reliable means of water conditioning. To reduce the load on the resin bed and hence reduce regeneration time, the water can be pre-softened using a lime/soda process.

Where the raw water is very hard, and where condensate return rates are low (< 50%), a more sophisticated form of ion exchange process, such as dealkalisation, will be needed.

### *Dealkalisation*

Simple base exchange, as the name implies, doesn't reduce contaminants, but converts them to a less harmful form. The level of TDS therefore does not change, nor does the alkalinity of the water.

A dealkalisation unit can be used upstream of the base exchange softener. It removes temporary hardness and so does have an impact on TDS reduction. Raw water passes through a resin bed charged with a weak acid solution. The acid reduces the alkalinity value and, in the process, carbonic acid is formed. This leads to the production of carbon dioxide, which is subsequently removed using a degassing unit.

The dealkalised water is often then passed to a base exchange softening unit, sometimes with the addition of sodium hydroxide to raise the pH value.

There are several types of dealkaliser but the most common variety is shown in Figure 7.3. It is really a set of three units, a dealkaliser, followed by a degasser and then a base exchange softener.

### *Reverse osmosis*

Reverse osmosis (RO) is a process in which water is forced through a semi-permeable membrane leaving a concentrated solution of impurities, which is rejected to waste. Traditionally, the costs of RO plant have been higher than the base exchange and dealkalisation methods described, but technology is improving and costs are falling. RO plant is capable of producing very good quality water (99.5% pure is achievable). By removing contaminants, the TDS level of the feedwater is reduced, therefore requiring less frequent blowdown from the boiler, with a subsequent reduction in lost energy.

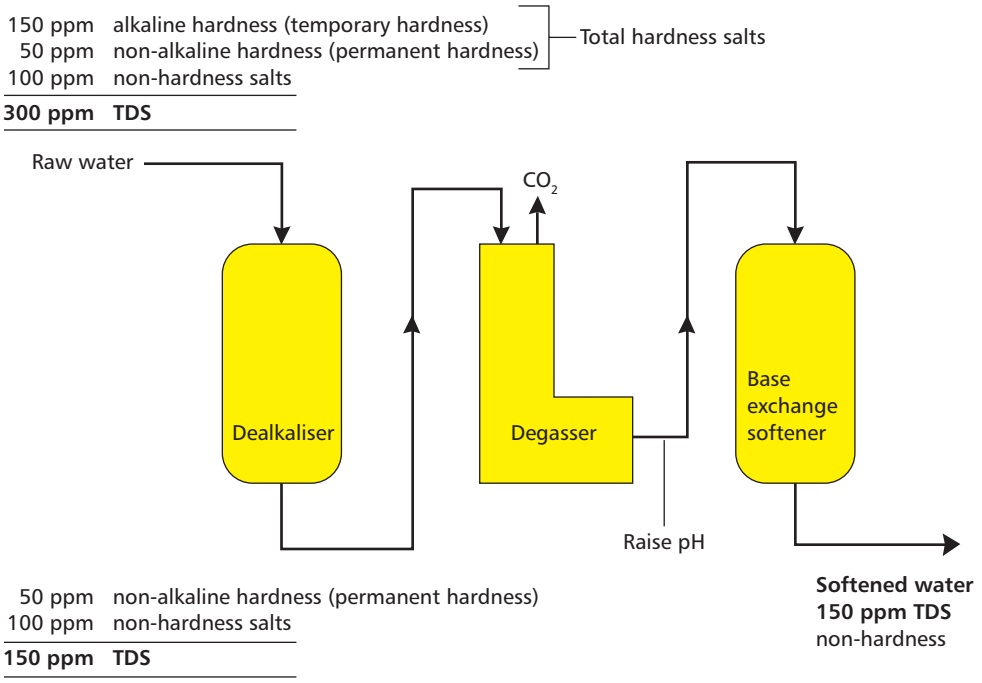



**Figure 7.3** Dealkalisation/degassing process

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Generally, boilers supplying steam for traditional building services purposes, such as heating, hot water production, humidification, sterilisation and even cooling, will be of the shell type. These can withstand fairly high levels of TDS (up to 3500 ppm according to BS 2486), so keeping the feedtank temperature high and base exchange softening are the most popular forms of external treatment with these boilers.

Where unusual circumstances occur, such as raw water being very hard and condensate return rates low, some additional equipment, for example lime/soda and dealkalisation, may have to be considered.

### 7.1.3 Internal water treatment

The major external water treatment options outlined above are aimed at changing scale-forming salts into soft or mobile sludges. However, these sludges can still create problems inside the boiler. They must be kept in suspension to prevent them forming deposits on heat transfer surfaces. If the treatment process is applied correctly, these suspended solids will eventually collect as sludge in the bottom of the boiler and be removed by way of a suitably controlled bottom blowdown valve.

Sludge conditioners are chemicals used to prevent these solids from depositing on metal surfaces and keep them in suspension. Sodium phosphate is commonly used, along with polymer-based sludge conditioners.

## 7.2 Equipment fitted to a modern steam boiler

Steam boilers will normally be fitted with the appropriate equipment by the original manufacturer. Occasionally there may be a need to repair, replace or upgrade a particular piece of equipment. It is important that, when carrying out any work, the function of the apparatus is properly understood.

The ancillary items required on a modern steam boiler, together with their function, are listed in Table 7.2.

**Table 7.2** Main boiler ancillary items

| | |
|---|---|
| Water level controls | To prevent the water level from dropping too low (causing overheating) or rising too high (boiler water passing with steam into the distribution system) |
| Safety valves | To prevent boiler overpressure should normal pressure control fail |
| TDS controls | To control the TDS level to prevent energy wastage (too much blowdown) or boiler priming and scaling (TDS value too high) |
| Boiler crown valve | To enable boiler to be isolated |
| Feedwater check valves | To ensure that water goes into the boiler from the pump and not vice versa |
| Pressure gauge and gauge glasses | To allow visual monitoring of boiler pressure and water level |
| Boiler blowdown controls | To ensure boiler is blown down safely |
| Burner and combustion control equipment | To ensure safe and efficient operation |

### 7.2.1 Water level controls/alarms

Along with the safety valve, these are the most important items fitted to a steam boiler. The maintenance of the correct water level in a steam boiler is essential to its safe and efficient operation. The methods of sensing and the subsequent control of water level is a complex topic that is covered by a number of regulations. These regulations also cover the alarms required for safe operation. BS EN 12953 covers a series of standards, all relating to the fitting of the correct safety equipment on shell boilers.

Level controls or alarms will be either of the float or probe type. Both types can be fitted externally to the boiler. The probe design can also be fitted directly into the boiler shell. With externally mounted controls/alarms, a control chamber is needed. For internally mounted probes, a protection tube is used to ensure the water level sensed is representative of the level in the boiler.

*External level control chambers*

Level control chambers are fitted externally to boilers for the installation of level controls or alarms, as shown in Figure 7.4.

This control chamber could house a float- or a probe-operated device. Being external to the boiler, the level sensed is less turbulent, but purge valves must be used to ensure that the connections to and from the boiler are always clear. Any blockage here could give rise to the controls or alarms in the chamber not detecting low or high water conditions – both dangerous situations. A sequencing purge valve allows the water in the control chamber to be blown down to test controls and alarms are functioning correctly.

Sequencing purge valves are provided by a number of different manufacturers. Each may differ in operating procedure. It is essential that the manufacturer's instructions are followed precisely.

*Internally mounted level controls*

Level control and alarm systems with sensors (or probes) which fit inside the boiler shell are also available (see Figure 7.5). These provide a higher degree of safety than those

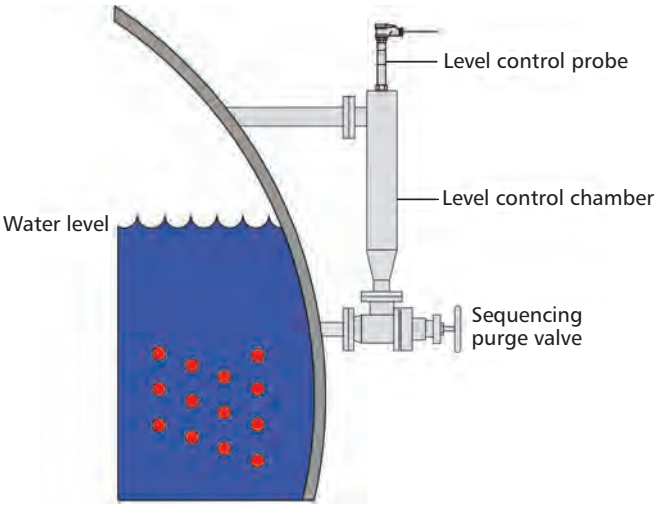



**Figure 7.4** External level control chamber

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fitted externally, requiring less supervision by a trained operator. The level alarm systems should also provide a self-checking function on system integrity and be 'fail safe'. Reference to BS EN 61508-1 provides information on safety integrity levels (SILs).

Because these controls are mounted internally, they are not subject to the procedures required to blow down external chambers. System operation is tested by an evaporation test to 'first low' position, followed by blowing down to 'second low' position. Protection tubes are fitted to discourage the movement of water around the sensor.

*Guidance on safe operation of boilers* (BG01) shows alternative level control and alarm arrangements, making reference to BS EN 12953.

### 7.2.2 Safety valves

A safety valve is required to protect the boiler shell from overpressure and subsequent explosion (see Figure 7.6). Boiler manufacturers will generally supply boilers with an appropriate safety valve(s), manufactured to relevant standards, already fitted.

Strict standards are in place regulating safety valve selection, sizing and installation. For Europe, reference should be made to BS EN ISO 4126-1 and for markets influenced by American standards, ASME 1 or ASME PTC 25.

Many different types of safety valve are fitted to steam boiler plant, but they must all meet the following criteria:

— The total discharge capacity of the safety valve(s) must be at least equal to the 'from and at 100 °C' capacity of the boiler. If the 'from and at' evaporation is used to size the safety valve, the safety valve capacity will always be higher than the actual maximum evaporative boiler capacity.

— The full rated discharge capacity of the safety valve(s) must be achieved within 110% of the boiler design pressure.

— The maximum set pressure of the safety valve shall be the design (or maximum permissible working) pressure of the boiler.

— There must be an adequate margin between the normal operating pressure of the boiler and the set pressure of the safety valve to prevent nuisance discharge.

A key requirement with all types of safety valve is that anything attached to the valve's outlet must not restrict the discharge of steam.

The discharge pipework must be unobstructed and drained at the base to prevent the accumulation of condensate. Discharge pipework should be kept as short as possible with the minimum number of bends to minimise any back pressure, which should be no more than 10% of the safety valve set pressure (BS EN ISO 4126). It will be normal for the internal diameter of the discharge pipework to be larger than the internal diameter of the safety valve outlet connection. In no circumstances should it be smaller. See also Section 5.7.4.

### 7.2.3 Boiler water quality control

The maintenance of water quality is essential to the safe and efficient operation of a steam boiler. The quality of the water in the boiler is expressed as the concentration of TDS.

Controlling the concentration of TDS in the boiler water is achieved by removing some of the water (blowing down) and replacing it with feedwater with a much lower TDS level. There are various ways to do this. One is by manually blowing down the boiler by opening the bottom blowdown valve (see Figure 7.12). A better approach is to use continuous blowdown, typically achieved using a DN15 or DN20 automatic blowdown valve connected to the boiler shell.

Whatever system is used, the TDS in a sample of boiler water is compared with a set point. If the TDS level is too high, a quantity of boiler water is released and replaced with lower TDS feedwater. It is important to control this blowdown process accurately. Too much blowdown means a waste of valuable energy. Too little means that a high level of TDS is retained, resulting in the carryover of contaminated boiler water into the steam distribution system. A typical automatic TDS control system is shown in Figure 7.7.

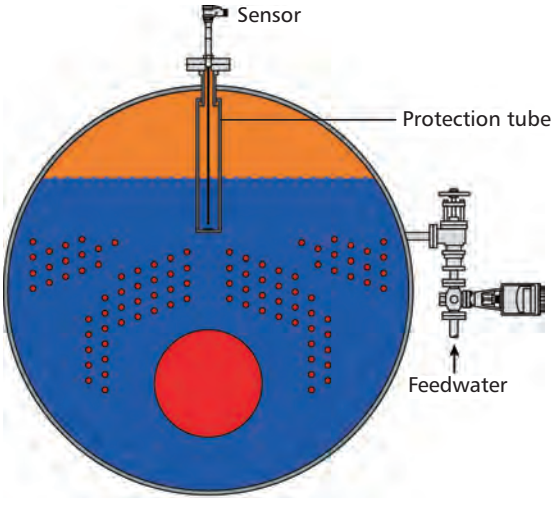



**Figure 7.5** Internally mounted level controls



**Figure 7.6** Boiler safety valve

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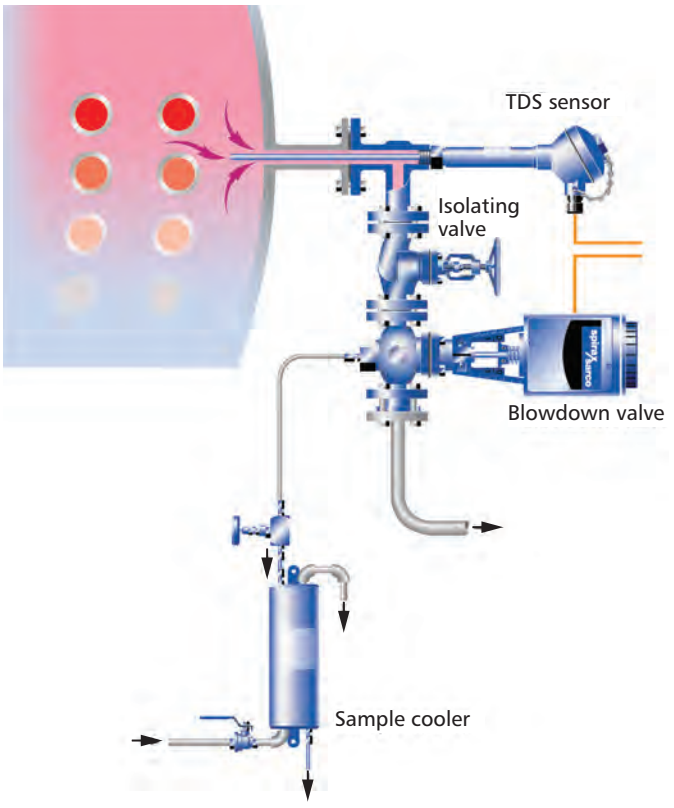



**Figure 7.7** Typical automatic TDS control system

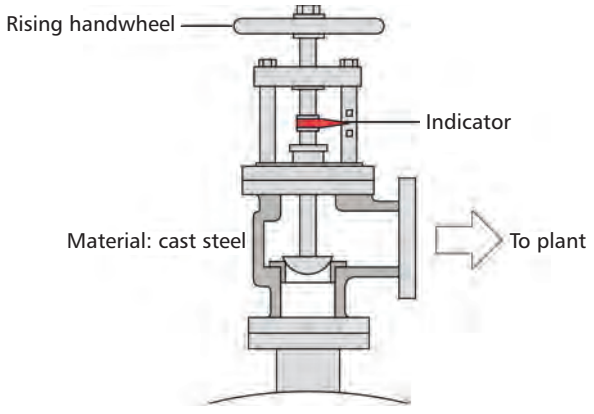



**Figure 7.8** Boiler stop (crown) valve

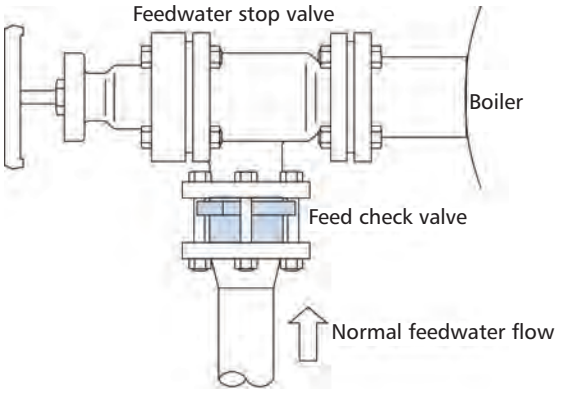



**Figure 7.9** Location of feedwater valves

### 7.2.4 Stop valves and other fittings

A steam boiler must be fitted with a stop valve (also known as a crown valve) which isolates the steam boiler and its pressure from the distribution pipework. It is generally an angle pattern globe valve of the screw-down variety. Figure 7.8 shows a typical stop valve of this type.

The stop valve is not designed as a control valve, and should be either fully open or closed. It should always be opened slowly to prevent any sudden rise in downstream pressure and associated waterhammer (see Section 6.3). This will also help to restrict the fall in boiler pressure and prevent possible carryover of boiler water (priming).

The valve should be equipped with a means of indicating its position. This allows the boiler operator to easily see the valve position, even from floor level.

Where individual steam boilers are connected into a common header, each boiler outlet should have two stop valves fitted, one of which incorporates a non-return feature. A free blowing drain should be fitted in between valves (BS EN 12953-6). This is to prevent one boiler pressurising another.

*Feedwater valves*

A valve to stop the flow of feedwater into the boiler should be fitted between pump and boiler, normally directly on the boiler shell. This arrangement should also include a non-return or check valve (see Figure 7.9).

The check valve (Figure 7.10) includes a spring equivalent to the head of water in the elevated feedtank. This prevents the boiler being flooded by the static head from the boiler feedtank when there is no pressure in the boiler.

*Gauge glasses and fittings*

All steam boilers are fitted with at least one device to indicate the level of the water inside the boiler shell, regardless of the boiler's operating conditions. The indicators are usually referred to as gauge glasses and must comply with BS 3463. They should also be fitted with a protector around them, but this should not hinder visibility of the water level (see Figure 7.11).

*Pressure gauge*

All boilers must be fitted with at least one pressure indicator. The usual type is a simple pressure gauge with a minimum dial diameter of 100 mm, connected to the steam space



**Figure 7.10** Boiler check valve

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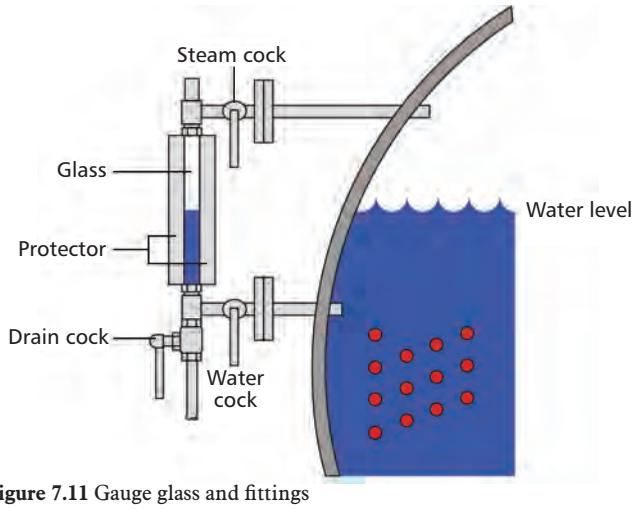



**Figure 7.11** Gauge glass and fittings

either directly or through the water level gauge column or its steam connection. The maximum allowable pressure must be indicated by a fixed and readily visible red mark on the pressure gauge.

*Bottom blowdown valve*

Assuming that the water has been conditioned properly, any sediment will accumulate in the bottom of the boiler. This must be periodically removed through the bottom blowdown valve. This is a large (usually 25–50 mm), rapidly opening valve, commonly a quarter-turn, key operated type.

In a typical safety routine, this valve might be opened for a period of about 5 seconds, once per shift. This process can be manual, but a better option is an automated valve, generally controlled on a timed basis.

Figure 7.12 illustrates the typical position of a bottom blowdown valve in a blowdown system.

### 7.2.5 Combustion control equipment

Combustion control equipment would generally consist of the burner (or fuel feeder in the case of a solid fuel/biomass boiler), forced draft fan to provide combustion air, induced draft fan to remove exhaust gases, air and fuel dampers, ignition equipment and flame controls.

Proving controls should be fitted to ensure that the air to fuel mixture is correct and the flame is on or off as appropriate. These controls must ensure that the boiler is locked out should any of these parameters not be as required.

## 7.3 New regulations and the training of boiler operators

### 7.3.1 Regulations

There are a number of different health and safety rules relating to the design and operation of steam boilers. These rules are generally country-specific so checks need to be made on the regulations in place where the boiler is to be installed and operated. In the UK, the body overseeing such regulations and codes of practice is the Health and Safety Executive (HSE). Some of the regulations relating to steam boiler installations are:

— Pressure Equipment Regulations 1999 (as amended)

— Pressure Systems Safety Regulations 2000

— Management of Health and Safety at Work Regulations 1999

— Provision and Use of Work Equipment Regulations 1998.

It is beyond the scope of this publication to examine these regulations in any detail. The HSE publication *Safe management of industrial steam and hot water boilers* is a useful reference document. So too is *Guidance on safe operation of boilers* (BG01) produced by the Combustion Engineering Association (CEA) and the Safety Assessment Federation (SAFED) in consultation with HSE.

### 7.3.2 Training

Anyone designing a steam system has to be competent to do so. This competency should encompass an awareness of basic steam system design principles, and of the standards that exist in relation to pipework, valves, pressure vessels, etc. In particular, an awareness of requirements for compliance with standards relating to safety equipment is essential, examples of such standards being BS EN 12953 and BS EN 61508.

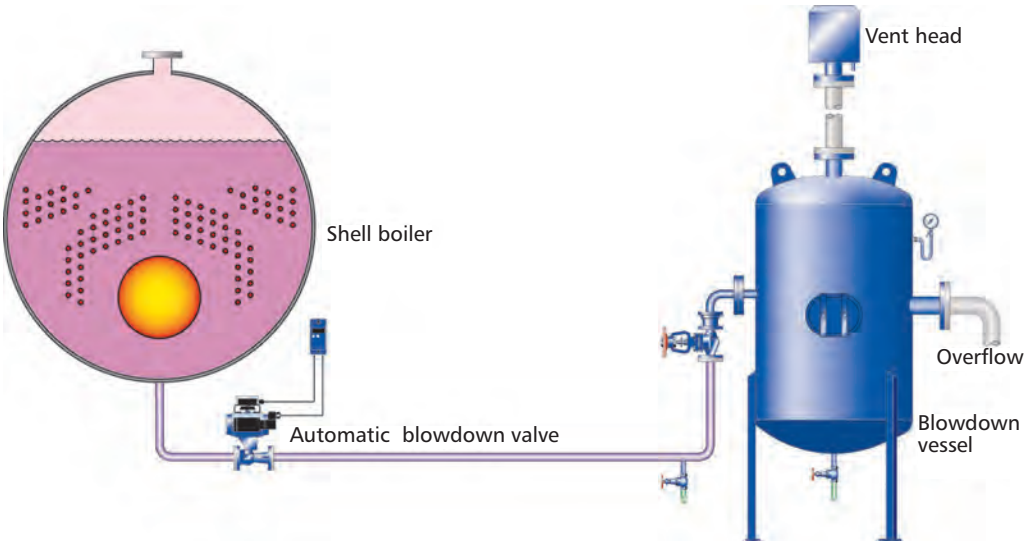



**Figure 7.12** Typical position for a bottom blowdown valve

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Guidance on training in the operation of steam systems is more readily available. *Safe management of industrial steam and hot water boilers* and *Guidance on safe operation of boilers* (BG01) both mention training requirements in some detail. Some of the key points are listed below:

— Employers must ensure that operators know enough about their steam boiler to operate it safely and efficiently.

— Training should be provided in a properly structured way.

— Trainers must be both competent and experienced.

— The Boiler Operation Accreditation Scheme (BOAS) is recommended.

— Regular assessments should be carried out to ensure that training is implemented and not forgotten.

— Retraining is important when any boiler modifications are made.

# 8 Efficiency and sustainability

## 8.1 Heat recovery in the boilerhouse

The steam boilerhouse is an area where major heat exchange processes take place, making it an ideal location to consider heat recovery techniques. The principal areas where heat may be lost in a boilerhouse are:

— up the chimney/stack

— by convection and radiation from the hot surfaces

— by loss of steam from vent pipes

— through TDS blowdown.

### 8.1.1 Losses up the chimney/stack

In a typical three-pass wet back boiler, the temperature of the hot flue gases passing up the chimney will be around 200°C. These hot gases represent a heat recovery opportunity but, before investing in this, checks should be made to ensure that the temperature of the flue gases is not higher than it should be. Reasons for a higher than required temperature would be:

— incorrect burner setting, resulting in more energy being produced from the fuel than is required for a specific steam output

— poor heat exchange in the boiler, commonly caused by fouling or scaling of heat transfer surfaces on both the flue gas and on the water side.

Assuming that both the above causes have been investigated and rectified, some of the heat in the flue gases can be recovered by a heat exchanger mounted in the flue gas outlet, referred to as an economiser. Boiler feedwater is passed through the economiser, raising its temperature. In broad terms, a 10°C rise in feedwater temperature results in a 2% improvement in boiler efficiency. The feedwater to the economiser should be from the discharge side of the pump to allow higher temperatures to be achieved without boiling or pump cavitation problems (see Figure 8.1).

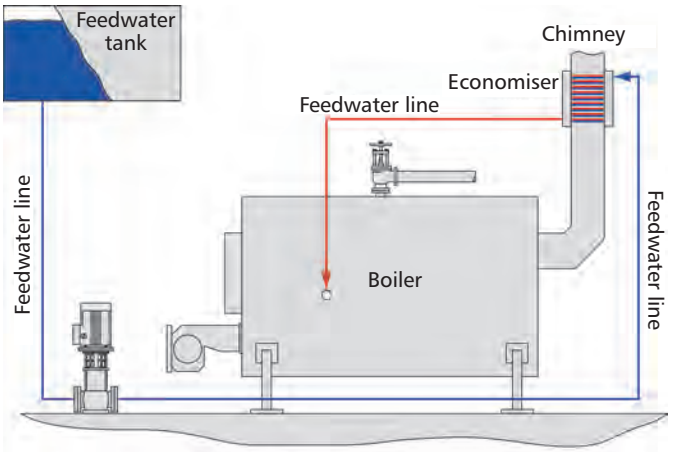



**Figure 8.1** Fitting of an economiser

There are two important considerations to take into account in the design of an economiser installation:

— the flow of water through it should be continuous to maximise heat recovery – this requires the boiler to have modulating level controls, not an on–off control

— the temperature of the flue gases should not drop below their dew point, otherwise acidic, and hence corrosive, condensate will be produced.

### 8.1.2 Convection and radiation from hot surfaces

Hot surfaces, such as the boiler shell, pipework and feedtanks, can all lose heat through convection and radiation. Good insulation will minimise these losses. Insulating the steam pipework has a double benefit, with reduced heat loss not just saving energy but also preventing the formation of wet steam.

A well-insulated shell boiler may only lose 0.5% of its full load rating. Although this may not appear significant, it will be a constant loss irrespective of steam output. Consequently, the percentage loss on a lightly loaded boiler will be much higher than 0.5%. For this reason, boilers are best operated close to their maximum capacity.

Feedtanks and deaerators operate at temperatures well below the boiler, but they have a large surface area, which means there is the potential for significant energy loss. These too should be well insulated. Insulation not only reduces energy losses, it also makes the boilerhouse a safer and more comfortable place to work.

### 8.1.3 Loss of steam from vent pipes

These vents will either be safety valve discharge pipes or atmospheric vents on tanks. Leaking safety valves can be the result of an incorrect setting. This would be unusual on a boiler as set pressures are strictly governed by the boiler design pressure and would normally be set by a competent person.

A leak from a safety valve vent is usually just a wisp of steam. A vent pipe discharging larger quantities is likely to be fitted to a feedtank handling returned condensate. Condensate return pipework will contain a mixture of flash

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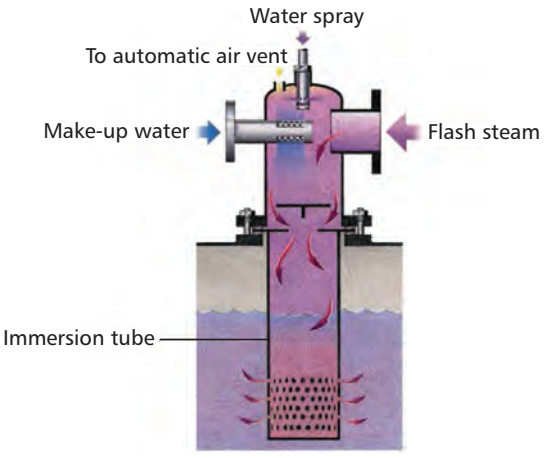



**Figure 8.2** Flash condensing deaerator head

steam and water. It may also contain some live steam if there are leaking steam traps discharging into it.

Flash steam is best recovered and used close to the point where it is generated (see Section 8.4). However, if this does not happen and it returns to the boilerhouse, it should be condensed, saving not just heat energy but conditioned water as well. Small quantities of flash steam can be condensed in the feedtank/deaerator using a flash condensing deaerator (see Figure 8.2).

Where larger quantities are involved, separating the flash steam from the condensate and then condensing this flash steam in a heat exchanger will recover energy. The separated water (condensate) can also pass through a heat exchanger. These heat exchangers can be mounted in series to pre-heat the boiler feedwater in a similar way to an economiser, with both being fitted on the downstream side of the feed pump (see Figure 8.3).

If the flash steam can't be condensed and used prior to entering the feedtank, an exhaust vent condenser (see Figure 8.4) can be fitted to simply quench any flash steam that would otherwise be lost. The condensate formed falls back into the feedtank to be reused.

### 8.1.4 TDS blowdown

When steam is formed in a boiler, the dissolved solids that come in with the feedwater will become more concentrated. The TDS control mentioned in Section 7.2 will bleed off some of this concentrated water, allowing it to be replaced by make-up water with a lower TDS value. Obviously, this loss of boiler contents means a loss of energy and water.

The water will be contaminated, perhaps at a level of up to 3500 ppm. If passed through a flash vessel, some water will flash off as steam. This steam, being free from impurities, can be injected directly into the boiler feedtank to pre-heat the contents. The now more heavily contaminated water still contains some heat energy. Passing it through a heat exchanger which has cold make-up water flowing through the secondary side will recover some of this heat before eventually the now cooler but still contaminated water is discharged to drain (see Figure 8.5).

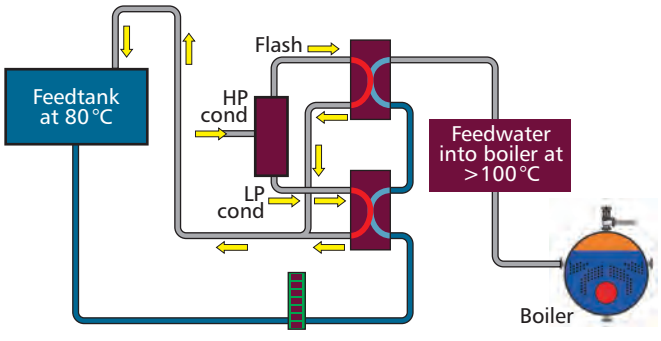



**Figure 8.3** Flash steam used to pre-heat boiler feedwater

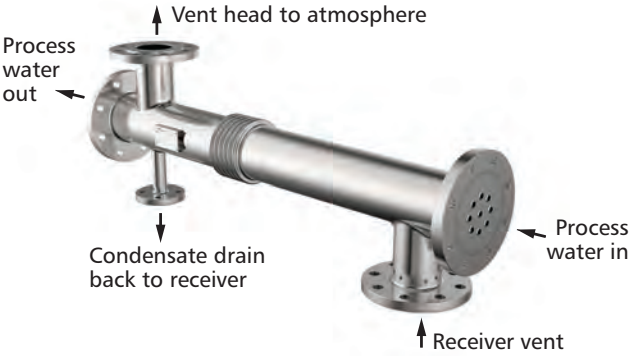



**Figure 8.4** Vent condenser

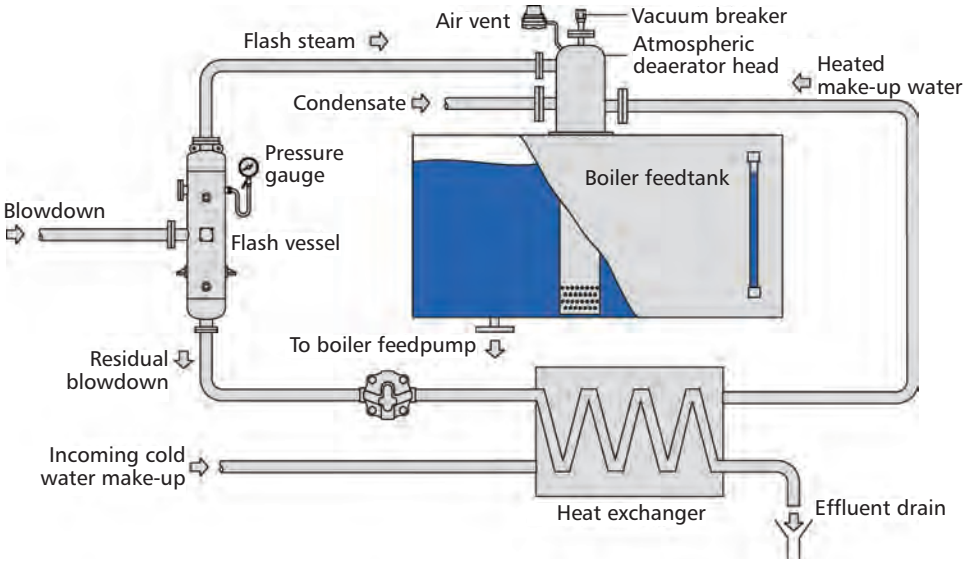



**Figure 8.5** TDS blowdown heat recovery

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Using this flash vessel and heat exchanger system, at a conservative estimate some 80% of the energy in the blowdown can be recovered. The water savings from the condensed flash steam add to the benefit.

For maximum benefit, the TDS control should provide a continuous flow of boiler water.

The other method of blowdown applied to a steam boiler is through the bottom blowdown valve. Here the aim is to remove accumulated sludge. Discharge needs to be rapid so bottom blowdown is an on–off process undertaken as required, typically once per shift with the valve opened only for a few seconds.

Because of the infrequent operation of this blowdown valve, the heavily contaminated nature of the water and the violence of discharge, trying to recover heat is not a viable proposition.

## 8.2 Reuse of condensate

### 8.2.1 Benefits of returning condensate

Condensate is too valuable a commodity to be simply discharged down the nearest drain. Regulations may prevent this practice anyway, due to temperature issues.

Condensate contains around 20% of the heat added in the boiler. This consideration, coupled with the fact that it is effectively distilled water, makes it clear why every attempt should be made to collect and return it. Returning condensate therefore:

— saves fuel, as it is already hot water

— saves water (any condensate returned means less make-up water to purchase)

— saves water treatment (condensate is distilled water and should require no further treatment)

— reduces blowdown rates and hence heat loss (boiler blowdown rates are affected by the quality of the make-up water, not the returned condensate)

— reduces effluent disposal charges (if applied).

It is common for operators to over-estimate the percentage of their steam output that is returned as condensate. A reasonably accurate assessment of percentage return rate can be achieved by measuring the conductivity (TDS value) of the make-up water, the condensate return and the feedwater going into the boiler. A simple ratio calculation will determine the percentage return:

$$\%\ \text{rate of condensate return} = \frac{\text{make-up water TDS} - \text{feedwater TDS}}{\text{make-up water TDS} - \text{condensate TDS}} \times 100$$

### 8.2.2 Mechanical and electrical pumps

Condensate can be returned by gravity, although this is somewhat unusual unless the boilerhouse is at the bottom of a hill or in the basement of a high-rise building. More commonly, condensate would be drained by gravity to a local collecting point. Here, a pump would be fitted to pump the condensate back to the boilerhouse (see Figure 8.6). These pumps can be electrically or mechanically operated.

Table 5.10 compares the advantages of mechanical versus electrical condensate pumps.

### 8.2.3 Condensate contamination

Occasionally, condensate is not returned because it might be contaminated. This contamination is generally from process applications and the contaminants may be oils/fats (from dairy or oil refining), acids (from metal treatment processes), sugars (sugar refining ) or other compounds that would cause difficulties if they were to get into the boiler.

Various types of contamination detection equipment are available to determine when contamination occurs and, in those circumstances, discharge the condensate to an alternative location. A typical arrangement would be as shown in Figure 8.7.

## 8.3 Heat loss reduction

Heat loss from pipework and components in the steam distribution system represents not just a waste of energy but also a reduction in dryness fraction – the steam becomes wetter and contains less heat energy.

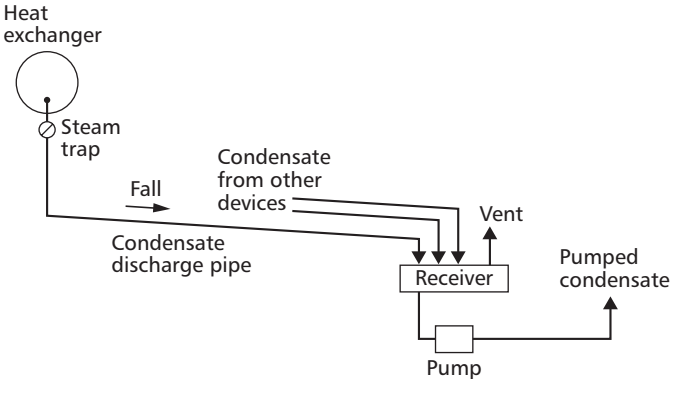



**Figure 8.6** Condensate collection and return

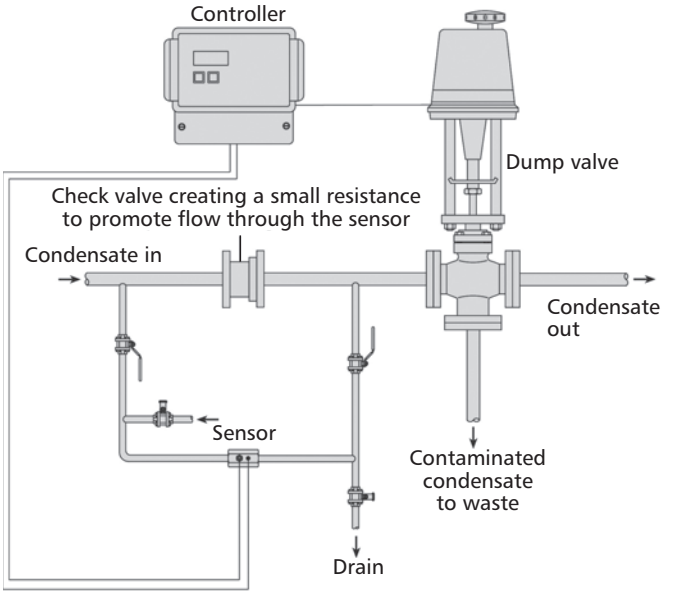



**Figure 8.7** Condensate contamination detection

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Heat lost from the condensate return system means that more fuel has to be consumed in the boilerhouse.

Insulation is the most cost-effective way of reducing heat loss. Various types of insulation materials can be used, and an even wider range of surface finishes, depending on the application. Determining the type of insulation, the thickness and the surface finish are specialist subjects, outside the scope of this publication, but some general principles apply.

First, heat loss is based on the formula:

$$Q = U A \Delta T$$

where:

$Q$ = the heat lost

$U$ = the overall heat transfer coefficient

$A$ = the area of heat transfer surface

$\Delta T$ = the temperature difference between the surface and the surrounding atmosphere.

Insulation efficiency increases with thickness but additional thickness means a greater surface area from which heat can be lost. Increasing thickness therefore reduces $U$ but increases $A$. A balance must be struck between $U$ and $A$ to achieve the most economical thickness.

The temperature differential ($\Delta T$) has a major influence on heat loss. Table 8.1 shows the heat loss from various sizes of *uninsulated* pipe with varying values of $\Delta T$. An estimate of steam pipe heat loss can be made using this table and a general insulation efficiency of 90% (this should be checked with the manufacturer but using 10% of the values in the table is a good, if conservative, starting point).

Second, compared to other pipeline components, insulation is fragile and should be protected. Any insulation which is removed for maintenance/inspection purposes should be replaced and any which is damaged should be repaired. Insulation that has become waterlogged due to leaks or being submerged in underground ducts will increase the heat loss from the pipe and, in some circumstances, wet insulation can be worse than no insulation at all.

Third, the larger the surface area, the greater the heat loss. Flanges add to a pipe's surface area and are equivalent in heat loss terms to approximately 300 mm of bare pipe. A stop valve would be the equivalent of approximately 1.2 m.

Other items, such as mechanical pumps and separators, will have larger surface areas so it is even more important to insulate these.

One other large surface area is the receiver on an electrical condensate pump set. Care should be taken in considering whether to insulate this as the subsequent higher temperature of the condensate may cause cavitation in the centrifugal pump, so affecting its operation.

Finally, insulating hot surfaces is generally done to prevent heat loss but it also enhances the safety of the steam and condensate system. Steam pipes can be at temperatures up to 200 °C in building services applications. Condensate return lines will generally be <100 °C. Bare surfaces on both pipework systems are therefore able to cause burns and must be suitably protected in areas that are accessible to personnel.

**Table 8.1** Heat loss (W/m) from uninsulated steel pipe in 20 °C ambient temperature

| Temperature differential steam to air (°C) | Pipe size (mm) | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|
| | 15 | 20 | 25 | 32 | 40 | 50 | 65 | 80 | 100 | 150 |
| 50 | 56 | 68 | 82 | 100 | 113 | 136 | 168 | 191 | 241 | 332 |
| 60 | 69 | 85 | 102 | 125 | 140 | 170 | 208 | 238 | 298 | 412 |
| 70 | 84 | 102 | 124 | 152 | 170 | 206 | 252 | 289 | 360 | 500 |
| 80 | 100 | 122 | 148 | 180 | 202 | 245 | 299 | 343 | 428 | 594 |
| 100 | 135 | 164 | 199 | 243 | 272 | 330 | 403 | 464 | 577 | 804 |
| 120 | 173 | 210 | 256 | 313 | 351 | 426 | 522 | 600 | 746 | 1042 |
| 140 | 216 | 262 | 319 | 391 | 439 | 533 | 653 | 751 | 936 | 1308 |
| 160 | 263 | 319 | 389 | 476 | 535 | 651 | 799 | 918 | 1145 | 1603 |
| 180 | 313 | 381 | 464 | 569 | 640 | 780 | 958 | 1100 | 1374 | 1925 |
| 200 | 368 | 448 | 546 | 670 | 754 | 919 | 1131 | 1297 | 1623 | 2276 |
| 220 | 427 | 520 | 634 | 778 | 877 | 1069 | 1318 | 1510 | 1892 | 2655 |

## 8.4 Flash steam recovery

### 8.4.1 Flash steam formation

The first step in a flash steam recovery application is to establish whether the amount of flash steam that is being generated could be reduced, without impacting the performance of the steam plant. This may mean using a different type of steam trap to sub-cool the condensate, using a condensate cooler/flash steam condenser as part of the heat exchanger package or employing a method of temperature control based on controlling the flow of condensate out of a heat exchanger rather than the steam flowing into it.

On the basis that some flash steam has formed, it should be captured or recovered and reused. It has exactly the same heat content as live steam but will generally be at a lower pressure. Because the pressure is low, it is unrealistic to pipe it over long distances (as large diameter pipes would be needed). So a prime consideration has to be to find an application for it close to where it is generated. This application needs to have a requirement for steam that is in line with the flash steam being formed.

Finally, to make a flash steam recovery system viable, a reasonable quantity of flash steam must be formed. This is not usually a problem as a very rough estimate of flash steam formed would be 10% of the condensate load. The true calculation is explained below.

Flash steam is formed when the pressure of hot condensate is lowered. This happens downstream of a steam trap. The amount of flash steam formed, as a percentage of the condensate flow rate, is calculated as:

$$\frac{\left(\begin{array}{c}\text{enthaply of}\\ \text{saturated water at}\\ \text{upstream pressure}\end{array}\right) - \left(\begin{array}{c}\text{enthalpy of}\\ \text{saturated water at}\\ \text{downstream pressure}\end{array}\right)}{\left(\begin{array}{c}\text{enthalpy of evaporation}\\ \text{at downstream pressure}\end{array}\right)}$$

or

$$\frac{h_f\,(\text{at } P_1) - h_f\,(\text{at } P_2)}{h_{fg}\,(\text{at } P_2)}$$

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**Example**

*Steam at a pressure of 7 bar g is supplied to a kitchen boiling pan. Condensate is formed and enters the steam trap as saturated water, at a gauge pressure of 7 bar g and a temperature of 170 °C. Steam tables indicate that the amount of heat in the condensate at this pressure is 722 kJ/kg.*

After passing through the steam trap, the pressure in the condensate return line is 0 bar g. At this pressure, the maximum amount of heat each kilogram of condensate can hold is 419 kJ and the maximum temperature is 100 °C. There is an excess of 303 kJ of heat, which evaporates some of the condensate into steam. Figure 8.8 demonstrates the formation of flash steam.

The heat needed to produce 1 kg of saturated steam from water at the same temperature, at 0 bar gauge, is 2257 kJ. An amount of 303 kJ can therefore evaporate:

$$\frac{303}{2257} = 0.134 \text{ kg steam per kg condensate}$$

From each kilogram of condensate in this example, the proportion of flash steam generated therefore equals 13.4% of the initial mass of condensate.

An alternative to doing the calculation is to use a chart, such as that shown in Figure 8.9.

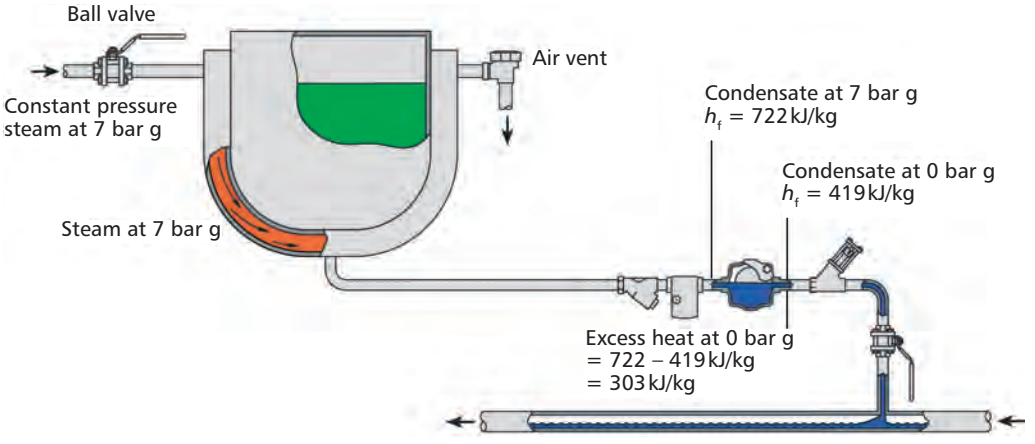



**Figure 8.8** Flash steam formation

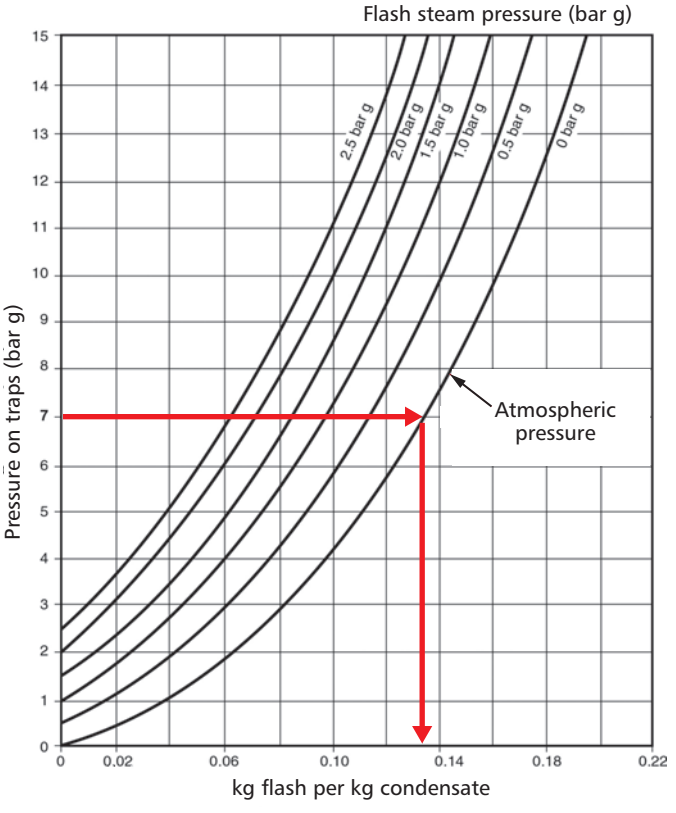



**Figure 8.9** Flash steam formation chart

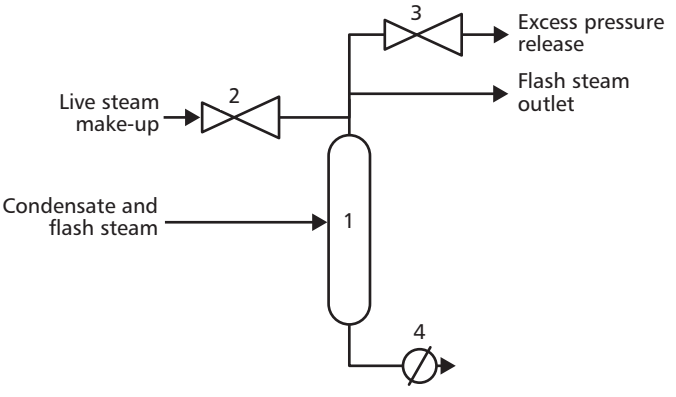



**Figure 8.10** Flash steam recovery system components

## 8.4.2 Flash steam recovery system components

Flash steam originates from hot condensate, so it will be formed in the condensate return system. The first step is to separate the flash steam from the condensate using a flash vessel (item 1 on Figure 8.10).

The next component to add is a means of controlling the flash pressure – a typical pressure being 1 bar g – and supplying live steam to the process when no flash stream is being generated. This would be a steam pressure reducing valve (item 2 on Figure 8.10).

The third item is a control mechanism to release pressure when there is no demand for the flash steam. A safety valve could be used here but a better choice would be a back pressure or surplussing valve (item 3 on Figure 8.10). With such a valve, the pressure is measured *upstream* and the valve progressively opened as the upstream pressure rises.

Finally, a steam trap set (item 4 on Figure 8.10) is required to prevent flash steam being lost into the condensate return system.

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### 8.4.3 Typical flash steam recovery applications

Flash steam recovery from three different systems – air heater batteries, unit heaters and TDS control blowdown – is detailed in Figures 8.11–8.13.

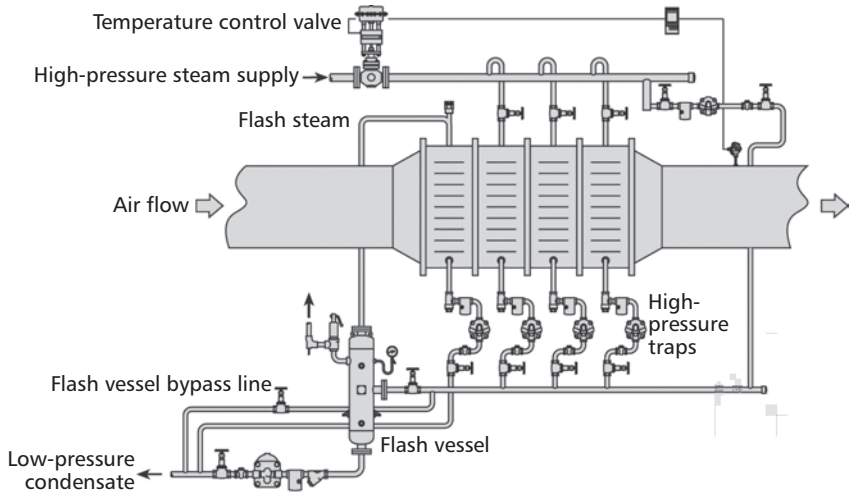



**Figure 8.11** Flash steam recovery from air heater batteries

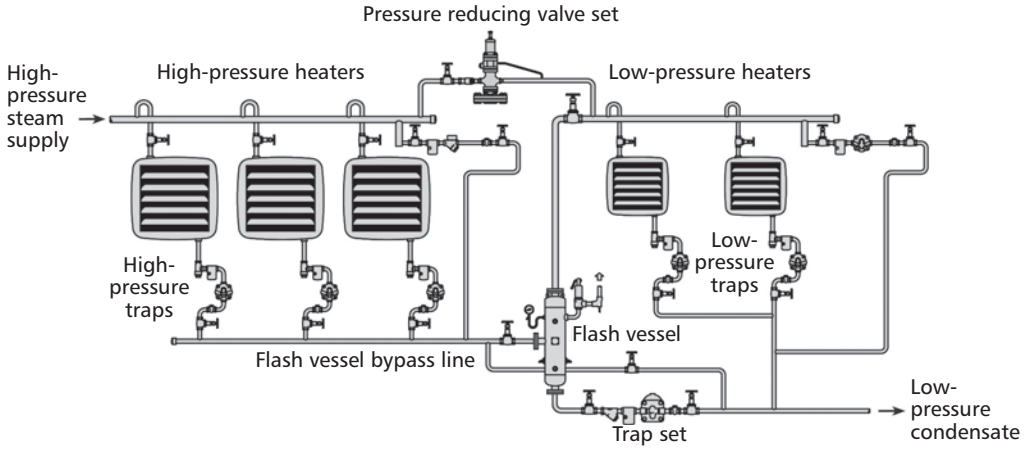



**Figure 8.12** Flash steam recovery from unit heaters

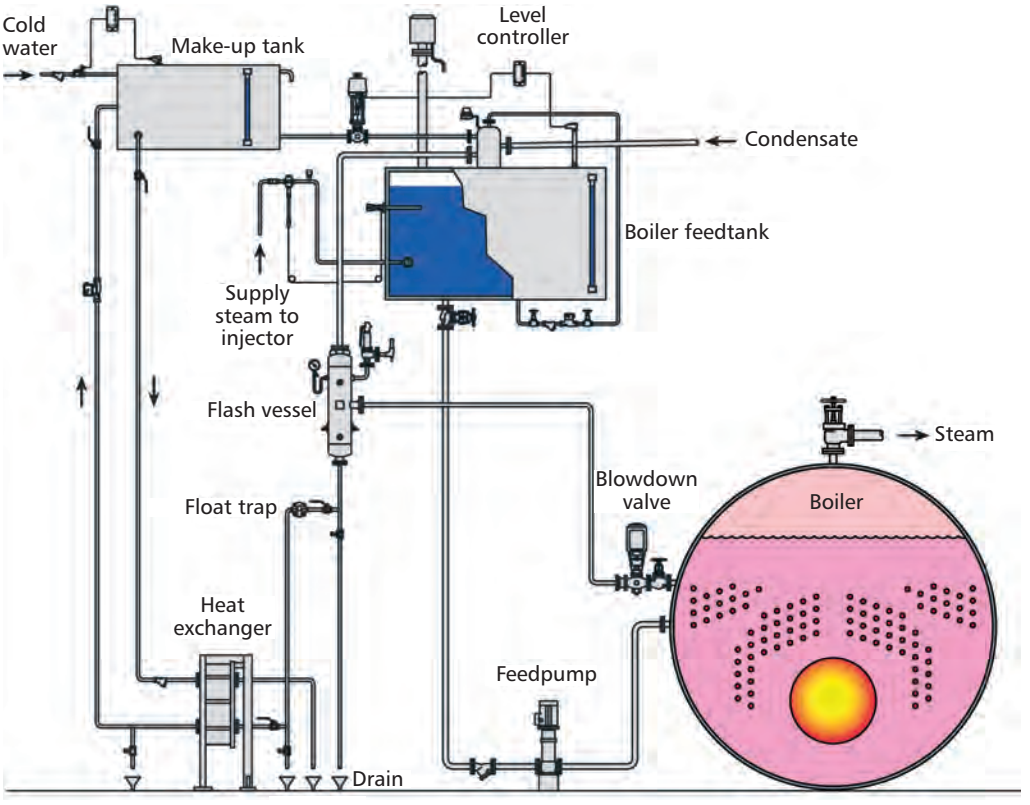



**Figure 8.13** Energy recovery from TDS control blowdown

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## 8.5 Preventing live steam loss

Live steam is lost from the distribution system though holes in the pipe (caused by erosion/corrosion) or loose items such as flanges, valve glands and pipe fittings.

Using the correct water treatment, operating the boiler correctly and making sure that pipes are accurately sized will prevent holes being formed.

Leaks from joints are more common than holes. These can be caused by poor initial installation, using incorrect jointing materials, inadequate provision for expansion, impact caused by waterhammer, etc. Welded connections are common in some industries, virtually eliminating leaks. Building services pipework connections tend to be screwed or flanged, both of these methods being prone to leaks.

Valve gland seals are a common area for steam leaks. This could be caused by general wear and tear due to frequent use or a lack of proper maintenance. An alternative to the conventional packed gland is a bellows sealing arrangement, as illustrated in Figure 8.14, which is suitable for both isolation and control valves.

Wet steam tends to be more 'searching' than dry saturated steam, so a large number of leaks may be a sign that the steam quality (wetness) is poor. Any deposits seen on the outside of gland-packed valves can also be an indication of boiler water carryover.

Leaking steam from vent pipes could be flash steam, mentioned in the previous section. If the vent pipe is a safety valve discharge, any steam escaping is a sign of a leaking safety valve. The most common cause of a leaking safety valve is incorrect setting. The aim in setting a safety valve should be to get the set pressure as far away from the normal system pressure as can be *safely* managed.

The setting of safety valves should be based on the equipment they protect, not the normal steam pressure to which they will be exposed. The difference between system and set pressure is what holds the valve closed. The greater the differential, the less likely the valve is to leak.

Perhaps the biggest area for potential live steam loss is through leaking steam traps.

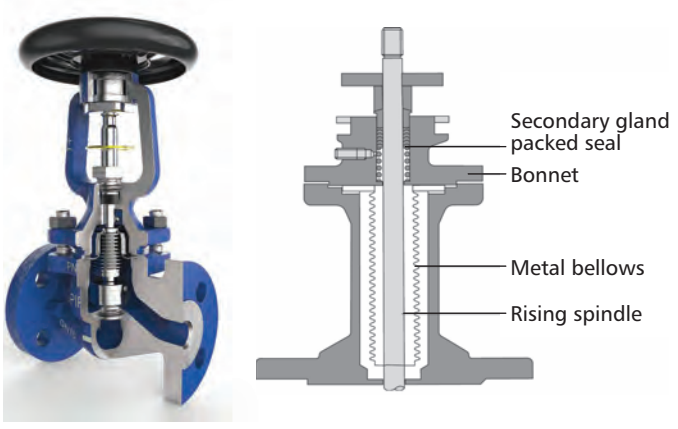



**Figure 8.14** Bellows sealed valve

Any steam trap is prone to leak at some time in its life. This leak can be caused by mechanical damage, by dirt or pipe scale or simple wear of the seating mechanism.

The presence of leaking traps is likely to be obvious with a significant discharge of steam emitting from the condensate receiver and other collecting tank vent pipes. A rise in condensate return system pressure would be another clue.

What is less obvious is where the leak is coming from. The only way to determine this is by regular monitoring of each trap's operation. This can either be an automatic and continuous process using trap monitoring instrumentation or a regular testing programme conducted by an experienced audit engineer.

# 9 Maintenance

## 9.1 What happens if maintenance is neglected?

Steam systems are very tolerant of neglect. Under most circumstances, they will not stop working if not maintained properly, but there are two issues that will be compromised – efficiency and safety.

### 9.1.1 Efficiency

In a steam and condensate system where leaks are not repaired, pumps do not work and insulation is not replaced, the cost of a tonne of steam (see Section 2) will increase. Small steam leaks will become large steam leaks. Repairing a leak will move from tightening connections to replacing the joint, as there is a strong possibility of permanent damage being done due to wiredrawing.

Control valves will eventually fail, leading to more breakdowns and perhaps loss of critical services when needed most.

Fouling on heat transfer surfaces in both the boiler and heat exchangers results in a higher than required flue gas temperature (boiler) and more flash steam being formed (heat exchangers). Both of these conditions represent possible additional energy losses from the system. Other implications could be a reduction in heat output and potential permanent damage if the fouling is sufficiently severe.

A steam trap that fails open will waste steam, but the equipment that it is draining may not be badly affected. Operating costs will rise. A trap fitted on a heat exchanger that fails closed will be more obvious. The output will drop and remedial action will be required.

A condensate pump that stops working means valuable hot water would be lost – a loss that, in many cases, would not be obvious.

### 9.1.2 Safety

There are many components in a steam and condensate system that are there for safety reasons and hence must be maintained.

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Level controls and alarms on the boiler are safety critical items. They must be part of a regular checking routine which is included in the boilerhouse standard operating procedures.

Safety valves that are not regularly checked could be stuck closed. The setting could be wrong, resulting in leaks or a lack of proper protection.

High limit temperature controls could also fail in an open position and so be unable to respond to a temperature overrun.

Some steam traps may be fitted to critical items. A trap fitted to a drain point on the steam distribution system failing closed will not be immediately apparent and could result in the dangerous build-up of condensate in the steam pipe, eventually leading to waterhammer damage.

## 9.2 Regular audits and what to look for

The frequency of an audit programme will depend on the age and general condition of the steam and condensate system. Some items of equipment require a regular mandatory inspection for insurance purposes. A routine walk-around survey should be carried out at least once annually. Table 9.1 lists some of the most important items to look for and their implications for factors such as cost, safety and production efficiency.

## 9.3 Routine maintenance

The boilerhouse is a key area for routine maintenance tasks to take place. All boilerhouses should have standard operating procedures which list and schedule maintenance requirements. This may also apply to major plantrooms.

**Table 9.1** Audit check list

| What to look for | Implications |
|---|---|
| Steam leaks | Leaking steam is dangerous. It is also a waste of energy and water |
| Steam trap operation | A trap which has failed open will waste a lot of valuable steam. One that has failed closed may have safety implications or result in poor heat exchanger output |
| Listen for waterhammer | Waterhammer should not be considered to be inevitable in steam systems. Waterhammer indicates the presence of water in the pipework that has not been effectively removed. It is dangerous to ignore |
| Low points in steam pipework failing to drain properly | Water gathering in the steam system produces wetter steam and can also induce waterhammer |
| Control valves constantly moving | This may be what is known as 'hunting' and can be caused by ineffective condensate removal, an incorrectly sized valve or the wrong controller settings. A constantly moving valve/actuator will have a reduced life |
| Insulation missing or in poor condition | Increased heat loss and production of wetter steam |
| Safety valve settings | A valve set too high is obviously dangerous. One set too close to the normal operating pressure is prone to leakage |
| Condensate leaks | A waste of energy and good quality water. Could also be a sign that internal corrosion is taking place |
| Condensate being discharged to drain | A waste of energy and good quality water |
| Condensate pump receivers overflowing | As above. Pump has perhaps stopped working or is undersized |
| Deposits on valve glands and other leakage points | This could be a sign that boiler water is being carried over into the steam system |
| Condition of pipe supports | A broken support can cause the pipe to sag, so providing a low point for water to gather |
| Products wrongly installed | Upside down or wrong way round valves may not operate correctly |
| Check condition of condensate on first drain point | This will give an indication of whether boiler carryover is occurring |
| Lack of air vents | Improper air removal means poor heat transfer, which in turn means slow warm-up times in heat exchangers |
| Steam meters not reading | Ineffective or incorrect monitoring of steam consumption. Can have an impact on attempts to reduce steam usage |
| Lack of strainers and separators on sensitive pipeline components (valves and meters) | Reduced service life. Inaccurate control or reading |
| No isolation valves (or valves in very poor condition) | Inability to isolate system components for maintenance purposes. Possible safety implications |
| Significant changes to system operation | May have health and safety implications and require any examination schemes or operating procedures to be rewritten and consideration given to retraining |
| Manual processes in the boilerhouse (e.g. blowdown, start-up) | Will not always be done on time and correctly. Consider automating these processes |
| High frequency noise (as opposed to waterhammer bangs) | Pipes or valves undersized, meaning greater pressure drop and internal erosion |

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Steam leaks should be repaired early. They seldom seal themselves and commonly result in more extensive, and expensive, damage. They should be dealt with on a priority basis, depending on the size of the leak and not just ease of access. The same applies to leaks on the condensate system, although these do tend to be repaired as they arise due to the fact that a water leak is a greater nuisance than a steam leak.

Regular checks should be made on traps and pumps. These can be monitored automatically, and remotely. Alternatively, checking can be carried out manually. A reasonable frequency would be every six months.

The purpose of a strainer is to capture dirt and pipe scale which forms inside the steam and condensate system. These need to be cleaned out periodically. Frequency of cleaning will depend on the condition of the pipework and other components. Do not assume that strainers will not need to be cleaned out on new installations. On start-up, a significant amount of scale and rust can be disturbed from inside the pipe, so cleaning strainers should be part of a commissioning routine.

Traditional gland seals on control and isolation valves should be checked, tightened and, if appropriate, lubricated.

Any safety vales which are leaking should be repaired, their setting checked and reset by an approved authority.

Bypasses are frequently installed around control valves and steam trap sets. They are generally installed for emergency reasons, but frequently left fully or partially open afterwards. They can represent a significant loss of steam (on steam traps), poor control (on temperature control valves) or a safety hazard (on pressure reducing valves where their additional capacity may affect safety valve sizing). Because of the frequent abuse of bypasses, their use should be avoided if possible.

## 9.4 Safety issues associated with maintenance

Because steam (and condensate) systems are under pressure, regulations will generally be in force to dictate how they can be safely used. In the UK, the Pressure Systems Safety Regulations (PSSR) 2000 cover all pressure systems. Regulation 12 in particular refers to maintenance requirements and stipulates that the user of an installed system should keep it 'properly maintained in good repair, so as to prevent danger'. These regulations also cover the need for a suitable maintenance programme. They do not specify exactly what needs to be done; that will be dependent on the individual system, but they do require the production of a written scheme of examination. Conducting this examination should highlight any safety issues in the steam and condensate system.

Although steam is not toxic or corrosive, it is still a dangerous hot gas under pressure so has to be treated with care and respect. Protective equipment should be worn when maintaining a steam and condensate system, the minimum items being safety glasses/visor and gloves. Because pipework and components will often be at high level or in places that are difficult to access, safety helmets and harnesses may also be needed.

Isolation valves play a key part in any maintenance programme. Their ability to close easily and completely must be checked before any maintenance is carried out downstream. Some key valves will have lockable handles to ensure that they can only be opened and closed by responsible people. Quarter-turn valves, such as ball and butterfly, can provide quick isolation but they can also be inadvertently opened and closed, so should be suitably 'locked' in the appropriate position.

Any valve which is used to bleed steam into a cold distribution system must be opened slowly, not only to ensure proper warm-up without undue pipe stress but, more importantly, to avoid waterhammer damage. This warm-up period could extend over a period of minutes, or even many hours for a large, extensive steam system. Care must therefore be taken with quarter-turn valves for such applications. A globe or parallel slide valve should be the preferred choice and consideration given to automating such valves to ensure that the start-up process is properly controlled.

Where downstream isolation is critical, double block and bleed valves are now commonly used. This system comprises two isolation valves in series with a bleed arrangement in between so that any escape from the first valve is bled away safely, so making it easier for the second valve to complete the closure. This double block and bleed feature can be built into one component as shown in Figure 9.1.

Valves should only be opened and closed by someone who knows their function and is trained and authorised to operate them.

Even if the isolation valve is providing 100% closure, there will still be pressurised steam in the pipework system. The steam pressure must be allowed to decay, generally achieved by allowing the steam to cool and condense. Once this is done, carrying out repairs to a steam system is a simpler matter than in a liquid-based system, where the cooling down period may take longer and a much greater quantity of liquid may have to be run to drain.

It should be borne in mind that the condensate return system may be under pressure too. It will contain water, so provision has to be made to drain this safely to an appropriate place.

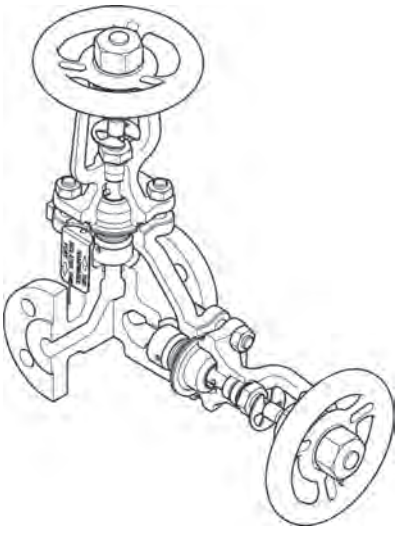


**Figure 9.1** Double block and bleed valve

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