ARI - Application Technology

A Practical Guide
to Steam and Condensate Engineering

Technological concepts for an efficient and effective steam plant

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We have more than 60 years experience ...

... as manufacturers of quality valves – we would like to share our knowledge and experience with you in this Guide.

As a plant operator, planner or builder, it is your responibility to ensure that processes are designed and operated reliably, making efficient use of valuable energy resources.

The aim of this Practical Guide to Steam and Condensate Engineering is to demonstrate possible solution strategies – from steam generation through steam distribution to the extremly important management and return of condensate.

4. Edition

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1.0 Heat Engineering Concepts

1.1 Introduction

Energy – an increasingly valuable resource – is used in a wide range of industries for both heating and cooling.

The heat transfer process can be:

- Direct – Where heat is transferred directly from its source to the substance to be heated. A metal pipe heated by means of a gas flame for the purpose of bending represents a good example. The heat from the gas flame acts directly on the product to be heated.
- Indirect – Where heat energy is transferred to a substance through an intermediate wall. An example is an old fashioned kettle that absorbs heat on a gas cooker. The heat from the gas flame is transferred to the water through the bottom of the kettle and then into the water. In industry, steam may be generated and used to heat water or thermal oil for the manufacturing process.

In industry, steam may be generated and used to heat water or other process fluids for industrial and HVAC applications including sterilisation, blanching, air heating and distilling.

This guide describes the principles of heat transfer by means of steam. The starting point is a boiler installed at a central location and used for steam generation. The steam is supplied to the various heat users and the water returned to the boiler house in the form of condensate

The term 'enthalpy' is used in the following to refer to 'heat' and 'heat content'. Enthalpy is generally defined as the total amount of energy contained in a system. This energy can be present in the form of heat in gases, liquids or steam. However, it also includes chemically bound energy, for example in fuels. Enthalpy additionally describes the amount of energy that is required for a phase change from a liquid to a gaseous state or from solid to liquid.

1.2 What is steam?

Let us imagine an old-fashioned kettle whistling away in the kitchen. We fill it with water, stand it on the cooker and switch on the heat source. After a few minutes, the water begins to boil and steam emerges from the spout. Yet what happens exactly in the water-filled kettle – equivalent to a boiler – from the moment heat is applied to the time when the steam emerges from the spout?

The heat transfer process – brought about, for example, by a gas flame underneath the kettle – begins as soon as the latter is placed on the cooker. Small gas bubbles form on the kettle walls as the temperature of the water rises. After a while, these bubbles drift upwards to the surface and the gases dissolved in the water – one of which is air – are expelled.

Observation 1

The solubility of the gases contained in the water decreases with an increase in water temperature. We will turn our attention to this phenomenon later when we discuss the principle of a deaerator (*refer to Chapter 3.14 Deaerator)*.

Small bubbles now also form on the bottom of the kettle as the water heats up. Although these bubbles likewise rise, they disappear before they reach the water surface. These are not gas bubbles but small steam bubbles that condense as they progress through the colder layers of water. Whereas the temperature on the bottom of the kettle has already reached 100 ºC, in the higher water layers it has not. Moreover, we can observe that after a while the steam bubbles start to rise higher and higher. At the same time, it is clear that the water level rises with the temperature increase in the kettle. It has risen significantly since the moment the steam bubbles started to form.

Observation 2

Water expands as it heats up.

The total volume of the steam bubbles is up to 1700 times larger than that of the water from which they are formed.

The temperature of the complete contents of the kettle reaches 100 ºC at a time „X". From this point on, steam bubbles regularly rise to the surface of the water and escape. The water begins to boil and steam emerges from the spout. If we now measure the temperature of the water in the kettle and that of the escaping steam, the value for both will be 100 ºC.

If we subsequently increase the supply of heat (e.g. by turning up the gas), the quantity of steam bubbles will likewise increase. However, regardless of how much we turn up the heat, the temperature of the water and that of the escaping steam will always remain a constant at 100 ºC.

Observation 3

Under atmospheric conditions, in other words at an absolute pressure of 1013 mbar, water boils at 100 °C. Water with a temperature higher than 100 °C cannot exist under atmospheric conditions. The steam that is formed from water at 100 °C has the same temperature as the water itself.

If the boiling water is heated further, it begins to evaporate. In addition to the sensible heat required for heating, a significantly larger amount of energy must be supplied in order to evaporate the liquid. This 'enthalpy of evaporation' is represented by the symbol h_{fg}. The total enthalpy of the steam that is formed is comprised of the sum of the liquid enthalpy and the enthalpy of evaporation, and is represented by the symbol h_g:

$$\mathbf{h_g = h_f + h_{fg}}$$

As mentioned earlier, water boils at 100 °C under atmospheric conditions (1013 mbar). At a pressure of 1000 mbar (1 bar), water starts to boil at 99.6 ºC. From now on, for the sake of simplicity, we will adhere to the popular definition and assume that water boils at a pressure of 1 bar and 100 °C.

1.3 Boiling and evaporation point of water

Fig. 1-1 shows the gas consumption that is needed to boil and evaporate water.

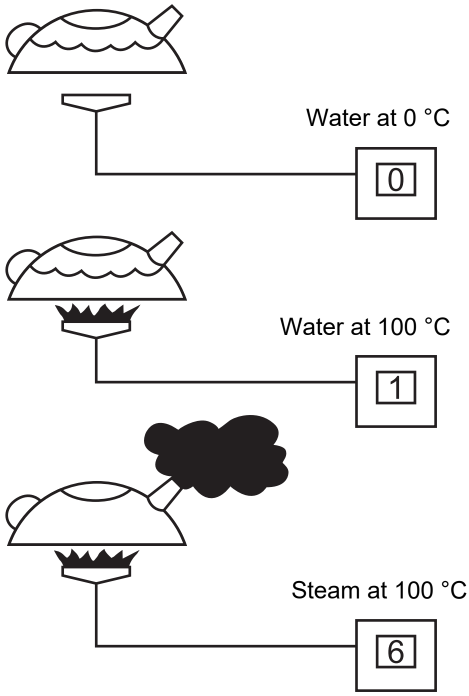

Fig. 1-1: Kettle with gas meter

The gas meter is set to zero at the start of the experiment. After the water has been heated to 100 °C, the meter shows '1 unit of gas' has been consumed. A further six units of gas are required to evaporate all the water, in other words it takes approximately six times as much energy to evaporate water as it does to boil it. Conversely, this means that 1 kg of steam at 100 °C contains about six times as much heat as 1 kg of water at 100 °C. If heat is removed from the steam at a temperature of 100 ºC, condensate with a temperature of 100 ºC is formed. Put another way, the condensing process results in around six times as much heat from 1 kg of steam at 100 °C as is contained in water at the same temperature.

Fig. 1-2 a shows the amount of energy that is needed to heat and evaporate 1 kg of water under atmospheric conditions. The specific heat of water is 4.18 or approx. 4.2 kJ/kg K. 100 x 4.2 = 420 kJ of sensible heat (h_f) is required to heat 1 kg of water from 0 ºC to 100 ºC. The enthalpy of evaporation (h_{fg}) necessary to completely evaporate 1 kg of water at 100 ºC is 2258 kJ/kg and can be taken from the steam tables (refer to Fig. 1-3).

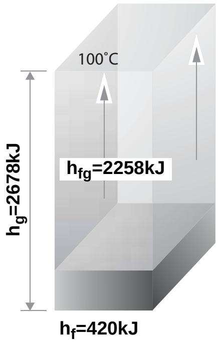

Fig. 1-2 a: Evaporation

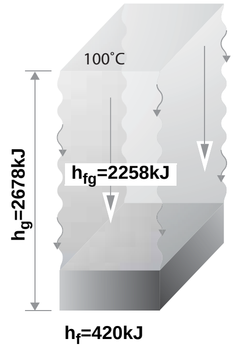

Fig. 1-2 b: Condensation

The total heat required to completely evaporate 1 kg of water from a temperature of 0 °C under atmospheric conditions is ($h_g = h_f + h_{gf}$) 420 + 2258 = 2678 kJ. When heat is transferred from steam to the product or in a heat exchanger, the steam condenses. The heat that is released in the process is known as the latent heat of condensation. If heat is extracted from steam under atmospheric conditions, 2258 kJ/kg of the latent heat of condensation will be consumed while 420 kJ/kg will remain in the 100 ºC condensate as sensible heat (refer to Fig. 1-2 b).

The boiling point is higher in a closed vessel.

The enthalpy of evaporation is equal to the amount of heat that is released when steam condensates. The residual condensate has the same temperature as the steam from which it was formed.

Processes with a steam pressure of 1 bar are rare because steam pressure varies according to the required process temperature.

- Every steam pressure has a saturation temperature.
- Saturated steam has a saturation temperature of 180 ºC at 10 bar, 152 ºC at 5 bar, 234 ºC at 30 bar, etc.

Several other properties of steam are also dependent on the pressure in addition to the saturation temperature, such as the liquid enthalpy, enthalpy of evaporation, and specific volume. These values are listed in the saturated steam tables.

1.4 Steam table for saturated steam

In *Chapter 10.0 Appendix* of this book contains steam tables for saturated and superheated steam. Fig. 1-3 shows an excerpt from the table for saturated steam.

You can find the corresponding state variables for each steam pressure in this table. The meanings of the table columns are explained below.

Steam pressure **P** **bar (abs.)**	**Saturated steam temperature** **T_s** **°C**	**Specific volume** **v_g** **m^3/kg**	**Specific mass** **ρ** **kg/m^3**	**Sensible heat** **h_f** **kJ/kg**	**Total enthalpy** **h_g** **kJ/kg**	**Enthalpie of evaporisation** **h_{fg}** **kJ/kg**
0.25	64.99	6.204	0.1612	271.99	2618.3	2346.4
0.30	69.12	5.229	0.1912	289.30	2625.4	2336.1
0.40	75.89	3.993	0.2504	317.65	2636.9	2319.2
0.45	78.74	3.576	0.2796	329.64	2641.7	2312.0
0.50	81.35	3.240	0.3086	340.56	2646.0	2305.4
0.55	83.74	2.964	0.3374	350.61	2649.9	2299.3
0.60	85.95	2.732	0.3661	359.93	2653.6	2293.6
0.65	88.02	2.535	0.3945	368.62	2656.9	2288.3
0.70	89.96	2.365	0.4229	376.77	2660.1	2283.3
0.75	91.79	2.217	0.4511	384.45	2663.0	2278.6
0.80	93.51	2.087	0.4792	391.72	2665.8	2274.0
0.85	95.15	1.972	0.5071	393.63	2668.4	2269.8
0.90	96.71	1.869	0.5350	405.21	2670.9	2265.6
0.95	98.20	1.777	0.5627	411.49	2673.2	2261.7
1.00	99.63	1.694	0.5904	417.51	2675.4	2257.9
1.5	111.37	1.159	0.8328	467.13	2693.4	2226.2
2.0	120.23	0.8854	1.129	504.70	2706.3	2201.6
2.5	127.43	0.7184	1.392	535.34	2716.4	2181.0
3.0	133.54	0.6056	1.651	561.43	2724.7	2163.2
3.5	138.87	0.5240	1.908	584.27	2731.6	2147.4
4.0	143.62	0.4622	2.163	604.67	2737.6	2133.0
4.5	147.92	0.4138	2.417	623.16	2742.9	2119.7
5.0	151.84	0.3747	2.669	640.12	2747.5	2107.4
5.5	155.46	0.3426	2.920	655.78	2451.7	2095.9
6.0	158.84	0.3155	3.170	670.42	2755.5	2085.0
6.5	161.99	0.2925	3.419	684.12	2758.8	2074.0
7.0	164.96	0.2727	6.667	697.06	2762.0	2064.9
7.5	167.75	0.2554	3.915	709.29	2764.8	2055.5
8.0	170.41	0.2403	4.162	720.94	2767.5	2046.5
8.5	172.94	0.2268	4.409	732.02	2769.9	2037.9
9.0	175.36	0.2148	4.655	742.64	2772.1	2029.5
9.5	177.66	0.2040	4.901	752.81	2774.2	2021.4
10.0	179.88	0.1943	5.147	762.61	2776.2	2013.6

Fig. 1-3: Excerpt from the steam table

Column 1: Steam pressure (P)

The steam pressure specified in the steam table is the absolute pressure in bar. This is extremely important. When we speak of pressure, we are usually referring to gauge pressure, in other words the value that is read off on a pressure gauge. However, process engineers prefer to use absolute pressure for calculation purposes. This discrepancy often leads to misunderstandings in practice, sometimes with undesirable consequences.

When the SI system was first introduced, it was agreed that only absolute pressure should be used. All pressure measuring devices (other than barometers) in a process plant show gauge pressure. When reference is made in this book to "pressure", we always mean absolute pressure. Under standard conditions (sea level), the barometric pressure is 1 bar. Gauge pressure is normally indicated in barg. This guide uses bar a, or bar absolute

Column 2: Saturated steam temperature (T_s)

The saturated steam temperature is shown next to the steam pressure. The temperature at this pressure is identical to the boiling point of the water and is thus the temperature of the saturated steam. This same temperature column is also used to determine the temperature of the condensed steam.

Saturated steam at 8 bar has a temperature of 170.4 ºC. The condensate from steam at a pressure of 5 bar therefore has a temperature of 152 ºC.

Column 3: Specific volume (v_g)

The specific volume is indicated in m^3/kg. It decreases as the steam pressure increases. Steam at a pressure of 1 bar has a specific volume of 1.694 m^3/kg. Put another way, 1 dm^3 (1 litre or 1 kg) of evaporated water expands to a volume 1694 times its original form. The specific volume at a pressure of 10 bar is ($1/\rho$ =) 0.194 m^3/kg, in other words 194 times the volume of water. The specific volume is required to determine the diameter of steam and condensate pipes.

Column 4: Specific mass (ρ = rho)

The specific mass (also referred to as density) is expressed in kg/m^3. It indicates the amount of steam that will fit into 1 m^3. The specific mass increases as the pressure increases. At a steam pressure of 6 bar, 1 m^3 has a mass of 3.17 kg. This value increases to 5.15 kg at 10 bar and to more than 12.5 kg at 25 bar.

Column 5: Sensible heat or Liquid Enthalpy (h_f)

The sensible heat is expressed in kJ/kg. This column shows the amount of heat that is needed to boil 1 kg of water at a defined pressure or the amount of heat remaining in the condensate after 1 kg of steam has condensed at the same pressure. The sensible heat is 417.5 kJ/kg at a pressure of 1 bar, 762.6 kJ/kg at 10 bar and 1087 kJ/kg at 40 bar. The sensible heat increases as the steam pressure increases and accounts for an ever larger proportion of the steam's total enthalpy. The higher the steam pressure, in other words, the more residual heat there is in the condensate.

Column 6: Total enthalpy (h_g)

The enthalpy is expressed in kJ/kg. This column shows the steam's total enthalpy. It is additionally clear that the enthalpy increases up to a pressure of 31 bar and decreases thereafter. The enthalpy is 2801 kJ/kg at a pressure of 25 bar but only 2767 kJ/kg at 75 bar.

Column 7: Enthalpy of evaporation (condensation) (h_{fg})

The enthalpy of evaporation (condensation) is expressed in kJ/kg. This column shows the amount of heat that is needed to completely evaporate 1 kg of boiling water at a defined pressure. Conversely, it also specifies the amount of heat that is released if the (saturated) steam is completely condensed at a defined pressure.

h_{fg} is 2258 kJ/kg at a pressure of 1 bar, 1984 kJ/kg at 12 bar and only 1443 kJ/kg at 80 bar. The enthalpy of evaporation (condensation) decreases as the pressure increases.

Rule:

The amount of heat that is needed to completely evaporate boiling water decreases as the pressure increases. Conversely, less heat is released if the saturated steam condenses at the same pressure.

Examples from the steam table

Example 1: Boiling and evaporation (Fig. 1-4 a)

To evaporate water at a pressure of 7 bar, the water must be boiled at a temperature of 165 °C. The enthalpy (h_f) in this case is 697 kJ/kg. The amount of heat (h_{fg}) that must be supplied in order to evaporate the water is 2065 kJ/kg. The total enthalpy (h_g) of the steam that is produced is thus:
697 + 2065 = 2762 kJ/kg.

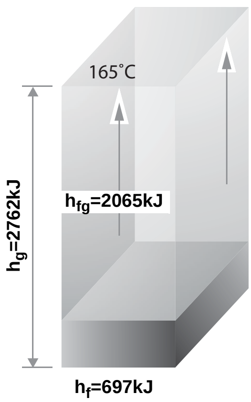

Fig. 1-4 a: Boiling and evaporation

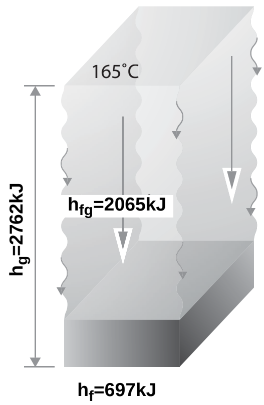

Fig. 1-4 b: Condensation

Example 2: Condensation

The bar graph for condensation (Fig. 1-4 b) reveals that 2065 kJ/kg of heat per kg of steam is released when steam condensates at a pressure of 5 bar (165 °C). The residual condensate has a temperature of 165 °C and an enthalpy h_f = 697 kJ/kg.

Example 3: Heat exchanger

How much heat is required to heat 8 m^3 (8000 kg) of water from 10 °C to 85 °C? How much steam is needed for this purpose if the steam pressure in the heat exchanger is 7 bar? Water has a specific heat of 4.2 kJ/kg K. The amount of heat required is calculated as follows: mass x temperature rise x specific heat = 8000 x (85 - 10) x 4.2 = 2520000 kJ. At a steam pressure of 7 bar the latent heat of condensation is 2065 kJ/kg. The amount of steam necessary to heat the water is 2520000/2065 = 1220 kg. The temperature of the condensate that must be drained off from the steam system by the steam trap is 165 °C.

Example 4: Steam boiler

If feedwater with a temperature of 105 °C (h_f = 440 kJ/kg) is supplied to a steam boiler operating at 12 bar (T = 188 °C and h_f = 798 kJ/kg), this water must first be preheated from 105 °C to 188 °C before it begins to evaporate. The following heat is required to preheat the feedwater: 798 - 440 = 358 kJ/kg. A further h_{fg} = 1984 kJ/kg (from steam tables) of heat must then be supplied in order to evaporate the water. The steam produced in this way has a total enthalpy of 798 + 1984 = 2782 kJ/kg.

1.5 Superheated steam

Superheated steam is steam with a temperature and enthalpy higher than the values specified for this pressure in the saturated steam table. Example: Steam at 10 bar with a temperature of 200 ºC is superheated steam (by comparison, the saturated steam temperature is 180 °C).

Superheated steam has a much lower heat transfer coefficient than saturated steam. If superheated steam is used to transfer heat, e.g. in a heat exchanger, part of the heated surface is used to cool the steam down to saturated steam temperature. The amount of heat that is exchanged during this saturation process is small, yet it occupies a relatively large proportion of the heated surface. The most important step in the heat transfer process takes place during condensation. There is a persistent and widespread misconception that a higher product temperature can be achieved with superheated steam than with saturated steam at the same pressure. Superheated steam is principally used to drive steam turbines (for instance for power generation) and is only employed for heating in the production process if no saturated steam is available. The specific volume of superheated steam is considerably higher than that of saturated steam at the same pressure (compare the specific volume v_g of superheated steam and saturated steam). In addition to various technical and design modifications, it is therefore necessary to take account of the lower absorptivities of the pipe that is installed when changing over from saturated steam to superheated steam.

Superheating with flue gases

The superheating process takes place in a separate tube bank that is mounted in the steam boiler's flue gas flow. Heat continues to be supplied to the saturated steam at a constant pressure in this tube bank. The temperature and enthalpy of the steam are increased.

Superheating by reducing the steam pressure

Saturated steam with a pressure of less than 31 bar can be superheated by reducing the pressure. No work is done and no heat supplied or dissipated as a result of a reduction in steam pressure and the enthalpy remains constant. This process is termed 'adiabatic' as no work is done.

1.6 The h-p-t diagram

Why is the h-p-t diagram mentioned in this book in addition to the steam table?

The h-p-t diagram (Fig. 1-5) has proven to be an invaluable instrument in practice for determining heat engineering concepts such as evaporation, condensation, flash steam formation, superheating, steam moisture content during pressure reduction, etc. It enables a few of steam's thermal properties that are also included in the steam tables to be read off directly and very simply.

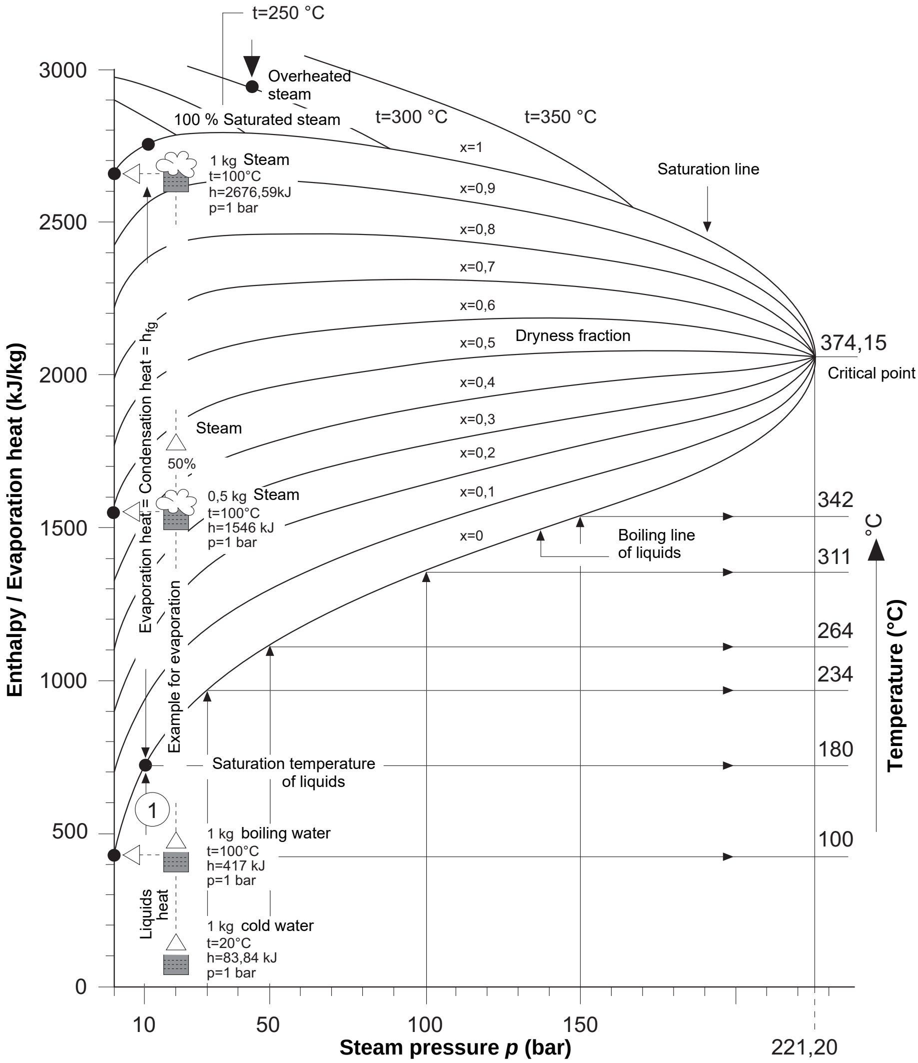

Fig. 1-5: h-p-t diagram

The diagram shows the steam pressure on the horizontal axis, the temperature on the right hand vertical axis and the enthalpy on the left-hand vertical axis. The bottom curve is the water saturation curve. For example, the axis on the right reveals that the saturation point at 50 bar is 264 ºC while the axis on the left indicates the liquid enthalpy as 1155 kJ/kg.

The top curve is the steam saturation curve. The area between the water saturation curve and the steam saturation curve is referred to as the evaporation and condensation or coexistence region, where steam and condensate / water phases coexist. This region also contains several curves (x = 0.1 to x = 0.9) that will be explained later (x = mass ratio of steam / water). The area below the steam saturation curve is the condensate / water region and the area above it the superheated region.

The point at which the saturation curve and the saturation curve meet is referred to as the critical point. The pressure at this point is 221.2 bar and the temperature 374.15 ºC. The enthalpy of evaporation at the critical point is zero. No water can exist above this state and the liquid and vapour phases are not distinguishable.

Examples:

The water saturation curve shows that the liquid has an enthalpy of 760 kJ/kg at a pressure of 10 bar. If 200 kJ of heat is supplied to this liquid, the total enthalpy is 960 kJ. Let us follow a horizontal line from 960 kJ on the left-hand axis to the 10 bar line in the evaporation and condensation region. We discover that this point is located at x = 0.1. After 200 kJ has been supplied to the condensate at 180 °C, 10 % of this condensate will have evaporated. If 840 kJ were to be supplied instead, the total enthalpy would be 1600 kJ and 42 % of the condensate will have evaporated. Conversely, if 400 kJ of heat were to be extracted from saturated steam at a pressure of 10 bar with an enthalpy of 2776 kJ/kg, the total enthalpy would be 2376 kJ. Let us now follow a horizontal line from 2376 kJ on the left-hand axis to 10 bar. We can see from the result that 20 % of the steam has condensed.

Pressure reduction

What happens in a high-pressure steam system if the steam pressure upstream of a heat exchanger is reduced by means of a control valve or if the pressure in a low-pressure steam system is controlled with pressure reducing valves?

If the pressure of the saturated steam is reduced using a control or pressure reducing valve, the pressure changes but the enthalpy of the steam remains constant because no work has been done and no heat exchanged. This reduction can be represented in the h-p-t diagram by drawing a horizontal line leftward from the point at which the saturation curve intersects the pressure curve up to the point at which this line intersects the required pressure.

Two examples are described in the diagram: in the first, steam is reduced from 20 to 10 bar while in the second, it is reduced form 150 to 60 bar (an extreme value is used for the sake of clarity). In the first instance, the point of intersection of the 10 bar line is clearly located in the superheated region. The reduced steam is thus superheated.

In the second example, the horizontal line intersects the 60 bar pressure curve in the evaporation and condensation region. Part of the steam will condensate because the enthalpy of saturated steam at 150 bar is 2615 kJ/kg, which is lower than the enthalpy of steam at 60 bar, namely 2785 kJ/kg. The intersection occurs on the line x = 0.9, in other words 10 % of the steam condenses. The steam is wet.

If saturated steam is reduced from 60 to 10 bar, the situation is slightly more complicated. Initially, the horizontal line passes through the evaporation and condensation region and condensation takes place. Later, however, the line reaches the superheated region and part of the condensate that formed previously now evaporates again.

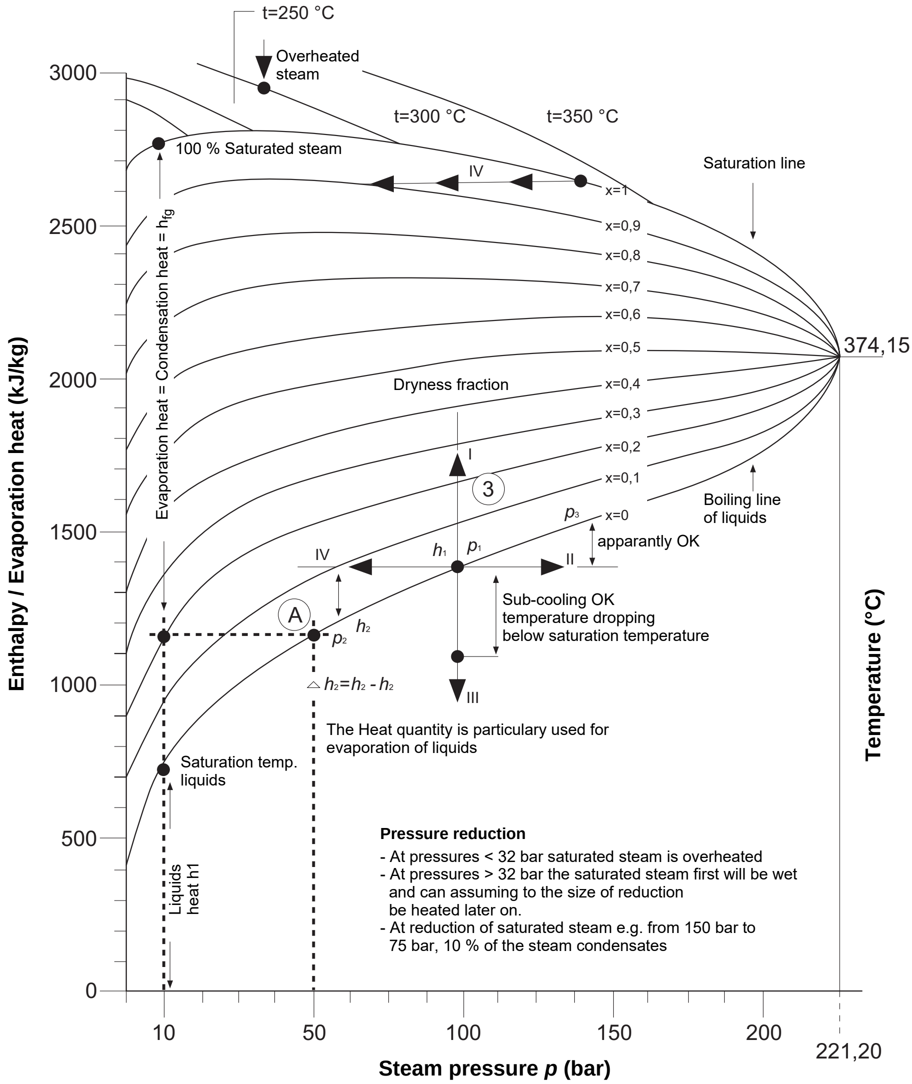

Fig. 1-6: Example in the h-p-t diagram

Condensate flashing

A point A is specified at an arbitrary point on the saturation curve in Fig. 1-6. If the pressure is reduced starting at A, for instance from 50 to 10 bar, the horizontal line runs leftward through the evaporation and condensation region up to the 10 bar mark, where it intersects the curve x = 0.2, in other words 20 % of the condensate has evaporated again.

Wet steam

Strictly speaking, there is no such thing as wet steam - there is either steam or condensate. The density of condensate is higher than that of steam. In a horizontal pipe, condensate drops directly to the bottom of the pipe. When we speak of wet steam, we mean steam in which condensate is entrained as a result of heat losses during transport. The denser condensate is entrained due to the high steam velocity. Water hammer often occurs as a direct consequence of this entrained condensate (refer to *Chapter 5.0 Pipe Drainage*).

It is not unusual for boiler water to be entrained in the steam. There are several possible reasons for this: the steam boiler could be overloaded, the water level in the boiler could be too high or the TDS concentration of the boiler water could likewise be too high. In this case, steam formation in the boiler takes place at such a high rate that boiler water is entrained in the steam when it breaks through the water surface.

White deposits at leakage points on pipe components are an indication that the boiler water is entrained with the steam.

1.7 Temperature of a steam-air mixture

Dalton's Law predicts how a steam-air mixture behaves. Steam and air each have their own partial pressure. The sum of the partial pressures is equal to the total pressure of the mixture. Let us assume that the total pressure in a vessel is 4 bar and that the mixture consists of 75 % steam and 25 % air. In this example, the steam would have a partial pressure of 0.75 x 4 = 3 bar, while the partial pressure of the air would be 0.25 x 4 = 1 bar. The mixture temperature corresponds to the saturated steam temperature for 3 bar, specifically 133.5 ºC. Saturated steam without any air would have a temperature of 143.6 ºC at 4 bar. The table in Fig. 1-7 shows the mixture temperature for various mixing ratios and steam pressures. This information is essential for processes such as sterilisers and autoclaves.

Saturated steam		**Mixture temperature for volume % air**		
p bar	**t ºC**	**10 %**	**20 %**	**30 %**
2	**120.2**	116.7	113.0	110.0
4	**143.6**	140.0	135.5	131.1
6	**158.8**	154.5	150.3	145.1
8	**170.4**	165.9	161.3	165.9
10	**179.9**	175.4	170.4	165.0

Fig. 1-7: Influence of air in steam on the mixture temperature

1.8 Energy costs per tonne of steam

We would now like to estimate the energy costs in the following with the help of an example.

Steam boiler for 10 bar steam (h_f = 2776 kJ/kg)
Efficiency of the boiler plant: 90 %

Condensate return: 100 % (T = 100 °C, h_f = 420 kJ/kg)
Calorific value of natural gas: 31.65 MJ/m^3

The gas consumption for one tonne of steam is therefore calculated as follows:

$$\frac{2776 - 420}{31.65 \times 0.9} = 82.7\ m^3$$

If only 50 % of the condensate is returned, the consumption is increased to:

$$\frac{2776 - 0.5 \times 420}{31.65 \times 0.9} = 90.1\ m^3$$

The steam costs are thus 8.9 % higher than if the entire condensate is returned. The enthalpy of the cold make-up water is disregarded in this calculation for the sake of simplicity. No account is taken of any of the incidental costs for water (water costs, effluent treatment costs, depreciation, etc.). If no condensate is returned, then 97.6 m^3 of natural gas is necessary to produce one tonne of 10 bar steam.

Warning!
It is advisable to take a critical view of fuel consumption calculations. Consultants will often base their calculation on a deaerator temperature of 105 °C, regardless of the amount of condensate that is actually returned. The calculation for a 10 bar steam boiler with an efficiency of 90 % is as follows:

$$\frac{2776 - 440}{31.65 \times 0.9} = 80.2\ m^3 \text{ of natural gas}$$

In reality, this calculation should be based on 50 % condensate return and a feedwater temperature of 10 °C, for example.

In this case, the gas consumption would be:

$$\frac{2776 - (0.5 \times 420) - (0.5 \times 42)}{31.65 \times 0.9} = 89.3\ m^3 \text{ of natural gas per tonne of steam}$$

Condensate

If one tonne of condensate with a temperature of 100 °C is not returned, the energy loss is as follows, assuming a boiler efficiency of 90 %:

$$\frac{420}{31.65 \times 0.9} = 14.7\ m^3 \text{ of natural gas}$$

The energy of the condensate from 5 bar steam (152 °C, h_f = 640 kJ/kg) is equivalent to 22.5 m^3 of natural gas for a boiler efficiency of 90 %.

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2.0 Boiler Plants

2.1 Low-pressure boiler plant

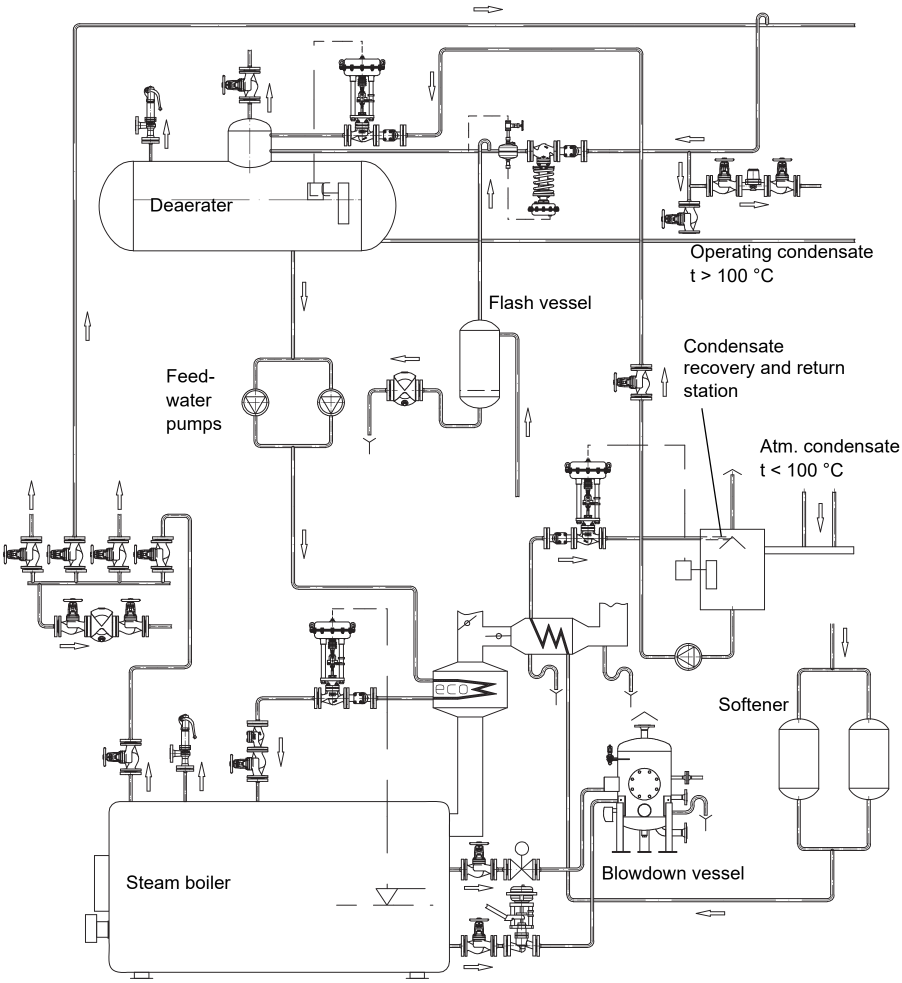


Fig. 2-1: Example of a low-pressure boiler plant

The low-pressure boiler plant described here is installed in hundreds of boiler houses all over Europe. The components illustrated in the diagram are explained below in brief. The details of their functions will be discussed in the next few chapters.

Steam boiler

Shell boilers (or fire tube boilers) are normally used for steam pressures up to 25 bar and capacities up to 25 t/h (Fig. 2-2). This boiler type has a cylindrical outer shell containing one or two furnace tubes (or combustion chambers), including a burner, reversal chambers and fire tubes. The fire tubes and the reversal chambers are surrounded by boiler. The burner is connected to the front of the corrugated furnace tube, where the combustion process takes place. The generated steam exits the boiler at a saturation temperature corresponding to the steam pressure.

The steam can be superheated if necessary. The superheater required for this purpose is located in the front reversal chamber. It heats the steam above the saturation temperature while maintaining a constant pressure.

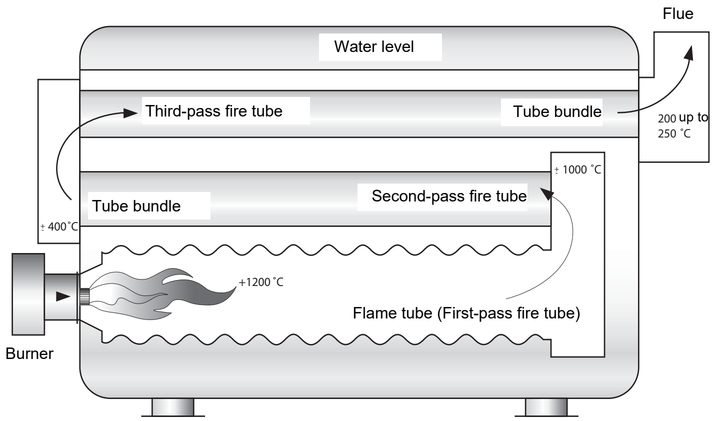


Fig. 2-2: Three-pass fire tube boiler

DIN EN 12953 describes the product standards for shell-type boiler plants.

Deaerator

The purpose of the deaerator is to remove oxygen and carbon dioxide from the feedwater. By largely eliminating these gases from the feedwater, corrosion can be prevented in the boiler as well the downstream steam and condensate system. A pressure of 1.2 bar (T = 105 °C) is typically maintained in the deaerator by steam injection. Condensate returned from the plant is directed to the deaerator, any losses are made up by softened water. The make-up water is sprayed into the dome section of the deaerator together with atmospheric condensate, thereby releasing the gases dissolved in the water, such as oxygen and carbon dioxide. These gases are discharged into the atmosphere with a small amount of steam. Refer to *Chapter 3.14 Deaerator*.

Economiser

The economiser (feedwater preheater) is usually installed in the chimney and used to recover residual heat from flue gases. This is well worth the effort if we remember that flue gases exit from a boiler with 10 bar steam pressure at a temperature of approx. 220 °C. The residual heat from the flue gases can be used to heat feedwater, which exits the deaerator at 105 °C, to a temperature of 130 °C before it enters the steam boiler. Between 4 and 5 % of the gas consumption can be saved with a feedwater preheater.

Continuous feedwater level control

In order to ensure that the correct amount of feedwater is supplied via the economiser for a particular flue gas flow rate, the boiler should be equipped with a continuous water level control system.

Flue gas condenser

Flue gas condensers are installed in the chimney downstream of the economiser. In addition to the sensible heat, they also recover the energy from vapours by cooling them to below their saturation temperature, releasing the enthalpy of evaporation. This heat is ideal for preheating the feedwater supplied by the water treatment plant. Flue gas condensers require a sufficient amount of cold water to work efficiently.

Returned condensate

The condensate that is formed during the heat exchange process on the plant, but has not come into contact with the atmosphere (or other contaminants) does not need to be atomised in the deaerator, though it is usually sparged into it below the water level.

Condensate vessel

Condensate that has come into contact with the atmosphere and therefore absorbed oxygen – and possibly other atmospheric gases as well – is generally supplied to the deaerator via a condensate collection vessel. Cold, softened feedwater is likewise supplied to this vessel. The atmospheric condensate and the softened feedwater are sprayed in the steam space of the deaerator and heated to a temperature of approx. 105 °C.

Softener

The softener removes hardness ions such as calcium and magnesium from the feedwater. Hardness is the cause of scale on the flame and fire tubes. Scale, in turn, reduces the rate of heat transfer and these deposits can ultimately lead to overheating of the fire tube material, serious leakages, and even explosion.

Continuous (or TDS) blowdown valve

Impurities like chloride and sulphate get into the boiler via the feedwater. The steam that exits from the boiler, on the other hand, is clean. The concentration of impurities in the boiler water increases as a result of the evaporation process: the water is said to be "concentrated". To prevent concentration from exceeding an acceptable level, an amount of water is drained from the boiler by means of the blowdown valve. The blowdown process can be controlled automatically on the basis of conductivity.

Blowdown flash vessel

Blowdown water has a energy content which is lost if it is dumped into the sewage system is high. Moreover, the sewage system could be damaged owing to the high discharge temperature, it may also be illegal The pressurised blowndown water should be directed to a flash vessel, and the (clean) flash steam used in the deaerator. Refer to *Chapter 3.2 Bottom or Intermittent and continuous blowdown valves*.

The heat in the condensate may also be used, via a heat exchanger, to heat the incoming feedwater, see Fig. 2-1.

Bottom blowdown valve

Chemical sludge and dirt particles can accumulate on the bottom of a fire tube boiler. The bottom blowdown valve is normally opened fully once a day for a short time to allow this dirt to be drained off. It is not worth recovering the heat from this drainage. The flash steam that is produced during the manual blowdown process is discharged into the atmosphere. The remaining blown down water is discharged into a blowdown vessel before overflowing into the sewage system. Manual blowdown valves must not be used while a water tube boiler is operating because they will interfere with the circulation of the boiler water. The same applies if the blowdown valve is connected to a fire tube boiler with water tubes in the reversal chamber.

2.2 Combined heat and power boiler house

Water tube boilers are used for high pressures and high steam outputs. Fig. 2-3 is a process flow diagram of a steam generator that works with combined heat and power.

It includes a few details from Fig. 2-1. The water tube boiler, the back pressure turbine the alternator and the feedwater treatment system have been added. These parts of the plant are explained in the next section.

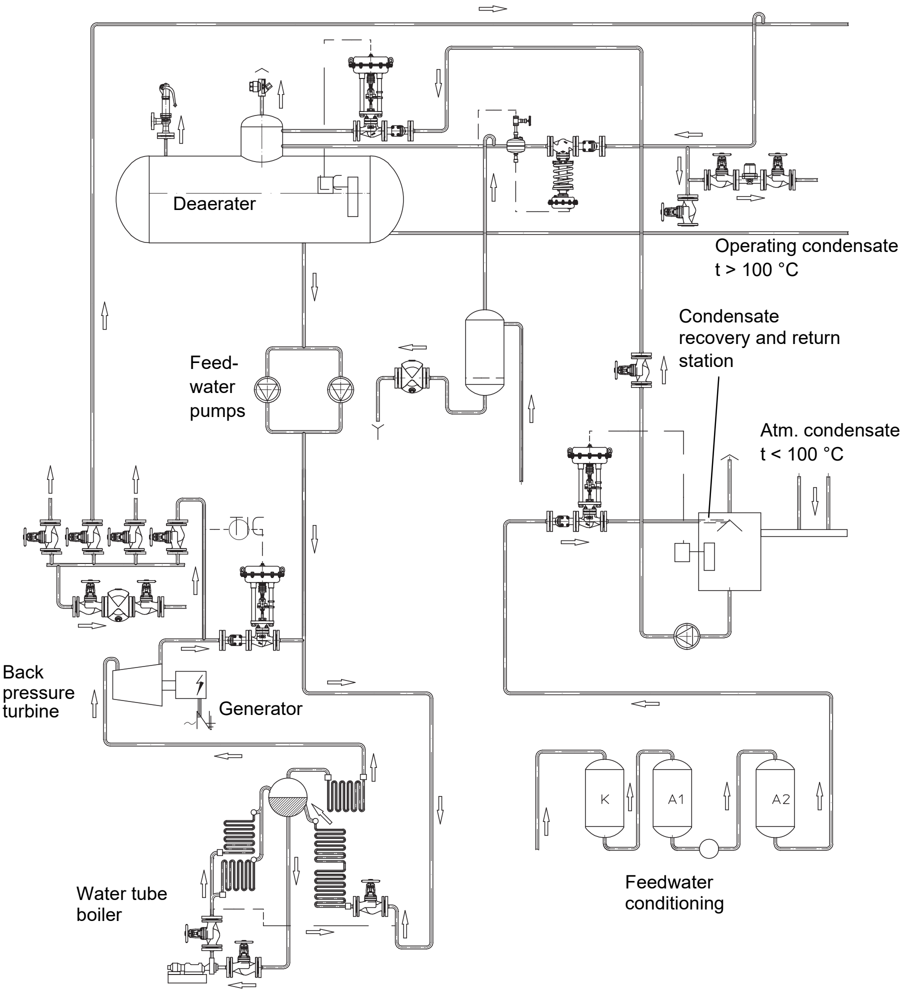


Fig. 2-3: Steam generator with combined heat and power

Water tube boiler

The water in a water tube boiler is contained in the tubes and combustion takes place in a furnace comprised of tube banks and tube walls. High-pressure steam boilers nearly always have a superheater because the steam is used to generate electricity using a turbine and alternator. Water tube boilers can have the following layouts:

- **Natural circulation boiler:**
  Water is pumped into the deaerator by means of a feed pump. The water circulates by convection between the top and bottom drums.
- **Forced circulation boiler:**
  The water is circulated by a pump.
- **Once-through boiler:**
  The water is pumped through the piping system in one direction.

The steam boiler in Fig. 2-3 shows a high-pressure boiler with forced circulation. A circulating pump connected to the drum (steam / water vessel) on the suction side circulates the water over the evaporator heating surfaces. The amount of water circulated through the evaporator heating surfaces is about six times as high as the generated quantity of steam. The steam bubbles that are formed in the evaporator are separated from the water in the drum, e.g. by means of cyclone separators. The steam collects in the drum over the surface of the water, which is continuously circulated to the evaporator. The economiser is mounted in the steam boiler, as indicated in the diagram. This is understandable in view of the fact that the flue gas temperature ahead of the economiser is relatively high (owing to the high steam pressure). In a 100 bar boiler it would be 350 ºC.

In order to extract as much heat as possible from the flue gases, the economiser installed here should be considerably larger than with a low-pressure boiler. In the same way as with low-pressure boilers (Fig. 2-1), an economiser is used to reduce the temperature of the flue gases to around 125 ºC.

Fig. 2-4 shows a corner tube boiler with natural circulation.
The furnace is comprised of water tubes that are welded on both sides with steel strips approx. 15 mm apart to obtain a gas-tight wall, also known as a membrane wall.

The tubes are connected to collection drums at the top and bottom. The tubes inside the membrane walls are sometimes referred to as risers, while the lower collection vessels (or drums) are fed from the drum via downcomers. The downcomer tubes are outside the heated part of the boiler. The upper collection vessels are connected to the drum.

The natural circulation in the riser tubes is maintained due to convection. tThe steam is separated from the boiler water in the drum. The saturated steam that flows from the drum to the superheater should not contain any water. The water and steam are separated by means of vertical baffles inside the drum. Other types of water tube boiler have cyclone separators mounted in the drums to dry the steam. The saturated steam is further heated above the saturation temperature in the superheater (refer to *Chapter 1.0 Heat Engineering Concepts*).

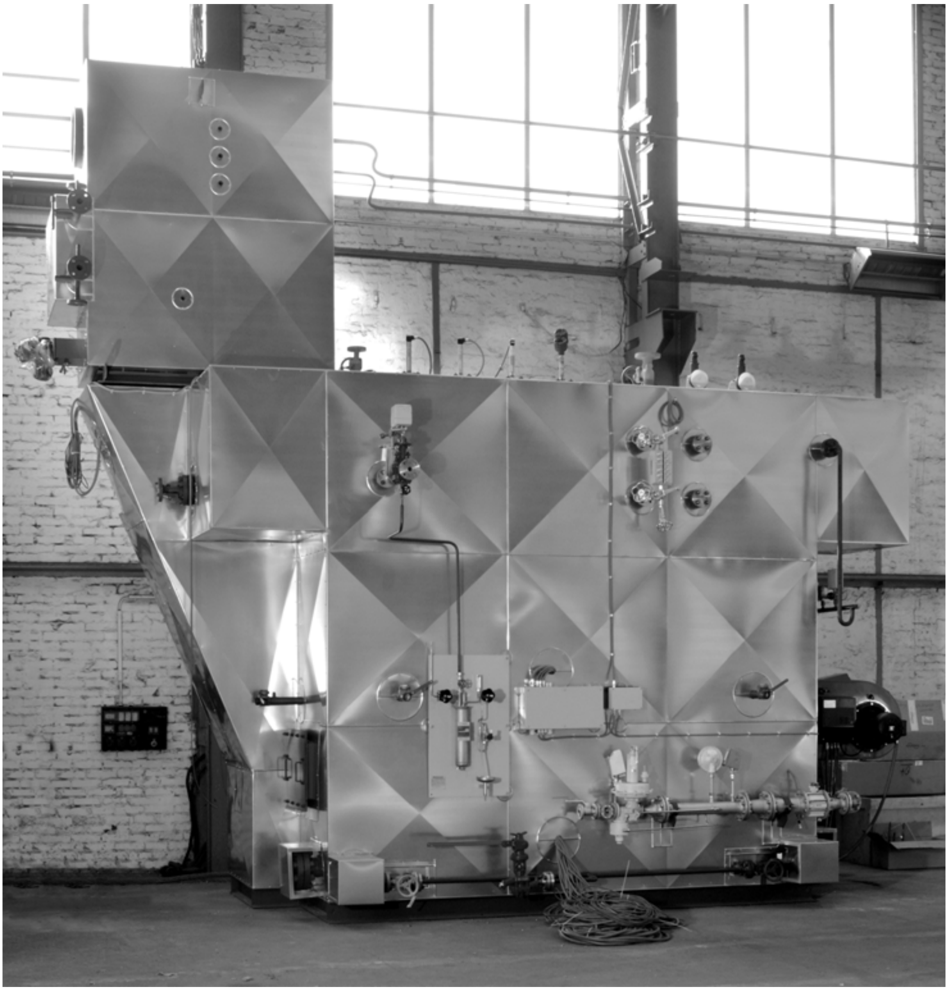


Fig. 2-4: Corner tube boiler

DIN EN 12952 describes the product standards for water tube boiler plants.

Back pressure turbine plants

Fig. 2-3 shows a back pressure turbine with a generator as part of a plant.

High-pressure superheated steam is generated in the water tube boiler. The superheated steam is used to drive the turbine generator. After the uncondensed steam has been utilised in the back pressure turbine to generate electricity, it can also be used for other processes. The turbine exhaust steam is usually still slightly superheated.

Steam turbo-alternator

This book can only provide a rough outline of the operating principle and applications. The superheated steam from the boiler is fed directly to the turbine. It is expanded with a nozzle so that pressure energy is converted to kinetic or velocity energy. The kinetic energy contained is transferred to the turbine blades, which in turn transmit their energy to the turbine wheel, through the shaft alternator.

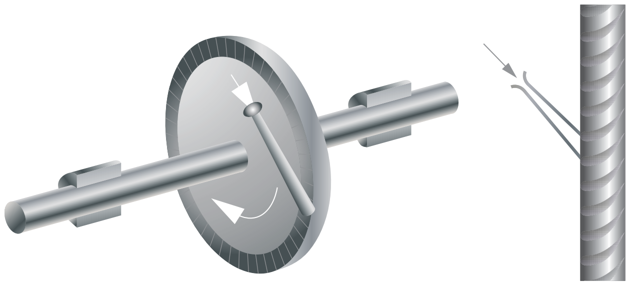


Fig. 2-5: Principal of a steam turbine

Not all the energy contained in the steam can be transferred to the turbine via a single turbine wheel, so several turbine wheels are mounted in series. Stationary blades are inserted in the casing to guide the steam exiting from one turbine wheel to the next turbine wheel. As the steam flows through the turbine, the pressure drops and the specific volume of the steam increases. For this reason, the turbine wheel diameters will increase from the inlet to the exhaust end of the turbine.

2.3 Turbine types

There are two main types of turbine that are used for different purposes (refer to Fig. 2-6).

Back pressure turbines

After the steam has driven the turbine, it is reused as process steam. The pressure of the exiting steam is controlled to suit the requirements of the particular process. This principle is referred to as combined heat and power. In addition to steam for process applications, the generator also generates electricity. The average overall efficiency is 85 %. In some applications, steam is extracted at various points on the turbine housing for higher pressure applications.

Condensing turbines

The exiting steam condenses in a condenser under vacuum. As a result of the vacuum condenser, these turbines work with a larger enthalpy difference (hence improved Rankine efficiency) between the inlet and exhaust than back pressure turbines and are capable of generating more electricity per tonne of steam. Condensing turbines are normally used in power stations. Power stations with condensing turbines have an overall efficiency of between 40 and 50 %.

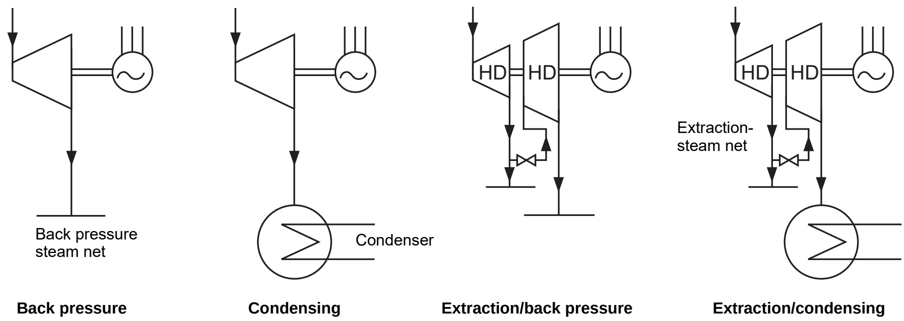


Fig. 2-6: Turbine types

Demineralised Water

Demineralised water is the only water suitable for high-pressure steam boilers. The steam pressures and temperatures occurring in this type of boiler are so high that a simple softening plant cannot prepare water of good enough quality. In addition to hardness constituents such as calcium and magnesium, various other impurities like sulphates, silicates, chlorides and sodium have to be removed. The carbon dioxide (CO_2) that is bound to water is likewise removed in the demineralisation process by means of a CO_2 scavenger.

2.4 CHP Plant with a Gas Turbine

To round off the most popular solutions for boiler houses, Fig. 2-8 shows a combined heat and power plant with a gas turbine, a waste heat boiler with auxiliary firing and a back pressure turbine.

A gas turbine plant comprises three main components (refer to Fig. 2-7):

- A compressor to compress the combustion air.
- One or more combustion chambers in which combustion air and fuel (natural gas, refinery gas, heating oil) are intensively mixed and ignited.
- A gas turbine in which hot flue gases expand as they exit the combustion chamber at 900 to 1200 ºC.

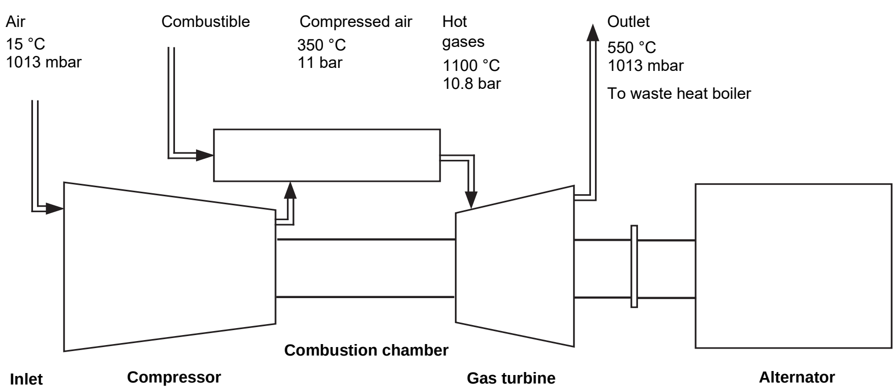


Fig. 2-7: Principle of a gas turbine

The compressor and the turbine are mounted on a common shaft; and they drive an alternator which generates electricity.

The flue gases exit the turbine at a temperature of approx. 500 ºC, then enter a waste heat boiler. Superheated steam is generated in this boiler with the residual heat from the flue gases. Gas turbines require a large amount of excess air (typically 4 times the required amount) owing to the temperature limitation of the turbine blades. In theory, stoichiometric combustion of 1 m^3 of natural gas requires 8.5 m^3 of combustion air. 4 x 8.5 m^3 = 34 m^3 of combustion air have to be supplied to a gas turbine per m^3 of natural gas.

At the end of the combustion process, the flue gases still contains sufficient oxygen to allow it to be used for further firing in the waste heat boiler.

The waste heat boiler generally consists of a high-pressure boiler with an economiser and a superheater. All waste heat boilers are designed with a forced circulation system. The superheated steam that is generated is fed to a back pressure turbine, which is connected to an alternator to generate additional electricity, and the pass-out (or exhaust) steam from the turbine may be used on the process.

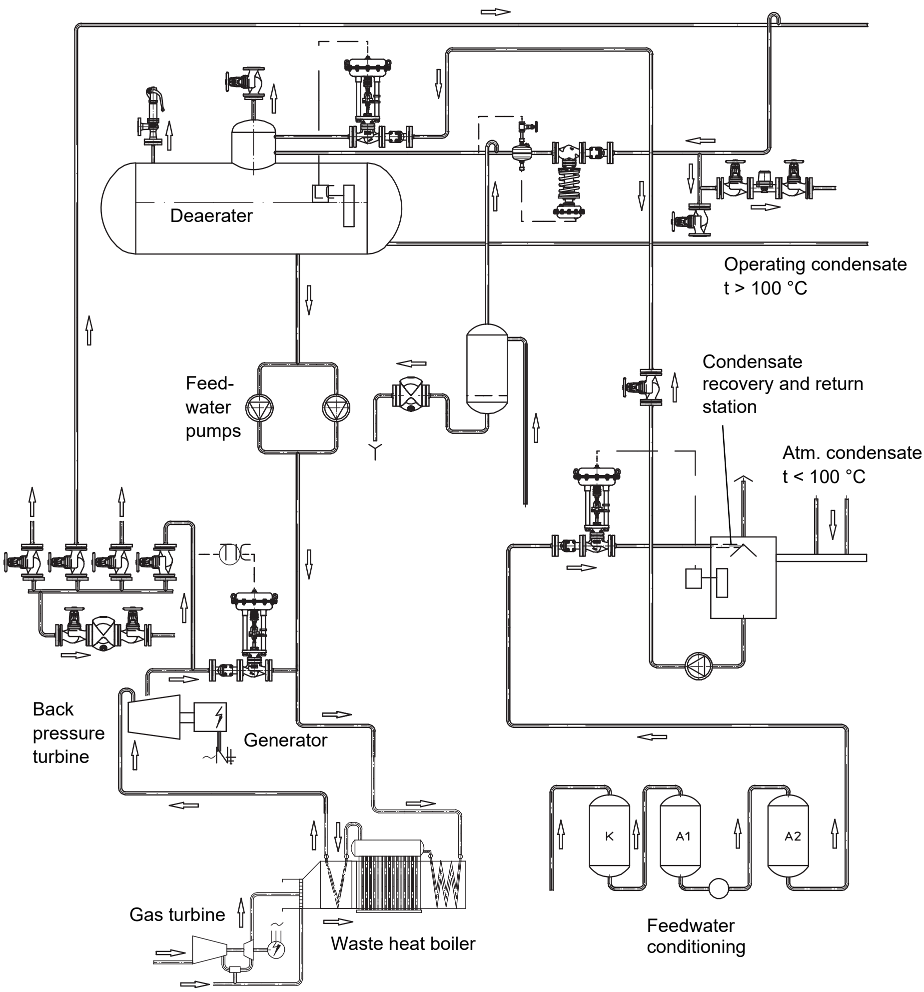


Fig. 2-8: Example of a combined heat and power plant with gas turbine

3.0 Boiler Accessories

3.1 Introduction

Numerous components are required for operation besides the actual boiler and the burners. Some of these components are required by law. However, even the parts that are not required by law are essential if the plant is to operate reliably. This book cannot deal with all the details relating to the legal regulations that apply in each case. These are published in regulations and directives (TRD) issued by the EU, TÜV and VGB. Regulations and safety measures for fire tube boilers are set out in DIN EN 12953. Water tube boilers are covered in DIN EN 12952. This chapter covers continuous blowdown valves, bottom blowdown valves, safety valves, water level indicators, feedwater valves and water separators.

3.2 Bottom or Intermittent and continuous blowdown valves

A small proportion of the boiler water is drained off through a valve during continuous and intermittent blowdown processes. It is necessary to differentiate between:

- Continuous blowdown, which aims to ensure that the salt content of the boiler water does not rise too high and
- Bottom or intermittent blowdown required by law, which drains off the boiler as quickly as possible in the event of an emergency.

If required, the device used for the latter process can also be used to remove the sludge collected at the bottom of the boiler.

Bottom or Intermittent blowdown valves

The process of removing the sludge at the bottom of the boiler using a manual blowdown valve is also known as "Bottom Blowdown". The intermittent blowdown valve is fitted at the bottom of the boiler, and is opened for approx. 2 seconds once or twice a day in order to purge the collected sludge from the bottom of the boiler.

A spring-loaded fast-acting drain valve, which is opened using a lever to counteract the spring action, is used to do this. Fig. 3-1.

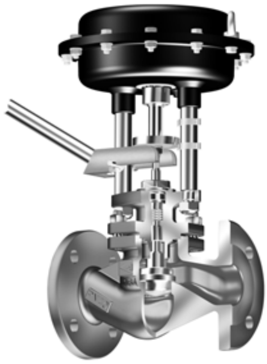


Fig. 3-1: Intermittent blowdown valve, ARI Type STEVI®BBD 415

On this type of valve, the boiler pressure is behind the plug, thus ensuring that the valve is well sealed. The seat and plug must meet strict requirements. The plug must be shaped in such a way that small particles of boiler scale and other dirt do not become trapped between the plug and the seat. Also, the valve's resting position should be 'closed'

In addition to the manual version, the valve may be operated pneumatically. This allows the intermittent blowdown process to be automated using a timer switch.

Because the release of energy is so great during bottom blowdown, regulations require that systems are in place to ensure that only one boiler is blowndown at one time. This usually means that if a manual system is in use, then only one operating handle is available. If an automated system is used, then the controls should be interlocked to prevent simultaneous operation of the valves.

Continuous (or TDS) blowdown valves

This type of blowdown uses a control valve rather than a spring-loaded valve. The continuous blowdown valve is fitted high on the side of the boiler to avoid dirt being entrained with the blowdown water.

On many boiler systems this continuous blowdown is fitted approx. 200 mm below the water surface of the evaporation space. It has been shown that, when the steam boiler is under full load, the highest concentration of salts is at this point. The disadvantage is that the highest salt concentration is at the bottom when the boiler is operating at low load and the blowdown loss is therefore also the greatest.

The simplest continuous blowdown valve is a control valve with a scale indicating the valve opening.

Systems have come on to the market in recent years which use the conductivity measure-ment to control boiler water continuous blowdown automatically.
Refer to *Chapter 3.17 Continuous (TDS) and bottom blowdown process*.

3.3 Safety valves

The steam pressure in the boiler is controlled automatically. If the steam pressure is too low, more combustion air and fuel are added. If the pressure increases, the supply of fuel and combustion air is reduced. If a fault develops in the pressure control system, or the steam consumption suddenly stops, it is possible that pressure within the boiler can increase. Regulations specify that the pressure in the boiler must never rise more than 10 % above the maximum permissible pressure. For this reason, the safety valve needs to open quickly to vent the excess pressure. The safety valve needs to be sized so that it has sufficient capacity to discharge all the steam without the permissible pressure being exceeded by more than 10 %. If two safety valves are specified, the total seat cross-section must meet these requirements.

Safety valves are safety related components, which are subject to regular checks by an independent authority (e.g. TÜV).

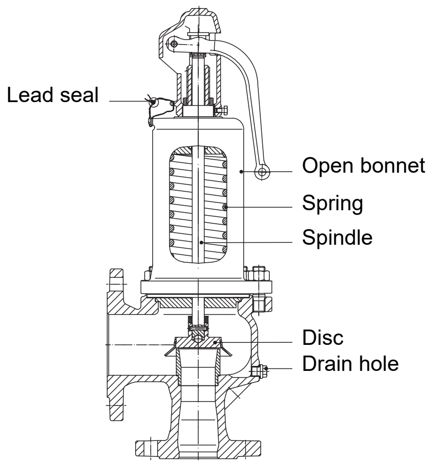


Fig. 3-2: Spring-loaded safety valve, here ARI Type SAFE 902

In a safety valve, the plug (disc) is closed, preventing the steam exiting out to the atmosphere. The spring in the bonnet transfers its force to the plug via the stem. The spring preloading, and hence the valve set pressure, can be varied using the adjusting screw. A lead seal is fitted to the cap to prevent this setting being adjusted by unauthorised persons. The flexibly mounted plug is in contact with the seat. Both the seat and the plug are precision lapped to provide a good seal, and a long service life. As the safety valve is normally closed, the top surface of the seat is covered by a plate to prevent deposits from forming. For high temperature steam applications, an open bonnet prevents the spring from becoming too hot: if a spring becomes too hot it reduces its force and results in a fall in the set pressure. A drain hole is provided to ensure that the body of the valve is clear is condensate which could impair the valve's performance.

3.4 Water level indicators

The correct water level in a steam boiler is an important factor for correct operation. If heat is not transferred to the water in the boiler, the boiler pipes and walls become so overheated that they may suffer heat damage and ultimately, failure.

To comply (with TRD 401 each steam generator), with the exception of the continuous-flow steam generator, must be equipped with at least two devices that allow the water level to be determined from the boiler control room; this is most commonly a gauge glass. A gauge glass can be replaced by:

- Two remote water level display devices or
- A reliable water level control system or a water level limiter which at least shows the water level directly.

With the first system, the boiler operator in charge does not have to be able to see the gauge glass.

Low water level

According to TRD 401 an indicator to show the lowest water level needs to be affixed in close proximity to the water level indicator.

- The lowest water level (NW) must be at least 100 mm above the highest point of the steam generator heating flues.
- The lowest water level (NW) is also to be defined as a point where it takes not less than 7 minutes to evaporate the water down to the highest point of the steam generator heating flues. (Sometimes referred to as 'sinking time')
- The lowest point of the water level indicator is 50 mm below NW - in other words, 50 mm above the highest point of the heated surface.
- If the water level drops below this value (no more water in the water level indicator), the burner should switch off automatically. This process is so important for safety that two control circuits that operate independently of each other are required (sensor A and B).

Other regulations not listed here must be complied with, depending on whether the boiler plant is monitored continuously, to a limited extent or not at all.

Fig. 3-3 shows the exact position of a water level monitoring device.

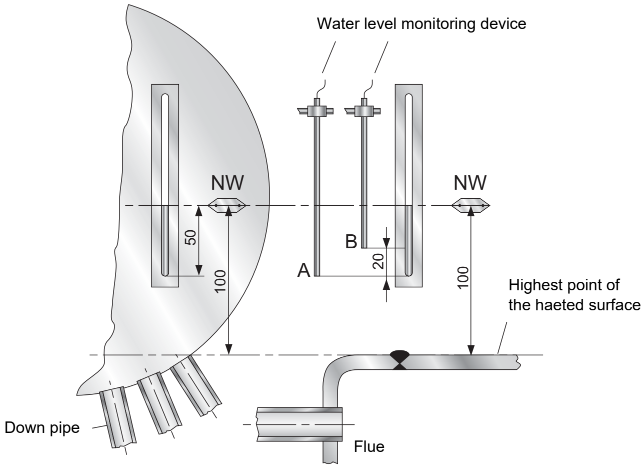


Fig. 3-3: Water level indicator for flame tube and water tube boilers

On a fire tube boiler the highest point of the heated surface is the top of the combustion chamber but on a water tube boiler the highest point is on the top downcomer tube.

Types of water level indicator

Reflex and transparent indicators are probably the most common types of water level indicators in use. A reflex water level indicator (Fig. 3-4) incorporates a thick square piece of glass with prismatic grooves on the water side. The water level indicator is fitted with pipe connections, screwed sockets and sealing rings, which are in turn connected to water level gauge cocks. Above the boiler water level, the glass prisms reflect the surrounding light. Below the boiler water level the liquid fills the prisms causing the glass to be transparent. Reflex water level indicators can be used up to a pressure of 20 bar.

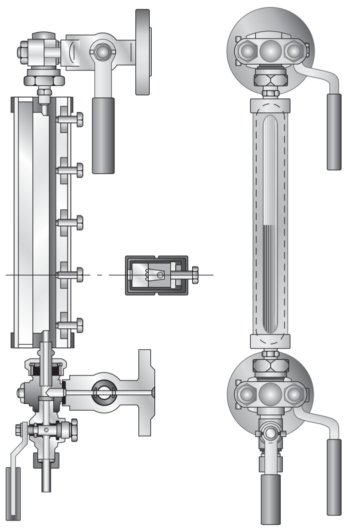


Fig. 3-4: Reflex water level indicator

For steam pressures above 20 bar transparent water level indicators are more common. The steam/water space is between two flat glass plates. The back of the indicator is illuminated (Fig. 3-5). To prevent chemicals in the boiler water from etching the glass, a thin mica plate is fitted inside of the glass. If a leak develops in the water level indicator, a ball closes an orifice.

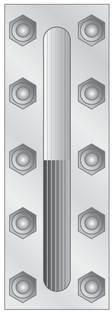


Fig. 3-5: Transparent water level indicator

3.5 Feedwater globe/check valves

Steam boiler plants with a feedwater pump meet the requirements of TRD 401, providing the following conditions are satisfied:

- If the energy source powering the feedwater pump fails, the emergency stop system on the boiler firing system must be triggered.
- Steam generators must be equipped with an adjustable boiler firing system. If an emergency stop has occurred, the design of the firing system and the steam generator must have enough safeguards to prevent evaporation of the water stored in the combustion chamber and boiler pipes.
- The steam pressure and water feed must be controlled automatically. Proof must be furnished that the controller for the water feed is reliable.
- A reliable low water safety device (water level limiter) must be provided in addition to the water feed control device.

Steam boiler plants with steam generators, which do not meet this demand, must have at least two feed pumps. The capacity of the feed pumps must be equivalent to 1.25 times the 'from & at' rating of all the system's steam generators allowed for operation.

If more than 5 % of boiler water as compared to the 'from & at' rating is continuously removed, the capacity of the feed pumps must exceed this volume by 5 %.

In addition, where at least two feed pumps are used, the following requirements must be observed:

- If the feed pump with the greatest capacity fails, the remaining feed pumps must be able to handle the total capacity.
- Two sources of energy independent of each other must be available. The feed pumps are to be connected to the source of energy so that, if one source of energy fails, the feed pumps that are still operational are able to handle the total required capacity.

It is permissible for both pumps to supply the boiler via one feedwater pipe. If several steam boilers are installed, one back-up pump is sufficient. This means that three feedwater pumps are to be installed for two boilers. Each feedwater pump should be provided with one check valve with a shut-off feature. The purpose of this valve is to prevent the contents of the boiler flowing out into the boiler house if a pipe breaks. In addition, the check valve prevents boiler water from flowing back into the feedwater system should the feedwater pump fail and a pump check valve fail to operate.

There are valves and fittings which combine both functions (shut off and stopping return flow) in a single housing (Screw down non-return or SDNR valve).

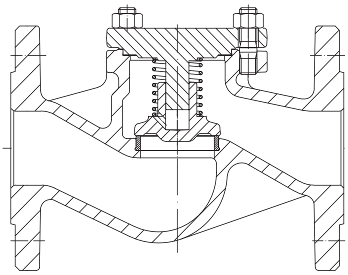


Fig. 3-6: Check valve,
ARI Type 003

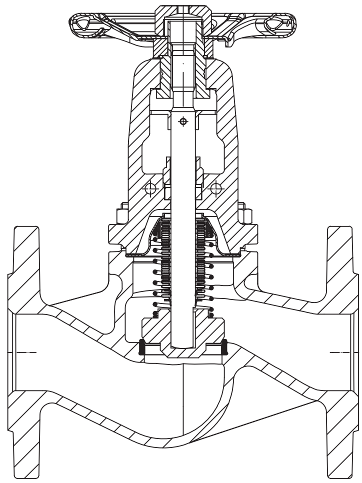


Fig. 3-7: Check valve with shut-off feature,
ARI Type FABA®-Plus 046
with loose plug and spring

3.6 Separators

The steam produced in the boiler typically contains between 5 and 10 % water (by mass). This mixture of saturated steam and fine particles of water is generally called wet steam. The water content many be much higher if:

- There is an excessively high level of water in the boiler
- The rate at which steam is drawn from the boiler is too high (overload)
- The TDS concentration in the boiler water is too high.

Wet steam also forms if the steam condenses in a distribution pipe, e.g. as a result of poor or no insulation or if the pipe to the consumer is too long and it does not contain enough drainage points. These last causes given are very simple to eliminate. *Chapter 5.0 Pipe Drainage* looks at the drainage of steam pipes.

The two most important prerequisites for a "dry" steam pipe are:

- Steam traps in the steam pipe every 50 to 75 m, depending on the insulation.
- Steam traps with drainage points at each rising section of the steam pipe.

Most process equipment is very sensitive to condensate entrained with steam. Water particles striking the wall of a pipe, on a bend, for example, at a speed of 20 m/s (72 km/h) will cause erosion; in the long term this can lead to leakage. Thin walled steam pipes within heat exchangers can develop leaks for the same reason. It is therefore advisable to install a separator upstream of the inlet connection.

Injectors also require dry steam for reliable operation. It is advisable to install a steam separator (steam drier) to protect sensitive equipment from wet steam. Fig. 3-8 shows a cyclone steam separator which operates using centrifugal force. Here, too, the heavier water particles are separated from the steam and flow down the wall towards the steam trap.

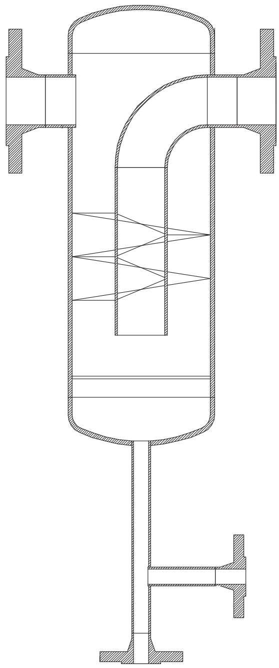


Fig. 3-8: Cyclone steam separator

Fig. 3-9 shows a water separator with a mist eliminator (demister). The separator contains a wire mesh pack, which consists of several layers of knitted and corrugated metal wires. This provides a certain density of mesh.

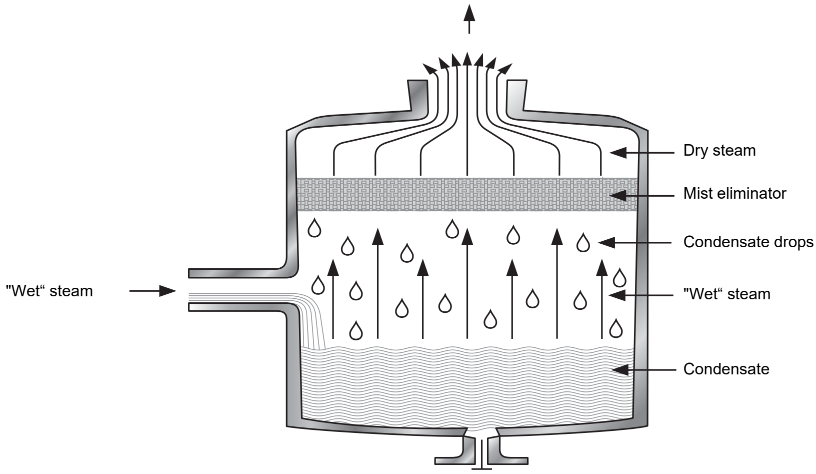


Fig. 3-9: Water separator with a mist eliminator (demister)

The water particles are trapped and collected by the wires. In contrast to the water particles, steam can very easily find a way through this tightly knitted mesh. The small particles collect at points where the wires of the mesh cross and form small droplets. Above a certain droplet size they fall in the opposite direction to the flow. In spite of the close density of the woven mesh, a free passage of approx. 98 to 99 % remains. The pressure loss is minimal. In order to achieve a dryness of 99.5 %, the speed of the steam in the demister should be between 2.5 and 7.5 m/s.

- If the speed is too low, the water particles do not stick to the wires
- If the speed is too high, the droplets are entrained in the steam

If there is likelihood that water particles will be entrained during steam generation, a separator should be fitted directly at the boiler steam outlet. As a general rule, boiler manufacturer's pre-install cyclone-type separators in water tube boilers.

3.7 Pressure reducing valves

Not all process equipment is designed to work at the operating pressure of a steam boiler. In order to condition the steam pressure upstream of an item of equipment, pressure reducing valves are fitted upstream. Example: Autoclaves, evaporators and also deaerators. In these cases, a pressure reducing valve controls the steam pressure so that the permissible pressure is not exceeded. A safety valve must be fitted downstream of the pressure reducing valve in order to ensure that, if the pressure reducing valve develops a fault, a hazardous situation cannot occur. A steam trapset must always be installed upstream of a pressure reducing valves. Fig. 3-10 shows a pressure reducing system with all associated devices, valves and fittings. Because the steam temperature would damage the diaphragm in the pressure reducing valve, a small vessel is fitted upstream of the control line.

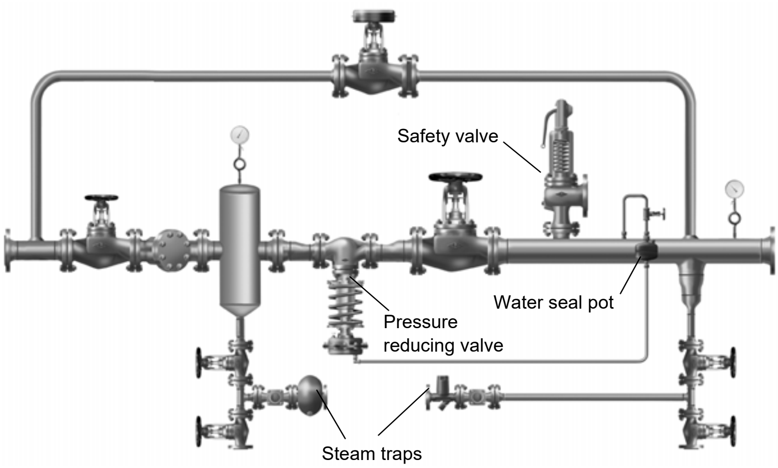


Fig. 3-10: Pressure reducing system, ARI Type PRESys®

Fig. 3-11 shows a directly controlled pressure reducing valve. The spring tension (and therefore the required steam pressure) can be set using an adjusting nut on the stem.

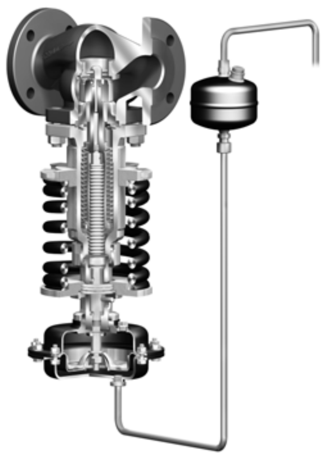


Fig. 3-11: Pressure reducing valve, ARI Type PREDU®

Self-operated pressure reducing valves operate using the pressure of the medium as the motive force. Their purpose is to reduce a high pressure (upstream pressure) to a lower one (downstream pressure). The valve is required to modulate to maintain a specific downstream pressure even in the case of changes in upstream pressure fluctuations, or changes in flowrate. A marginal seat is screwed into the through housing. The plug has a small parabolic shoulder, which ensures the control process is not affected by vibration even at very low flow rates. The pressure reducing valve has two stainless steel bellows. The lower one is used to seal the stem from the outside. The upper one is the balancing bellows, which is used to balance out forces at the plug. To achieve this, the upstream pressure reaches the interior on the outside of the bellows through a bore in the plug. The inner side of the bellows is connected to the downstream pressure side via openings. As the effective area of the bellows is the same size as the area of the seat, the differential pressures are compensated and the influence of fluctuations in upstream pressure is only minimal.

The pressure reducing valve is operated by the diaphragm. The downstream pressure acts on the diaphragm via the buffer unit (water seal pot) and balances against the force of the spring. The pre-tensioning of the spring can be changed by the adjusting mechanism so that both forces are balanced at the desired downstream pressure. If the flowrate of steam changes, the plug is adjusted until balance is achieved again.

The pressure reducing valve in Fig. 3-11 is directly actuated by the steam pressure. Electrically or pneumatically actuated control valves are used almost exclusively for precise pressure control. Pressure is measured using a pressure sensor, which passes a signal to the process controller. This compares the specified value with the actual value and, depending on the control deviation, outputs a signal of 4 - 20 mA to an electro-pneumatic positioner, which changes the position of the control valve.

3.8 Water treatment

Introduction

In spite of the many publications on boiler feedwater treatment, misunderstandings frequently occur.

Water chemistry is a complex subject which requires all the skill of a trained chemist. This chapter is a simple explanation of how and why boiler feedwater is treated in practice. Firstly, we will discuss softening feedwater in low-pressure boiler plants. The aim is to prevent deposits from forming on heated parts of the boiler. Feedwater treatment for high-pressure boiler plants will be discussed later.

Generally, boiler feedwater is a mixture of returned condensate and feedwater. Condensate is condensed steam, equivalent to distilled water, and it can generally be used again as feedwater with only a minimal amount of additional treatment. Additional feedwater is required to replace the loss of condensate in the water/steam circuit. This loss of condensate is produced by:

- Steam injection for heating, e.g. for a water tank
- Contaminated condensate, which has been made unusable for further use as a result of contamination by lubricating oil or process chemicals.
- Condensate is not being returned
- Continuous boiler blowdown
- Steam and water leaks
- Water vapour and flash steam into the atmosphere

Sources of feedwater

The main sources of feedwater are:

- Groundwater, such as wells and spring water
- Bodies of surface water such as rivers, lakes or ponds
- Drinking and service water

Natural sources of water cannot automatically be used as feedwater for boiler plants. Most sources of ground and surface water are contaminated by a variety of substances so that, if it is used as feedwater without being treated, it would result in damage being caused to the boiler plant.

Even drinking water processed in water works is not suitable to be used as boiler feedwater. Clearly, feedwater needs to be treated and monitored. Apart from steam boilers, feedwater pipes and condensate pipes must be protected against corrosion.

Water chemistry

In the following section, we will try not to use any more chemical terms than are absolutely necessary. Where possible, we have replaced chemical notations and formulas with the chemical names, adding the formula in brackets, if necessary.

The following chemical compounds occur in water:

Designation	Chemical formula
Calcium hydrocarbonate	$Ca(HCO_3)_2$
Magnesium hydrocarbonate	$Mg(HCO_3)_2$
Calcium sulphate	$CaSO_4$
Magnesium sulphate	$MgSO_4$
Calcium chloride	$CaCl_2$
Magnesium chloride	$MgCl_2$
Common salt	$NaCl$
Silicic acid	SiO_2

In summary, one can say that the chemical compounds in water consist of calcium and magnesium coupled with hydrocarbonates (bicarbonates), sulphates and chlorides. In addition to this, there is sodium coupled with chlorides and silicic acid. Calcium and magnesium compounds are hardening constituents. These compounds must first be broken down before the water can be used as feedwater. The molecules of the salts dissolved in the water split up (dissociate) into electrically charged particles called ions. Calcium hydrocarbonate splits into calcium (Ca) and hydrocarbonate $(HCO_3)_2$. Magnesium hydrocarbonate dissociates in the same way.

Calcium and magnesium hydrocarbonate are difficult to dissolve in water. They represent carbonate hardness (or carbonates of alkaline earths). Once the water is heated, they will be deposited on or in the boiler pipes. The carbon dioxide escapes as a gas and condenses with the steam to form acidic water. The pH value drops sharply and results in the dreaded carbonic acid corrosion in the condensate pipes. Calcium and magnesium compounds containing sulphates and chlorides form the noncarbonated hardness (or non-carbonates of alkaline earths) and remain soluble in water. Only when the concentration in the boiler water has become much too high, e.g. as a result of too little continuous blowdown, can non-carbonates cause scale deposits.

The sum of carbonate harness and non-carbonate hardness produces total hardness (total of alkaline earths).

Requirements for feedwater

It is sufficient for a low-pressure boiler if the carbonate hardness is removed from the water using an ion exchanger (softener). Chloride and sodium are not critical if the concentration is low.

In the case of high-pressure boilers, if possible all impurities should be removed. A complex ion exchanger, a so-called demineralisation plant, is used for this. The rule that applies is the higher the steam pressure, the higher the requirement for water purity. Demands in relation to the composition of boiler feedwater are especially high where steam is used to drive a turbine.

In order to prevent the hardness from dis-associating in the softener in low-pressure systems (which do not always work perfectly), metered chemicals are added to eliminate the residual hardness. In this form, the hardness cannot be deposited in or on the boiler pipes but instead it collects at the bottom as sludge. The sludge is then removed during the bottom blowdown process. Orthophosphates and polyphosphates are then used as residual softening agents. Recently, increasing use has been made of polymers and complex binders for reasons of environmental protection.

If scale is deposited on or in the boiler pipes, it reduces the amount of heat transferred. As the layers of scale become thicker, the wall of the pipe overheats and there is a serious risk that the pipe will burst, Fig. 3-12.

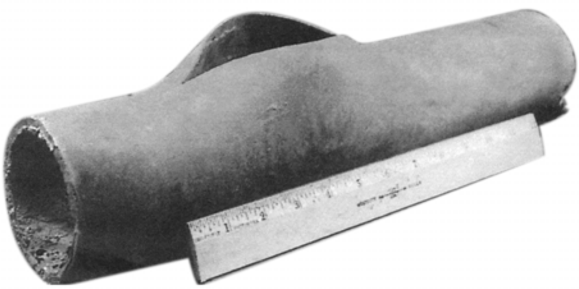


Fig. 3-12: Burst boiler pipe

Fig. 3-13 shows the difference in the temperature characteristic between a clean pipe wall (maximum temperature 380 ºC) and a wall where the thickness of boiler scale is 1 mm (maximum temperature 670 ºC).

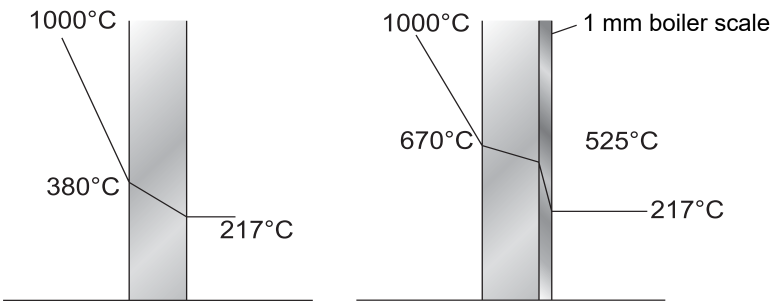


Fig. 3-13: How the layer of scale influences the temperature of the pipe wall

The ion exchanger operates according to the principle that magnesium and calcium ions are replaced by less damaging sodium and hydrogen ions. This softens the water and demineralises it.

An ion exchanger is a vessel, also called a filter, which is filled with plastic waterproof granules. These granules are able to exchange their absorbed ions for the ions present in the water. Styrol resins are almost exclusively used for these granules. The sodium or hydrogen ions in the resin are replaced by the magnesium or calcium ions in the water.

All metal (Na^+, Ca^{++}) and hydrogen ions (H^+) have a positive charge. Acidic residues (Cl^-, SO_4^{--}, CO_3^{--}) and hydroxide ions (OH^-) have a negative charge.

Ions with a positive charge are called cations. Ions with a negation charge are anions. In aqueous solutions there are always an equal number of cations and anions. Ion exchangers are divided into two main groups:

- Weakly and strongly acidic cation exchangers
- Weakly and strongly basic anion exchangers

These exchangers soften, partly demineralise and fully demineralise water in systems with one or two filters.

3.9 Softening

Process

A cation filter or cation exchanger (also known as a 'base exchange' softening unit) is commonly used to soften water (replacing calcium and magnesium ions). A resin bed is enriched with sodium ions (made of NaCl) in common salt brine. Upon leaving the unit, all calcium and magnesium ions have been replaced by sodium ions and the water has been softened.

The exchange process only functions up to a certain saturation level. The filter reaches its performance limit when the number of exchanged ions is almost equal to the number of absorbed sodium ions. A full exchange is then no longer possible.

Exceeding the limit allows unwanted ions to escape through the unit; the water remains 'hard' and is no longer suitable as feedwater. Generally, base exchange units comprise two cylinders, one on-line, and the other on re-generation/stand-by.The resin bed is regenerated by back-flushing with a brine solution.The regeneration process takes place in three phases:
Removal from service —> Flushing —> Regeneration —> Washing —> Starting up

The purpose of flushing is to loosen up the exchange material and remove dirty particles.

A 10 % common salt solution is required for regeneration. The regeneration process is the reverse to the softening process. All calcium and magnesium ions absorbed from the resin are exchanged for sodium ions in the brine. The calcium and magnesium ions are discharged into the waste water system with the washing water. The resin granules should remain sufficiently long in contact with the brine in order to regenerate fully. The manufacturers of resins specify a regeneration time and the percentage of brine per m^3 of resin. The brine is displaced with water in the last stage of the process. Only then is the filter ready for a new softening process. This procedure is fully automated for the majority of softeners.

Composition of softened water	
Sodium hydrogen carbonate	$NaHCO_3$
Sodium sulphate	$Na2SO_4$
Common salt	NaCl
Silicic acid	SiO_2

Composition of untreated water	
Calcium hydrogen carbonate	$Ca(HCO_3)_2$
Magnesium hydrogen carbonate	$Mg(HCO_3)_2$
Calcium sulphate	$CaSO_4$
Magnesium sulphate	$MgSO_4$
Calcium chloride	$CaCl_2$
Magnesium chloride	$MgCl_2$
Common salt	NaCl
Silicic acid	SiO_2

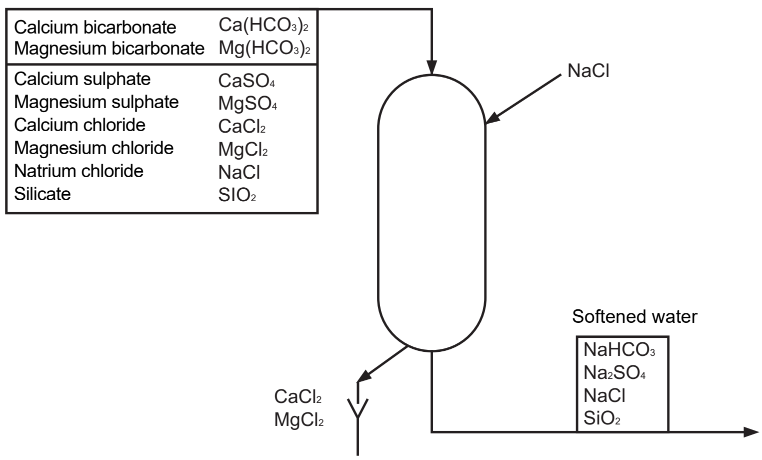


Fig. 3-14: Softening in the cation exchanger

The softening process is shown as a diagram in Fig. 3-14. After the process, all calcium and magnesium ions have been removed from the water and replaced by sodium sulphate, common salt, sodium hydro-carbonate and silicic acid. This mixture still contains hydro-carbonates and can cause carbonic acid corrosion.

Protection against residual hardness

If a fault occurs during the process (concentration of brine too low, time switch defective or cooling water in the condensate), the hardening constituents remain dissolved and can be deposited on or in the pipes.

In order to make the hardening constituents in the steam boiler inert, primarily phosphates (ortho-phosphates, tri-sodium phosphates) are metered so that the hardening constituents are precipitated as sludge on the bottom of the boiler. The concentration of phosphates in the boiler water must be between 25 and 50 mg/l and must be monitored regularly.

If the concentration drops suddenly, this is a sign that a source of hardness has entered somewhere in the system. The cause must be ascertained very quickly and eliminated. Chelates are also used in addition to phosphate metering. However, chelates are difficult to handle and therefore they are metered in combination with phosphates. Here the phosphates act as a tracker. An environmentally friendly agent binds the phosphates, which distribute the hardness in such fine particles that it is not deposited as scale.

3.10 Formation of corrosion

Apart from the care required to supply the boiler with water of the required quality, undesirable gases must be prevented from entering the boiler. The presence of oxygen in the feedwater causes corrosion in the boiler and the rest of the steam system.
The presence of carbon dioxide results in acidic corrosion or carbonic acid corrosion in the condensate system. A deaerator removes the oxygen (see *Chapter 3.14 Deaerator*).

Carbon dioxide is partially removed in the deaerator. The remaining carbon dioxide is neutralised by means of decarbonisation and chemicals.

Carbon dioxide

In water chemistry carbon dioxide comes in free and bound form. Free carbon dioxide (CO_2) is absorbed from the air. Bound carbon dioxide comes in the form of hydrocarbonate ($-HCO_3$).

Free carbon dioxide and oxygen are expelled in a deaerator. However, the temperature in the deaerator is too low to split and remove the hydrocarbonate. The hydrocarbonate is taken with the feedwater to the steam boiler. As a result of the high temperature the hydrocarbonate breaks down into carbon dioxide and water. The carbon dioxide enters the steam system. During the condensation process the carbon dioxide dissolves in the condensate. The condensate becomes acidic, with a pH value between 4 and 5. It subsequently causes corrosion in the condensate system, especially if the plant is out of service and condensate remains in the pipes.

Neutralising chemicals in the form of volatile alkaline agents such as ammonia or amines are metered in order to prevent this acidic corrosion. Sometimes amines are also metered in order to apply a protective layer to the inside of the pipes in the condensing part of the system. A disadvantage of the amines that form this protective layer is that they dissolve old layers of corrosion, which in turn clog steam traps and cause additional sludge to form on the bottom of the boiler. It is advisable to be especially cautious!

Decarbonisation is another method of removing hydrocarbonate. Hydrocarbonate is split into carbon dioxide and water by metering acid into the softened water. The carbon dioxide is then expelled in a CO_2 sprinkler via air flow. After this treatment the pH value of the acidic water must be increased to a value of approx. 8 by means of chemicals (for example, caustic soda).
Metering caustic soda causes an increase in drain loss.

CO_2 sprinkler

Fig. 3-15 shows a CO_2 sprinkler. The sprinkler consists of a packed column over a water basin.

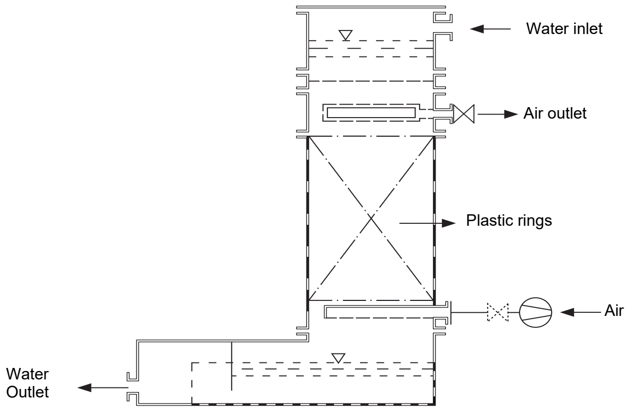


Fig. 3-15: CO_2 sprinkler

The acidic water evaporates in the upper part and then trickles into a water basin via several layers filled with plastic rings. Air is blown in the opposite direction to the water. This means that air and water come into contact with each other very vigorously. The carbon dioxide is removed down to a figure of less than 10 mg/l by reducing the partial pressure for CO_2. Air and CO_2 are blown into the atmosphere. Because the water is aggressive (acidic), CO_2 sprinklers are generally made of plastic.

3.11 Softening and decarbonisation

To remove the bound carbon dioxide, the untreated water is first passed through an H ion exchanger or H filter during the decarbonisation process in a two-filter process and it is then passed via an Na ion exchanger or Na filter. The H filter is regenerated using hydrochloric acid and the Na filter is regenerated by means of common salt, Fig. 3-16.

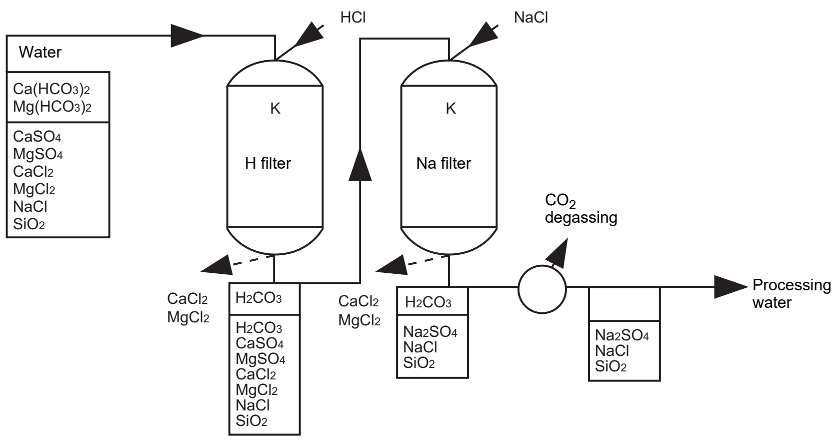


Fig. 3-16: Softening and decarbonisation, H and Na filters connected in series

The Ca and Mg ions are exchanged for hydrocarbonates in the H filter. Carbonic acid (H_2CO_3) is formed as a result. Everything else remains unchanged in the H filter. The Ca and Mg ions from the Ca and Mg chlorides are exchanged for Na ions in the Na filter.

Downstream of the Na filter the water containing the carbonic acid from the H filter is sprayed in a CO_2 sprinkler. Air is blown in the opposite direction into the sprinkler from below. The carbonic acid splits into water and carbon dioxide. The carbon dioxide is blown out in the air.

Sometimes CO_2 sprinklers are fitted between two filters. As a result of this, a slightly smaller Na filter can be fitted.

Both filters can also be operated in parallel. Here it is important to make sure that partial flows are correctly distributed and water quality is monitored continuously.

3.12 Demineralisation

Softening is not enough for high-pressure steam plants. Although softened water does not leave scale deposits, the high concentration of residual salts and the high steam temperature will cause corrosion. For this reason the feedwater for high pressure plants must be demineralised. Fig. 3-17 shows an example of a demineralisation system.

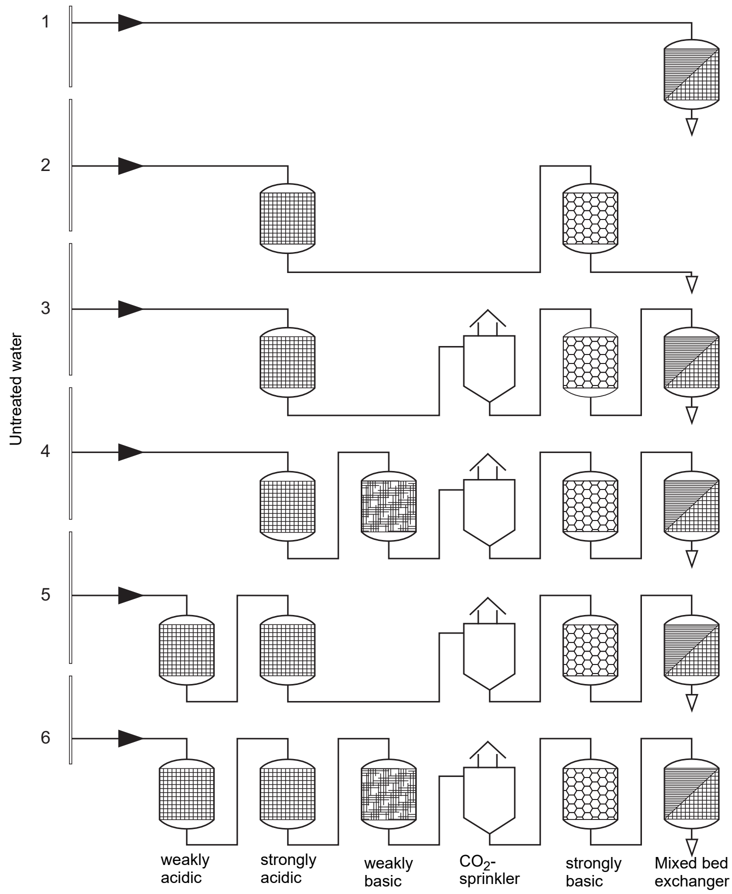


Fig. 3-17: Demineralisation system

A strongly acidic cation filter regenerated with hydrochloric acid is connected in series to a weak basic anion filter regenerated with a sodium hydroxide solution. The CO_2 sprinkler is fitted behind the anion filter. Water from the sprinkler is already almost demineralised. The CO_2 sprinkler also reduces the amount of chemicals required for the strongly basic anion exchanger. A strongly basic anion filter is fitted in addition behind the CO_2 sprinkler to remove the silicic acid. A mixed bed exchanger is fitted between to act as a "policing filter" to improve the content of residual salts and silicic acid.

It is particularly necessary to remove silicic acid in the case of high-pressure steam, e.g. for turbine systems. Silicic acid is not only deposited on the heated surfaces of the steam boiler but it may also be deposited on turbine blades, thus influencing the efficiency of the turbine.

Fig. 3-18 shows the basic circuits, which are used in different combinations, depending on the properties of the untreated water, the intended use of the demineralised water and the prevailing operating combinations.

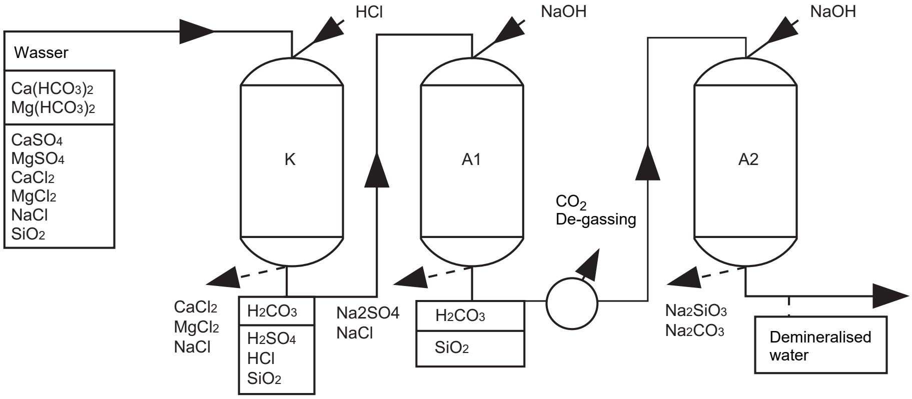


Fig. 3-18: Basic circuits of demineralisation systems

3.13 Monitoring make-up, feed and boiler water

Guidelines

A check should be made every day to monitor the quality of the make-up, feed and boiler water. Suppliers of chemicals specify the limit values for the many different characteristics of water in combination with the recommended chemicals.

The following tables provide the recommended values for fire tube boilers up to 20 bar steam pressure for feed and boiler water.

Boiler water		
Fire tube boiler up to 20 bar		
p value	mmol/kg	9 - 11
Silicic acid	mg/kg	15 x p value
Conductivity measurement	µS/cm	6000
Phosphates	mg/kg	30 - 80

Feedwater		
Flame tube boiler up to 20 bar		
pH		7.0
Hardness	mval/kg	0.036
Iron	mg/kg	0.3
Copper	mg/kg	0.1
Oil	mg/kg	3.0
Oxygen	mg/kg	0.1

Sample assessment

An example to illustrate water analysis:

		Make-up water	***Feedwater deaerated***	***Boiler water***
Conductivity measurement	*µS/cm*	*620*	*200*	*4000*
pH		*8*	*9.1*	*12.5*
p value	*mmol/kg*		*0.1*	*24*
m-value	*mmol/kg*	*1.8*	*0.8*	*28*
Chlorides	*mg/kg*	*20*	*10*	*320*
Hardness	*mval/kg*	*< 0.007*	*< 0.007*	
Phosphates	*mg/kg*			*60*
Sulphites	*mg/kg*			*90*

This is what an assessment of the analysis looks like:

The hardness is below the recommended value and is in order.
The chloride content of the make-up water (20 mg/kg) is a given variable but 10 mg/kg is measured in the deaerator. It can be assumed that the chloride content of the returned condensate is 0 mg/kg.

Consequently, the feedwater contains $= \frac{10}{20} \times 100\% = 50\%$ *make-up water.*

The chloride content in the boiler is 320 mg/kg.

With regard to the make-up water, the continuous blowdown percentage is therefore 20/320 x 100 % = 6.25 %. With regard to the feedwater, the continuous blowdown percentage is 10/320 x 100 % = 3.125 %.

The continuous blowdown percentage can also be calculated using the m-value or the conductivity. The p-value is above the recommended value of 10 mmol/kg and therefore further continuous blowdown is required. The phosphate and sulphite values are reduced as a result of the increased continuous blowdown process. It is possible that the metering rate for chemicals may have to be increased.

Checking the hardness

The unit for measuring hardness is ºD (German hardness) or mval/kg. It is converted as follows:

1 ºD = 0.357 mval/kg
1 mval/kg = 2.8 ºD

A fast tried and tested method of determining the hardness of water is to put the water in a test tube with a few drops of a standardised soap solution. The solution is then shaken.

- If the head of foam remains, the water is soft.
- If the head of foam disappears or if it does not form at all, the water is hard.

3.14 Deaerator

Introduction

The boiler feedwater should be deaerated before it is supplied to the steam boiler. Feedwater is generally composed of make-up water and returned condensate. The gases contained in condensate are oxygen (O_2) and free carbonic acid or carbon dioxide gas (CO_2). The gases are removed in a deaerator.

Small steam generators are often not equipped with a deaerator. In this case, the gas contained in the water is removed by adding chemicals. Oxygen in the feedwater causes pitting corrosion in the boiler and on pipes. The higher the steam pressure, the lower the level of oxygen permissible in the feedwater. For boilers with steam pressure of less than 20 bar, the maximum O_2 content is 0.03 mg/l. A maximum of 0.02 mg/l is permitted for higher steam pressures. Free carbon dioxide produces acidic condensate and therefore causes acid corrosion in the condensate pipes. This chapter starts with a brief description of how the deaerator works. This will be followed by a section in which the details of the different versions are covered. Finally, some recommendations will be provided for practical situations.

How deaerators work

There are two main principles:

- ***Spray deaerators:***
  The feedwater is supplied to the deaerator from above via the sprayer.
- ***Cascade deaerator:***
  The feedwater is sprinkled into the deaeration dome and cascades over a series of trays before falling into the body of the vessel.

For steam, the pressure is typically controlled to around 1.2 bar a. The mixture of make-up water and returned condensate is therefore heated to 105 ºC. This temperature is sufficient to drive out the gases contained in the water (105 ºC is the saturation temperature of steam at 1.2 bar a).

The deaerator operates in accordance with Henry Dalton's law on the absorption of gases. Put simply, the gases are removed from the water because the partial pressure (concentration) of the gases in the fluid is higher than the concentration of gases in the steam.

A prerequisite is that water is sprayed or distributed with the finest possible droplets into a space with saturated steam above the feedwater level. The atomisation should be so fine and the distribution so intensive that the temperature of feedwater is 105 °C before it reaches the level of the water in the vessel. Therefore, in this first stage of deaeration approximately 90 % of the gases contained are expelled with intensive spraying. In the second stage of deaeration steam is fed to the water space of the deaerator via a nozzle.

If pressurised condensate is available, this can also be fed to the deaerator. The flash steam released is used to drive out residual gases.

3.15 Spray deaerators

Design

Fig. 3-19 shows a spray deaerator. The mixture of make-up water and condensate is atomised as a fine spray from above into the steam space via a sprayer (1). The steam is supplied via a manifold (2).

In some large deaerators the manifolds are fitted in a V-type shape and are called screens (5).

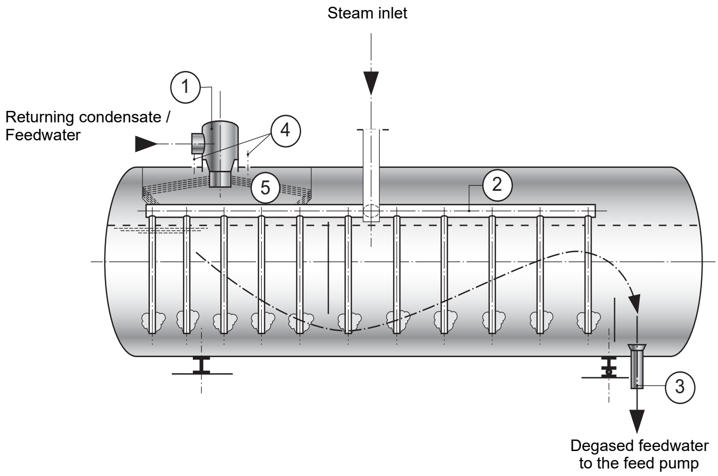


Fig. 3-19: Spray deaerators

The feedwater off-take to the feed pump (3) is to be fitted as far as possible away from the water inlet. The residence time of the water in the deaerator should be a minimum of 25 minutes. To prevent water that is not yet fully deaerated from escaping, a baffle plate has been fitted in the deaerator illustrated. This baffle is not fitted in smaller deaerators. The gases are drawn off at the point with the greatest concentration of gas. This is generally directly adjacent to the spray unit (4).

Deaerators are protected inside by a special coating in order to prevent corrosion. The condition of this protective coating should be inspected and repaired, if necessary, during routine inspections of the deaerator.

Sprayer

Fig. 3-20 shows a high-performance sprayer. The sprayer consists of a housing, in which a spring-loaded, perforated piston is suspended. Depending on the sprayer loading, the number of free perforations varies according to the spring pressure. In this way the spraying is almost completely in proportion to the desired capacity. A scale fitted on the spring, outside the deaerator, shows the loading.

Possible clogging of the perforations can be eliminated by cutting off the supply of water for a short time. During this operation the piston is pulled upwards to scrape off the dirt.

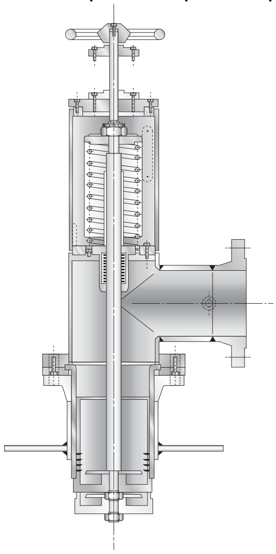


Fig. 3-20: High-performance sprayer

Another design for smaller deaerators is shown in Fig. 3-21. As the volume of water increases, the gap opens wider and the spray is not as fine. A disadvantage of this sprayer is the fact that it cannot be adjusted during operation. For this simple sprayer it is advisable to check the mechanical operation when the sprayer is cold each time the deaerator is serviced.

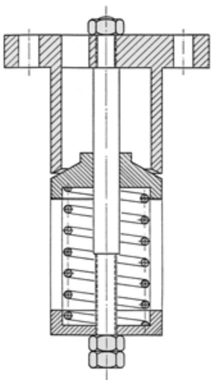


Fig. 3-21: Simple sprayer

Cascade deaerator

Fig. 3-22 shows a cascade deaerator. The cascade is mounted on the deaerator dome. The feedwater and returned condensate are sprinkled into the deaeration space by means of trays fitted above the deaerator vessel.

There the water spreads over the cross section of the deaerator. As the cascades are continuously overflowing, a thin wall of flowing water is formed, which combines to form a considerable water surface area. This process involves making intensive contact with the steam that flows in the opposite direction. As already described for the spray deaerator, the water is at a temperature of 105 ºC when it hits the surface of the water. Part of the steam is routed to the cascade space and the other part is supplied via a nozzle pipe on the base of the feedwater tank. The expelled gases, are taken away and vented.

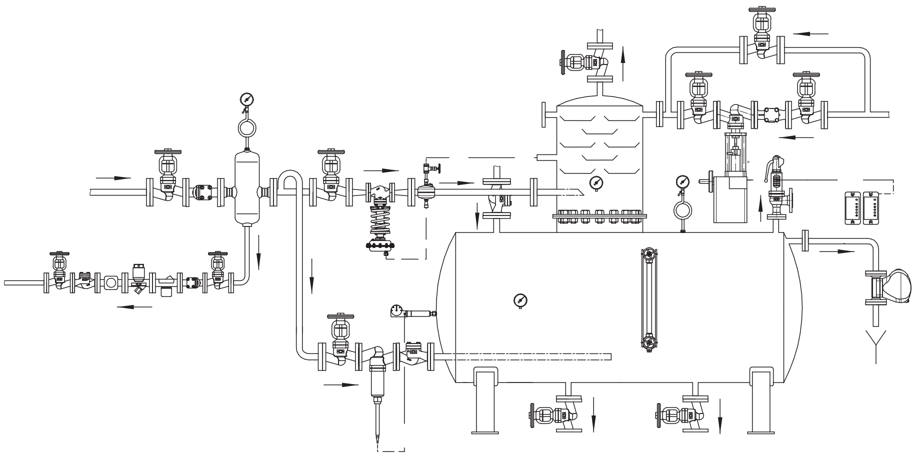


Fig. 3-22: Cascade deaerator

3.16 Practical information on operation

Flash steam

Finally, in all deaerators the gases released from the water collect in the steam space above the water level. They are then vented to atmosphere using an air vent. Some flash steam should always be discharged in order to completely purge of oxygen and free carbon dioxide from the steam space.

The higher the concentration of gas in the steam, the lower the efficiency with which gas is separated from the water. For this reason, the flash steam should be discharged as close to the water inlet as possible, i.e. at the sprayer or above the cascades.

If the deaerator temperature falls below the saturation temperature of the steam, it is a sign that insufficient flash steam has been released (e.g. below 1.2 bar a / 105 ºC).

A pressure measurement would show the total pressure of gases and steam. However, the partial gas pressure accounts for the majority of the prevailing pressure of 1.2 bar. Consequently, the actual steam pressure is under 1.2 bar and the water temperature is also below 105 °C. For this reason, it is advisable to measure the water temperature as well as the deaerator pressure.

Heat recovery from flash steam

In the case of larger deaerators, it may be economic to use the heat contained in the flash steam for preheating by passing it via a heat exchanger. The benefit gained from the steam must be weighed against the high-cost of materials for heat exchangers (because some constituents of the gas encourage corrosion)

Separation of the pump from non-deaerated water

As already noted, the residence time of the water in the deaerator should be at least 25 minutes. It is necessary to prevent incompletely deaerated water from flowing directly to the suction connection of the pump. In other words, there must be no contact between the incoming non-deaerated water and the feedwater pump.

For both deaerator types - cascade and spray deaerators - the position of the water spray nozzle should be as far as possible from the feedwater pump. Unfortunately, this is not always what happens in practice. Some manufacturers build barriers in the deaerator to force the water to take a longer route through the deaerator.

Mixture temperature resulting from make-up water and returned condensate

Sufficient live steam must be supplied in order to achieve the desired degree of gas separation. This is ensured if the mixture of make-up water and condensate is not greater than 90 °C to 95 °C. This also applies if water and condensate are supplied separately. Condensate supplied under pressure, which flashes in the deaerator, does not need to satisfy this condition.

Safety valve

The majority of deaerators are protected with a safety valve set at 1.4 bar a. Deaerators are subject to mandatory inspections at pressures higher than 1.5 bar a.

On some older deaerators, the overflow/pressure relief station takes the form of a water seal (manometric loop). However, in practice this system has hidden weaknesses. Every time there is a hammer or pulse in the system which exceeds the head pressure of the water column, the water seal empties and steam escapes. The deaerator pressure would then need to be reduced to restore the water seal.

Today safety valves are nearly always used for pressure relief to eliminate this reliability issue.

Correct metering of chemicals

Chemicals are added to scavenge oxygen and regulate the pH value. As already noted, oxygen scavenging chemicals are not added in the deaerator, but instead they have to be added just before (or in) the feedwater pump suction pipe. The chemicals will bind the remaining oxygen at this point and fulfil a role similar to the last safety precaution.

If chemicals were to be added before the deaerator, considerable quantities of chemicals would be consumed and they would be a substitute for the intended purpose of the deaerator but in an inefficient way.

Correct air venting

Good air venting is achieved where the concentration of gases is the highest, i.e. as close to the water inlet as possible. In the case of a spray deaerator, this is immediately adjacent to the sprayer and for a cascade deaerator it is at the highest point of the cascade. An air venting point close to the sprayer is more efficient than if it is distributed on the surface.

A typical example of incorrect air venting design is shown in Fig. 3-23.

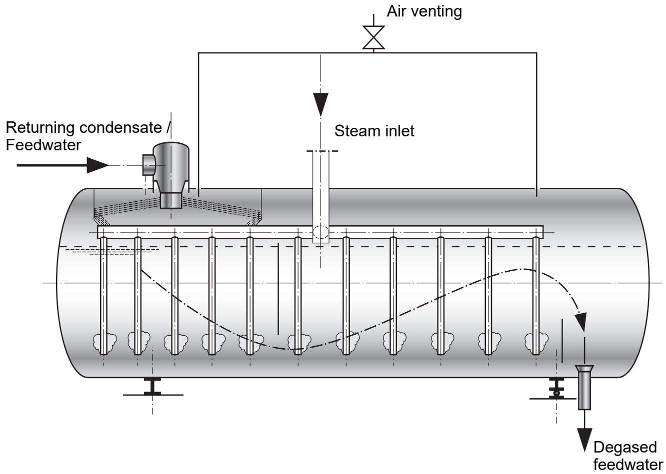


Fig. 3-23: Incorrect double venting of the deaerator

The figure shows air venting at two points - a correct air venting point close to the sprayer and an (unnecessary) second one on the other side of the deaerator. Both air venting points are discharged via a shared pipe. An orifice or a control valve can be added in the air venting pipe.

An air venting point close to the sprayer will not function or will not operate properly because the steam pressure close to the sprayer is always slightly lower than the pressure at the right-hand side of the deaerator. As a result of this, only the right-hand side would be vented and not the space with the greatest concentration of gas. If such an arrangement exists, only the air venting point close to the sprayer should be opened. Should it still prove to be necessary to vent the other side as well, this should be done via a separate pipe. Nevertheless, the air venting performance of this separate device would be less effective.

On/off level control

On some deaerators the water level is controlled via an on/off switch for the pump or an OPEN/CLOSED control.

This process involves a large quantity of cold water being supplied to the aerator within a very short time, which means that the steam control system must cope with peak load.

If a tonne of cold water is supplied to the deaerator within ten minutes, the steam boiler needs to produce an additional peak output of 1 t/h in order to maintain the required pressure in the deaerator.

Continuous control of the water level is an advantage in order to avoid these undesirable peak loads. If continuous control is not possible, the quantity of water supplied per switching cycle should be kept to a minimum. One option is to keep the difference between the maximum and minimum signalling level as small as possible. The additional electricity required for the higher switching frequency increases the thermal load on the make-up water pump.

Deaerators with a steam pressure of > 1.2 bar

If there is no opportunity to recover the residual heat from the returned condensate, it is possible for condensate temperature to exceed 105 °C.

The flue gas temperature downstream of the pre-heater rises in line with the increase in the inlet temperature of the water in the feedwater pre-heater. Because flue gas is aggressive, the flue gas temperature should be kept at a temperature that is above the dew point. If the deaerator temperature is 125 ºC, the flue gas temperature downstream of the feedwater pre-heater is approx. 20 ºC higher than if the water inlet temperature is 105 ºC.

Rule:

Chimney loss increases by 1 % for every 20 ºC rise in flue gas temperature.
Preferably, deaerators should be operated at 105 ºC.

If this cannot be achieved because the temperature obtained by mixing the make-up water and the returned condensate is too high, the temperature should be 10 to 15 ºC above the mixed temperature. Here the condensate supplied at a higher pressure is not taken into account.

Deaerators connected in parallel

Where there are two deaerators which are operated in parallel on the water side (the feedwater pumps take water from both deaerators), the steam pressure in the deaerators should be equal. To do this, both deaerators must be connected on the steam side. If the deaerators are only connected via the feedwater pumps, the water level and pressure may fluctuate considerably.

The moment one deaerator is supplied with water, the internal pressure falls. The other deaerator then feeds water into the first deaerator via the suction pipe of the feedwater pumps. The process is then repeated in reverse order.

The coupling pipe between the steam spaces of the two deaerators should be generously proportioned. At 105 ºC the specific volume of the deaerator steam should be 1.4 m^3/kg. The diameter is also dependent on the size of the deaerator. If a pipe with DN 100 is used, the flow is only 200 kg/h of deaerator steam at a speed of 10 m/s.

Feedwater cooler

The lower the feedwater temperature when it enters the feedwater pre-heater, the greater its efficiency. The lower limit of the inlet temperature is determined on the basis of the point at which the water vapour in the chimney just fails to condense. For systems fired by natural gas, this point is at 70 ºC.

In general, a feedwater cooler in the feedwater pump suction pipe can ensure that the temperature of the water is lower. Feedwater is cooled down upstream of the pre-heater in this heat exchanger and the make-up water is heated up. Fig. 3-24 shows this layout.

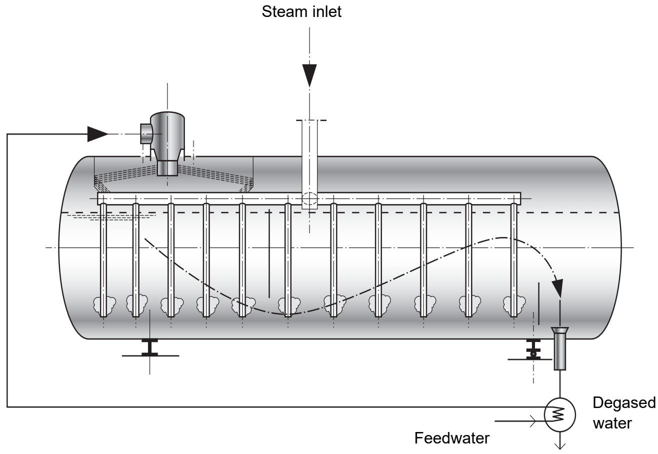


Fig. 3-24: Feedwater cooler

A heat audit should be carried out before such a measure is implemented. Here care should be taken to ensure that the maximum temperature of the mixture of make-up water and returned condensate is 95 °C, as it is supplied to the deaerator.

Cavitation in the feedwater pump

In order to prevent steam from forming in the suction pipe and consequently cavitation occurring in the feedwater pump, the deaerator should be fitted at least four metres above the pump inlet flange. Only then can it be ensured that there is sufficient upstream pressure in the suction pipe to prevent steam from forming in the event of a sudden fall in deaerator pressure.

This may be a helpful explanation. The suction line to the feedwater pump is filled with hot water that has a temperature of 105 ºC. If the pressure in the deaerator falls below 1.2 bar a, the water starts to boil at 105 ºC. This boiling effect also starts in the suction pipe and this causes the pump to fail. The risk of steam forming is particularly great at the point where the diameter of the suction pipe reduces upstream of the pump suction connection.

Vacuum breaker

If the steam boiler is taken out of service, the steam condenses and may create a vacuum in the deaerator. Under certain circumstances this vacuum causes water from the deaerator to draw water into the boiler (via the internal steam pipe). If the water is able to flow back to the steam system, for example, it creates a siphon effect. It is very easy to prevent this siphon effect by drilling a small hole in the internal steam pipe at the level of the steam space.

3.17 Continuous (TDS) and bottom blowdown process

Introduction

Live steam from the boiler is almost pure as a result of the evaporation process. Impurities that come from fresh water and returning condensate during the steam generation process remain in the boiler. The concentration of impurities increases during the evaporation process in the boiler, and the boiler water becomes more concentrated. Small quantities of boiler water are drawn off to examine the concentration in order to keep control of the permissible concentration.

In boiler plants two types of blowdown are common:

- ***Continuous (TDS) blowdown*** means that blowdown of boiler water takes place on a continuous basis to regulate the concentration of dissolved substances. The specified value depends on the operating pressure and is generally determined by consulting an expert. In the majority of cases, the pH value and conductivity of the boiler water is controlled manually.
- ***Bottom blowdown*** describes the process whereby the boiler water is blown down via a blowdown valve to remove the dirt, rust and silt that has collected at the bottom of the boiler. The sludge is produced by chemical binding of residual hardness with phosphates and other impurities, which are entrained in the condensate. Depending on the quality and colour of the boiler water, boiler water is blown down using one blowdown valve, once or twice per day for about 2 seconds.

If continuous (TDS) blowdown does not take place, the concentration of impurities could increase to the extent that the boiler pipes and walls become corroded. Another symptom of an increased concentration is that the water in the boiler may prime. This means that the water and the steam bubbles do not completely separate. This priming action may be so great that, apart from steam, boiler water may be lost from the boiler.

Theoretically, returning condensate is equivalent to distilled water and is almost pure. In practice, there are many reasons why condensate may be contaminated:

- Boiler water may be entrained with the steam. The most common causes are excessive TDS concentration or the water level is too high. In extreme cases, white stripes (chemical residues) can be seen on valves because contaminated steam has escaped at leakage points.
- In individual cases, the cause can also be a badly designed boiler. A correctly designed boiler produces approx. 95 % dry steam. This means that 5 % boiler water is entrained in the steam and is returned to the boiler with the returning condensate.
- Dregs of cooling water often remain in the pipes, particularly during processes where, for example, pipe coils are used for cooling and heating, and end up in the steam circuit together with the condensate. Some contaminants such as salts and chlorides are also found in fresh water, depending on which purification process is used (softening or demineralisation).

Continuous (TDS) blowdown

There are three options available for controlling the TDS values of boiler water during the continuous blowdown process:

- Manual control
- Automatic control
- Time control

a) Manual control

Depending on the results of the boiler water analysis, a greater or lesser amount of water is blown down via a special type of manual valve. This valve is configured internally to pass the boiler water over a number of stages. A proportion of the hot water flashes at each of the stages; this keeps the flow at a low speed and the valve has a longer service life. The valve stem has a coarse thread pitch. A pointer on the lever and a fixed show a percentage open, and the amount of water drained off can be determined using a nomogram.

b) Automatic control

The manual blowdown valve described above is equipped with an actuator. An electrode is installed in the boiler with where the conductivity of the boiler water is measured. The measured value is compared in the controller with the specified value (Fig. 3-25). If the specified value is exceeded, the continuous blowdown valve opens very slightly. If the figure is below the specified value, the continuous blowdown valve closes accordingly.

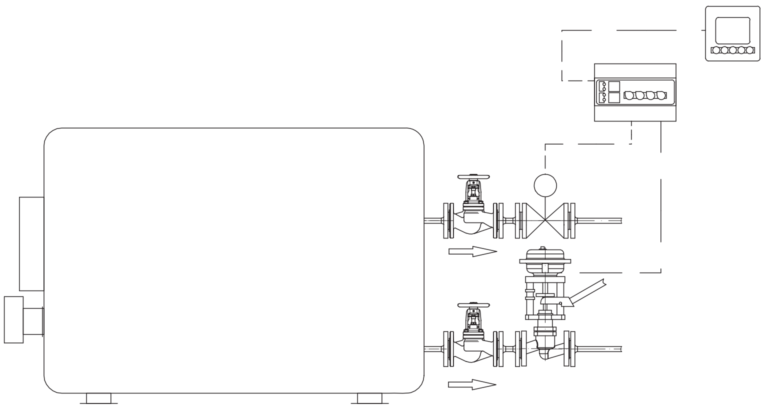


Fig. 3-25: Automatic continuous blowdown control

c) Time control

Where time control is used, blowdown no longer takes place continuously but instead it is triggered by a time switch or another signal via a circuit. This method is often used is continuous blowdown, during which the deaerator or the condensate tank is filled with fresh water.

Bottom blowdown

Because intermittent blowdown at the bottom of the boiler only takes place once or twice a day for about 20 seconds, it is not economically efficient to recover the heat contained. In general, the water discharged during the bottom blowdown process is directed to a blowdown tank, the flash steam is discharged and the remaining water of 100 ºC is cooled to 40 °C and discharged into the sewage system.

To prevent excessively hot blowdown water from entering the sewage system and possibly damaging the piping system, the hot water is collected in a tank (Fig. 3-26). The flash steam is then discharged into the atmosphere via a large pipe and the remaining water is cooled down to an acceptable temperature using cooling water.

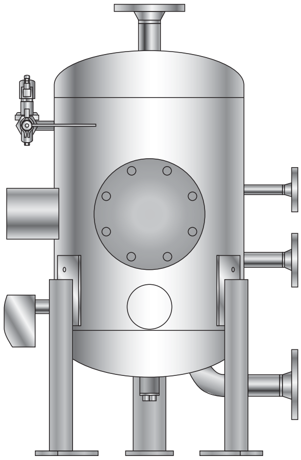


Fig. 3-26: Continuous and intermittent blowdown water tank

Warning:
If the blowdown process removes too much water and is carried out too often, it has a negative impact on fuel costs and also increases expenditure for water and chemicals.

Recovering heat from blowdown water

Heat contained in the continuous blowdown can be recovered in a heat exchanger.
The most common systems us the heat to heat up cold make-up water before it is supplied to the deaerator or the feedwater tank.

Unfortunately, the direct transfer of heat from the continuous blowdown water in the heat exchanger is not free of problems. Because the boiler water flashes after the continuous blowdown valve, the mixed water (flash steam/boiler water) enters the heat exchanger at high speed, which causes erosion at the base of the pipe. Leaks then often occur. This is reflected in the fact that the continuous blowdown water coolers are often out of action in many boiler houses. For this reason it is advisable to route the continuous blowdown water into a flash vessel. The flash steam should be routed into the pipe downstream of the pressure reducing valve, which controls the deaerator.

Heat transfer between the incoming feedwater and the rest of the continuous blowdown water (105 ºC) should take place in a heat exchanger (Fig. 3-27).

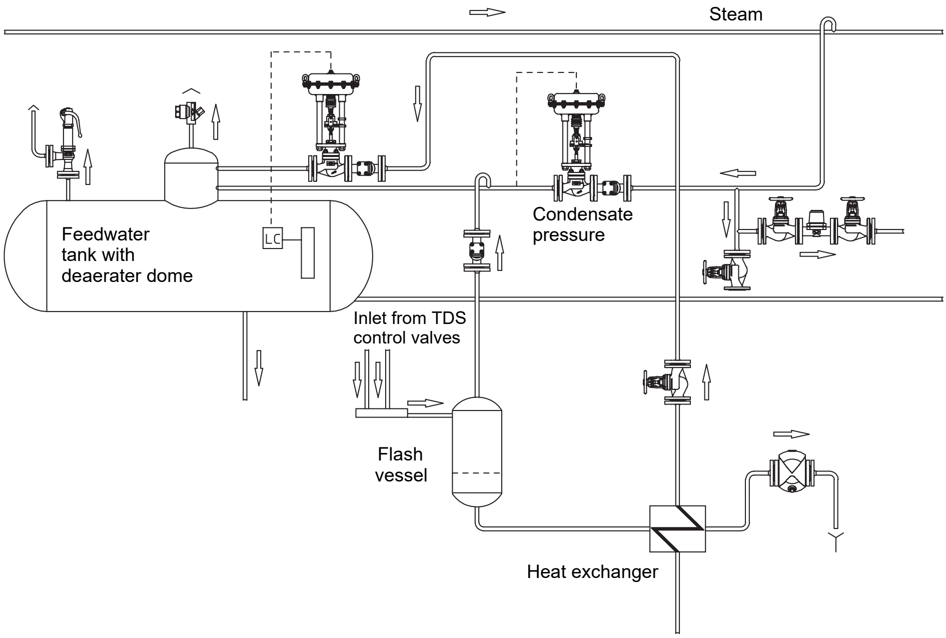


Fig. 3-27: Recovering heat from blowdown water

Continuous (TDS) blowdown flash vessel

Blowdown water from the boiler is directed to the flash vessel. The top (steam) connection is connected to the deaerator. This steam connection is 'T'ed into the steam pipe downstream of the deaerator pressure control valve. The blowdown water is drained off to the heat exchanger via a float-type steam trap positioned at a slightly higher point. The residual heat of the blowdown water is transferred to the incoming fresh water in the heat exchanger.

Bearing in mind that the aim is always to keep the heat exchanger filled with water, the discharge is via a swan neck connected at a higher point. A prerequisite for this system working correctly is modulating water level control system for the deaerator.

In order to prevent the blowdown water from entering the deaerator if the floating trap is faulty, the flash vessel should be equipped with a high water level alarm. This alarm signal can, if necessary, intervene in the function of the blowdown controller. When calculating the vessel diameter, the steam should not exceed a speed of 1 m/s. The speed in the discharging steam pipe must be kept at 10 m/s. The minimum diameter of the flash vessel is 250 mm.

Example:
The heat content of the water at 10 bar is 762 MJ/tonne. Bearing in mind that the heat from the blowdown water cannot be reused, the energy lost can be calculated as follows. If the blowdown amount is 4.5 % and the average boiler efficiency rate is 93 %, the figure is as follows:

$$\textit{Energy loss} = \frac{0.045 \times 0.762}{0.93} = 0.0369 \ \textit{GJ/tonne}$$

If the price of heat is €7.50 /GJ, €0.28 /tonne of steam generated are lost.

If average steam production is 50000 tonnes/year, the payback time on the capital employed is about 3 years if the heat in the water drained off is used and the heat of the blowdown water is, for instance, used to heat up fresh water to 40 ºC.

Calculation of the continuous blowdown percentage rate

In practice, there are two methods used as a basis for calculating the continuous blowdown percentage rate:

Method 1:
Calculation of the relationship between the chloride content in the feedwater and in the boiler water.

Method 2:
Calculation of the relationship between the chloride content in the make-up water and in the boiler water.

The resulting loss in kg is the same for both methods. Only the percentage rates and the starting points are different.

The resulting loss in kg is the same for both methods. Only the percentage rates and the starting points are different.

Definition of the continuous blowdown percentage rate as per method 1:

The formula is:

$$\text{Continuous blowdown percentage rate} = \frac{Cl_{sw}}{Cl_{kw}} \times 100\%$$

Cl_{fw} = Chloride content of feedwater

Cl_{bw} = Chloride content of boiler water

Example:
Cl_{fw} = 9 mg/l
Cl_{bw} = 200 mg/l

$$\textit{Continuous blowdown percentage} = \frac{9}{200} \times 100\% = 4.5\%$$

This means:

For each tonne of steam produced 1.045 tonnes of boiler water is supplied and 0.045 tonne of boiler water is blown down.

Definition of the continuous blowdown percentage rate as per method 2:

The formula is:

$$\text{Continuous blowdown percentage rate} = \frac{Cl_{zw}}{Cl_{kw}} \times 100\ \%$$

Cl_{mw} = Chloride content of make-up water

Cl_{bw} = Chloride content of boiler water

Example:
Cl_{mw} = 18 mg/l
Cl_{bw} = 200 mg/l

$$\textit{Continuous blowdown percentage} = \frac{18}{200} \times 100\ \% = 9\ \%$$

It can be deduced from the relationship between Cl_{fw} and Cl_{mw} or 9 and 18 mg/l, respectively, that the feedwater contains approximately 50 % fresh water in this case. Per tonne of fresh water, 1.09 tonnes of water are supplied and 0.09 tonnes are blown down.

A loss of 0.09 x 0.5 = 0.045 tonne remains per tonne of steam produced.

Since both calculations relate steam production to feedwater, it is possible that small deviations can occur.

Sampling

Samples of boiler and feedwater should always be taken via a sample cooler. If this procedure is not followed, the sample will be defective. The boiler water sample of a 10 bar boiler flashes and therefore loses approximately 16 % of the sample.

A rule of thumb for flashing is this. The saturation temperature at 10 bar is 180 ºC, so flashing is (180 - 100) x 0.2 = 16 %. This means that the concentration of the constituents in the sample is 16 % higher than for a cooled sample. It is also unsafe.

Boiling

Steam bubbles form in the boiler water and, because they have a lower density, they are buoyant. As a result of this, the bubbles rise. The closer the steam bubbles get to the water surface in the boiler, the greater their speed. At the point where the bubbles break the surface, they are moving at such a high speed that water is entrained.

If a good steam separator is fitted in or on the boiler, the entrained water is removed. The separator must be properly drained to avoid water hammer and corrosion.

4.0 Pipes

4.1 Introduction

Steam and condensate pipes play an important role in industrial processes and boiler houses. They transport the steam and condensate to the processes or the heat exchangers and back again. Strict requirements have to be complied with regarding the laying and sizing of the pipes. There are so many factors to be taken into account that it is advisable to leave this design work to an expert.

The scope of this book is not adequate to cover every single aspect in detail. This chapter provides a rough overview of pipe-related topics in day-to-day practice.

4.2 Standards

Nominal diameters

The nominal diameter or size of a pipe, flange or valve is specified as a DN (standard diameter) value. For clarity, nominal diameters are used for calculations in this section. The connecting dimensions of a DN 40 valve exactly fit the connection flanges of a DN 40 pipe. The most popular pipe sizes are classified in a DIN EN 10220 series.

Nominal pressure ratings

Steam pipes are usually made of seamless steel. The choice of material and the pipe wall thickness are determined by the steam pressure and temperature. Each pipe has a nominal pressure rating (PN). This rating specifies the maximum permissible working pressure at 20 ºC. The most widespread pressure ratings are PN 10, 16, 25, 40, 63, 100, 160, 250, 320, etc. The same pressure ratings are also used for all valves. PN 40 generally means that the part will withstand a maximum pressure of 40 bar at 20 ºC. In addition to the pressure, the choice of material is also influenced by the process temperature. High temperatures have an effect on the material thickness required.

Kvs-value

As a rule, DN 40 does not necessarily mean that control or other valves has a 40 mm outlet. The Kvs-value, which provides a means of defining pressure drop, was introduced for this reason. The Kvs-value of a valve is defined as follows: the flow of water at 20 °C in m^3/h and a density of 1ς, which gives a pressure drop of 1 bar.

Other standards

DIN EN standards are the norm in Germany. DIN stands for "Deutsches Institut für Normung" (German Institute for Standardisation) while EN denotes a recognised European standard. ANSI and ASME standards are also common in certain branches of the process industry (petrochemicals, offshore and chemical plants).

4.3 Determination of the pipe diameter

Amongst other things, the pressure drop in a pipe is dependent on the volume flow, the flow velocity and the viscosity of the fluid. The more steam flows through a pipe with a particular nominal diameter, the higher the friction on the pipe wall. In other words, the higher the steam velocity, the greater the friction against the pipe wall, and the greater the pressure drop. If a pipe is used to transport superheated steam to a steam turbine, the pressure loss should be as small as possible. However, this kind of pipe is more expensive than ordinary pipes; a larger diameter immediately puts up the cost dramatically. The investment calculation is based on the payback period for the investment amount compared to the profit from the turbine output.

This calculation is always based on the peak load of the turbine rather than to the average load. If 1000 kg of steam is discharged during a 15-minute period at peak load, for instance, the pipe must have a capacity of 60/15 x 1000 = 4000 kg/h.

Calculation

Chapter 6.0 Condensate Management explains how to determine the diameter of a condensate pipe. The calculations for steam, air and water pipes rest on practically the same assumptions. These calculations are therefore described in the following in order to round off the topic.

The fundamental formula for calculating the diameter is as follows:

$$Q = \frac{1}{4}\pi \times D^2 \times v$$

where:

Q = Steam, air or water flow rate in m^3/s

D = Diameter of the pipe in m

v = Maximum velocity in m/s

In practice, it is advisable to specify the flow rate in m^3/h and the pipe diameter in mm. The above formula must therefore be adapted as follows in order to calculate the required diameter:

$$D = \sqrt{\frac{354 \times Q}{v}}$$

where:

D = Diameter of the condensate pipe in mm

Q = Flow rate in m^3/h

v = Maximum velocity in m/s

Pipe calculations are always based on the volume flow (m^3/h) rather than the mass flow (kg/h). If only the mass flow of the steam is known, the steam tables must be consulted in order to convert from kg/h to m^3/h via the specific volume.

Example:
The specific volume of saturated steam at 11 bar is 0.1747 m^3/kg. The volume flow of 1000 kg/h of saturated steam at 11 bar is therefore 1000 x 0.1747 = 174.7 m^3/h. If the same amount of superheated steam is present at 11 bar and 300 °C, the specific volume is 0.2337 m^3/kg and the volume flow 233.7 m^3/h. In other words, a steam pipe that is suitable for transporting saturated steam cannot necessarily be used to carry the same amount of superheated steam.

The pressure is additionally required to calculate air and other gases. Compressor manufacturers specify compressor capacities in m_o^3/h, which means atmospheric m^3 at 0 °C.

If the compressor capacity is 600 m_o^3/h and 6 bar compressed air is used, the volume flow is 600/6 = 100 m^3/h; this is also the basis for the pipe calculation.

Maximum flow velocity

The maximum flow velocity in a pipe system is influenced by several factors.

- Plant costs: a low velocity means a large diameter.
- Pressure loss: a high velocity means a small diameter and more pressure loss.
- Wear: a high velocity means more erosion, especially with condensate.
- Noise: a high velocity means more noise, e.g. due to steam pressure reducing valves.

The table below shows the recommended flow velocities for various media.

Medium	Function	Velocity in m/s
Steam	Less than 3 bar	10 - 15
	3 - 10 bar	15 - 20
	10 - 40 bar	20 - 40
Condensate	Filled with condensate	2
	Condensate-steam mixture	6 - 10
Feedwater	Suction pipe*	0.5 - 1
	Discharge pipe	2
Water	Drinking water	0.6
	Cooling water	2
Air	Compressed air	6 - 10
* Suction pipe of the feedwater pump: the low velocity means a smaller pressure drop, so that cavitation is avoided at the inlet of the feed pump.		

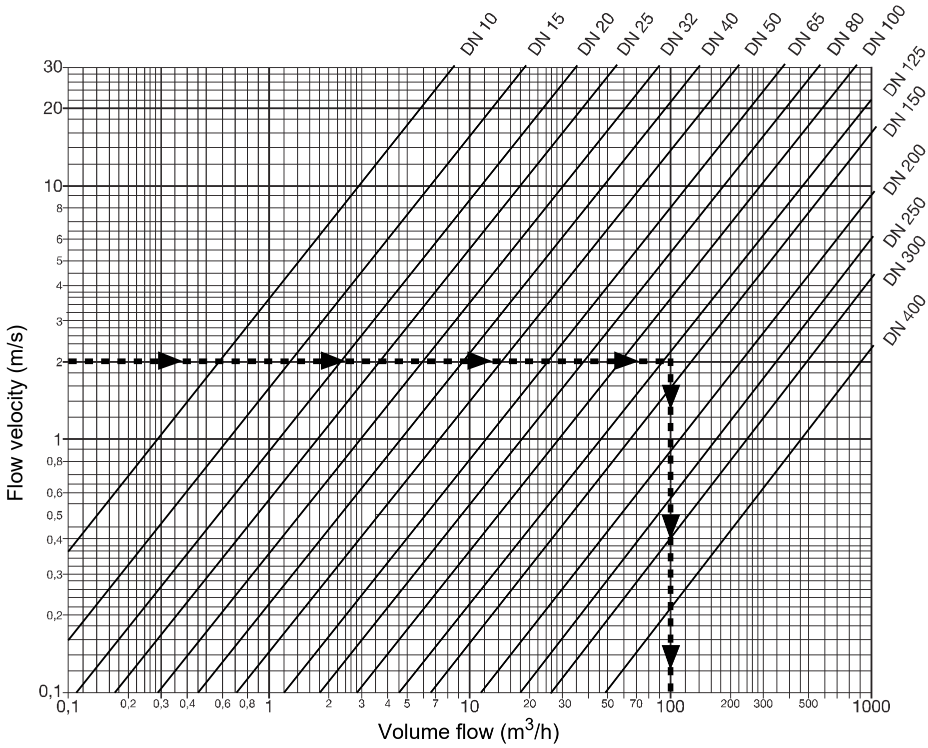


Fig. 4-1: Recommended flow velocities

Examples:

a) Water

Calculation of the pipe diameter for 100 m^3/h of water at v = 2 m/s.

$$D = \sqrt{\frac{354 \times 100}{2}} = 133\ mm\ .\ \textit{Selected nominal diameter: DN 125 or DN 150.}$$

b) Compressed air

Calculation of the pipe diameter for 600 m$_o^3$/h of compressed air at 5 bar and a velocity of 8 m/s.

Conversion from 600 m$_o^3$/h to actual m^3/h: $\frac{600}{5} = 120 m^3 / h$.

$$D = \sqrt{\frac{354 \times 120}{8}} = 72\ mm\ .\ \textit{Selected nominal diameter: DN 65 or DN 80.}$$

The decision in favour of DN 65 or DN 80 is determined by the purpose for which the water or air is intended. It should be noted that the diameter calculation is based on the mean value and takes no account of sporadic peak loads.

c) Saturated steam

Calculation of the pipe diameter for 1500 kg/h of saturated steam at 16 bar and a velocity of 15 m/s.

According to the steam table, the specific volume of 16 bar saturated steam v_g is 0.1237 m^3/kg.

$$D = \sqrt{\frac{354 \times 1500 \times 0.1237}{15}} = 66 \text{ mm}.$$

Once again, the decision for DN 65 or DN 80 depends on the potential peak loads. Future developments may also have to be considered.

d) Superheated steam

If the steam in this example is superheated to 300 °C, the specific volume changes to v_g = 0.1585 m^3/kg.

$$D = \sqrt{\frac{354 \times 1500 \times 0.1585}{15}} = 75 \text{ mm}, \text{ in other words DN 80.}$$

The nomogram in Fig. 4-9 (Page 76) shows how to determine the pipe diameter for water or air graphically without calculations. The nomogram in Fig. 4-10 (Page 77) shows this process for saturated steam and superheated steam.

e) Condensate

If the pipe is a condensate pipe without any flash steam, the diameter calculation is identical to that for water.

If hot condensate is transported in a condensate pipe after passing through the steam trap, this condensate is flashed. Chapter 6.0 Condensate Management explains how to read off the percentage of flash steam.

Calculation rule:

Percentage of flash steam = (temperature upstream of the steam trap minus temperature downstream of the steam trap) x 0.2. The volume of the flash steam is required to calculate a condensate pipe.

The volume of the residual water is so small in relation to the volume of the flash steam that it can be disregarded.

Calculation of the diameter of a condensate pipe with 1000 kg/h of condensed steam at 11 bar (h_f = 781 J/kg), flashing to the 4 bar condensate system (h_f = 604 kJ/kg, v_g = 0.4622 m^3/kg and h_{fg} = 2133 kJ/kg).

The percentage of flash steam is as follows: $\frac{781 - 604}{2133} \times 100\% = 8.3\%$

The flash steam flow rate is as follows: 1000 x 0.083 = 83 kg/h or 83 x 0.4622 = 38 m^3/h.

The percentage of flash steam by volume is approximately 97 %.

The pipe diameter for the mixture at a velocity of 8 m/s is as follows:

$$D = \sqrt{\frac{354 \times 1000 \times 0.083 \times 0.4622}{8}} = 40 \text{ mm}$$

The percentage of flash steam for an atmospheric condensate system (v_g = 1.694 m^3/kg) is as follows:

$$\frac{781 - 418}{2258} \times 100\% = 16\% \text{ or } 160 \text{ kg/h}$$

In this case, the pipe diameter is as follows:

$$D = \sqrt{\frac{354 \times 1000 \times 0.16 \times 1.694}{8}} = 110 \text{ mm}$$

Velocity of the mixture

There is some disagreement amongst experts when it comes to the maximum permissible flow velocities for mixtures of flash steam and condensate. Velocities between 15 and 20 m/s are often mentioned in the literature. Based on practical experience, however, these values are far too high. The reason for this is that a condensate pipe transports 95 percent flash steam by volume and 5 percent water. The mixture has a velocity of 20 m/s. Drops of water that are entrained in the flash steam hit the first bend in the pipe at the same velocity as this steam (20 m/s, equivalent to 72 km/h!). A small amount of condensate collects at this point. The free flow is impaired as a result of this backing-up. The condensate gradually builds up, then suddenly shoots through to the next pipe bend as a plug, etc.

If a leak occurs in a condensate pipe, it is almost certain to be at a bend. The thin area in the bend is situated upstream at three-quarters of the elbow.

In addition, the velocity of 72 km/h is based on continuous flow, even though all steam traps – with the exception of float-type traps – operate intermittently. It is not uncommon for a steam trap (especially a bimetallic type) to remain shut for more than half the time!

Actual velocities of 100 to 140 km/h are nothing unusual in a condensate system.

Practice has shown that condensate drainage problems are usually attributable to pipe diameters that are too small and only rarely to the type of steam trap selected. A velocity of 6-8 m/s is recommended in the condensate pipe downstream of a steam trap.

4.4 Expansion of steam pipes

Introduction

When steam pipes are designed, a certain expansion must be allowed for owing to the temperature variations when starting up or shutting down the plant. The coefficient of expansion for steel is 0.012 mm/m ºC. This means that the pipe expands 0.012 mm per metre length for every one-degree rise in temperature. If a 10 bar saturated steam pipe with a length of 25 m is heated from 15 ºC to 180 ºC, it will expand (180 - 15) x 0.012 x 25 = 50 mm.

A superheated steam pipe at 450 ºC installed between the steam boiler and the turbine expands 5.4 mm per metre length. The forces of expansion are so enormous that the turbine would be literally uprooted from its foundations if nothing was done to accommodate the expansion.

Every material has a unique coefficient of expansion: copper, for example, has a coefficient of 0.016 mm/m ºC while stainless steel has a coefficient of 0.019 mm/m ºC.

Compensating expansion

In the absence of suitable measures to compensate for expansion, considerable stresses and forces will act in or on the apparatus, pipes and valves. The pipes can be distorted to such an extent that they break away from their points of support.

Pipes must be laid so that they can expand or contract freely as they heat up and cool down. Steps must also be taken to prevent axial movement. The risks arising from inexpert installation tend to be very high and it is therefore advisable to trust the planning of your pipe system to a qualified specialist.

A few general recommendations for planners of pipe systems are provided in the following.

- A straight pipe section must never be installed between two anchors (refer to Fig. 4-2).

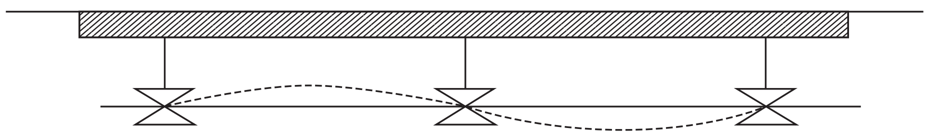


Fig. 4-2: Pipe with anchors only

Pipes should be installed so that there is only ever an anchor (fixed point) on one side; all other supports should be sliding (refer to Fig. 4-3).

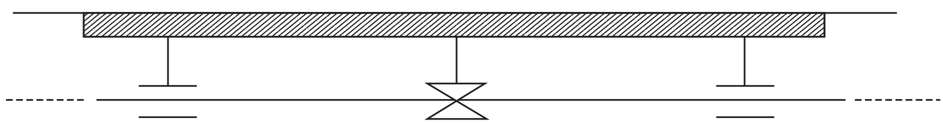


Fig. 4-3: One anchor point, all other supports sliding

- If a pipe is installed with a 90º bend, both sections must be able to expand freely. The distance between the anchor and the bend must be sufficient to accommodate the expansion (refer to Fig. 4-4).

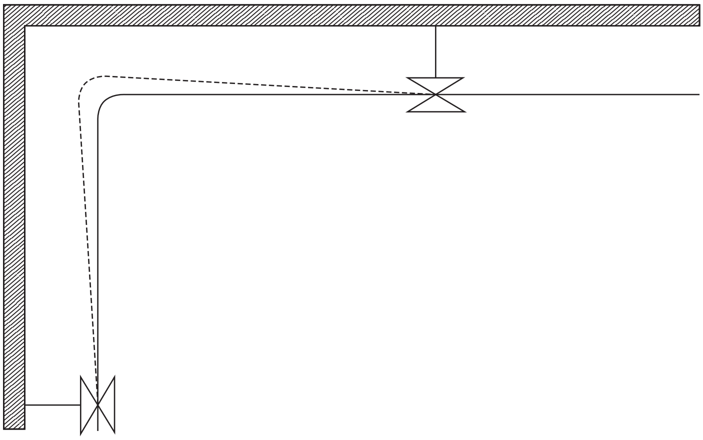


Fig. 4-4: Design with a 90º bend

- If a pipe is installed with branches, the anchors in the main pipe and in the branches must be located relatively close together. Branches in steam pipes must exit on the top side. Similarly, condensate pipes must enter on the top of the return pipe (refer to Fig. 4-5).

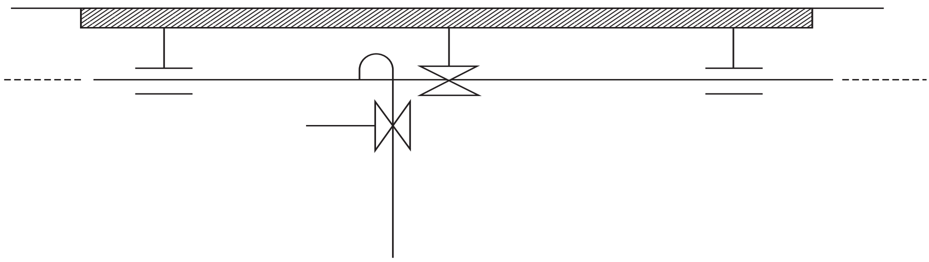


Fig. 4-5: Design with branches

- Fig. 4-6 shows a few designs of anchor and sliding points.

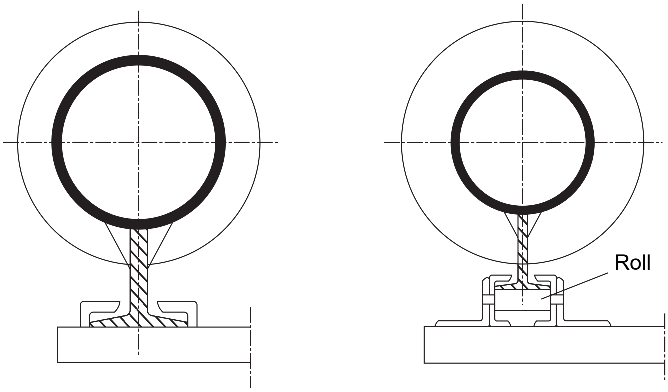


Fig. 4-6: Anchor and sliding point designs

If there is not enough room to expand in the longitudinal direction, expansion loops or bellows can be used (refer to Fig. 4-7). These loops are usually right-angled, although lyre loops are sometimes also used.

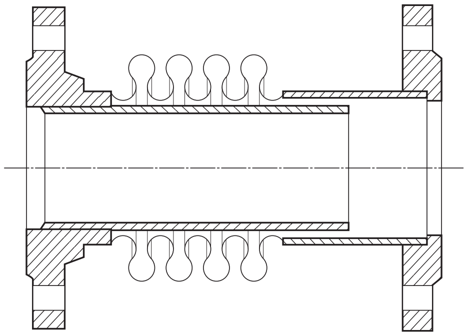


Fig. 4-7: Bellow Compensator

Assuming there is sufficient space, the expansion loops should be mounted horizontally. If not, they should be fitted vertically – with a drain point either ahead of, or preferably in, the rising pipe bend. The drain point consists of a drain port with a steam trap (refer to *Chapter 5.0 Pipe Drainage*). It is important to make sure that the expansion of the main pipe is not inhibited by the condensate pipe.

4.5 Heat loss via pipe supports

It has become standard practice in the last few years to weld roller supports to a pipe. However, this solution ignores the fact that a relatively large amount of heat is lost via welded rollers. As a result, the average temperature of the roller supports is only 30 to 40 ºC less than the steam temperature in the pipe!

It is therefore advisable to design the supports in a new plant with brackets. A glass fibre mat should then be inserted between each bracket and the pipe.

4.6 Heat transmission losses in uninsulated pipes and valves

The heat transmission losses in uninsulated indoor and outdoor pipes are given in tables for various temperature differences (between the ambient and process temperatures).

Rule:

An uninsulated valve losses the same amount of heat as 2 metres of uninsulated pipe with the same diameter.

Heat transmission losses in kW/m for uninsulated indoor pipes								
DN	**70 ºC**	**100 ºC**	**150 ºC**	**200 ºC**	**250 ºC**	**300 ºC**	**350 ºC**	**400 ºC**
25	0.08	0.10	0.15	0.20	0.40	0.60	0.75	1.00
40	0.15	0.20	0.35	0.45	0.75	0.90	1.30	1.50
50	0.17	0.25	0.40	0.60	0.90	1.20	1.60	2.00
80	0.27	0.35	0.65	0.90	1.30	1.75	2.25	2.80
100	0.35	0.40	0.75	1.10	1.50	2.00	2.75	3.50
150	0.45	0.55	1.10	1.60	2.25	3.00	4.50	5.60
200	0.60	0.80	1.50	2.20	3.30	4.50	6.00	7.80
250	0.75	1.00	1.80	2.80	4.00	5.50	7.00	9.80
300	0.90	1.20	2.20	3.20	5.00	6.80	9.00	11.50
400	1.00	1.60	3.00	4.50	6.50	9.00	12.00	15.50
The specified heat losses are rounded down.								

Fig. 4-8: Uninsulated indoor pipes

Example:
An uninsulated DN 150 pipe installed in a building loses 1.6 kW of heat per metre length if there is a temperature difference of 200 ºC. A DN 150 valve in the same pipe system loses 2 x 1.6 = 3.2 kW.

Based on a heat generating efficiency of 90 % and 8000 hours operation annually, the valve losses heat equivalent to:

$$\frac{8000 \times 3.2 \times 3.6}{31.65 \times 0.9} = 3250\ m^3.$$

Assuming a gas price of €0.20 /m^3, this represents a yearly loss of €650.- .

Since it costs around €240.- to insulate a valve, the investment pays off in approximately six months!

Heat transmission losses in kW/m for uninsulated outdoor pipes							
DN	**100 °C**	**150 °C**	**200 °C**	**250 °C**	**300 °C**	**350 °C**	**400 °C**
25	0.4	0.6	0.8	1.0	1.4	1.8	2.8
40	0.5	0.9	1.3	1.7	2.0	2.8	4.0
50	0.7	1.2	1.7	2.2	2.7	3.7	5.3
80	1.2	1.9	2.7	3.5	4.3	5.4	7.6
100	1.5	2.5	3.2	4.1	5.3	6.5	8.3
150	2.1	3.2	4.5	5.8	7.4	9.0	11.5
200	2.8	4.1	6.0	7.2	9.5	12.0	14.5
250	3.5	5.0	7.0	9.0	11.5	14.0	17.3
300	3.8	5.8	8.0	10.5	13.2	16.4	20.0
400	4.8	7.2	10.0	13.2	16.9	20.0	23.5
The specified heat losses are rounded down.							

Example:
An uninsulated DN 150 pipe installed in the open loses 4.5 kW of heat per metre length if there is a temperature difference of 200 ºC. A DN 150 valve in the same pipe system loses 2 x 4.5 = 9 kW. Based on a heat generating efficiency of 90 % and 8000 hours operation annually, the valve loses heat equivalent to:

$$\frac{8000 \times 9 \times 3.6}{31.65 \times 0.9} = 9100\ m^3 \text{ of natural gas.}$$

Assuming a gas price of €0.20 /m^3, this represents a yearly loss of €1820.- .

The saving that can be achieved by insulating the pipe is about 80 % – in this case 0.8 x €1,820.- = €1,450.- .
Since it costs around €240.- to insulate a valve, the investment pays off in two months!

Determination of the nominal diameter of water and air pipes

Example (refer to Fig. 4-9).

a) Water:
Pipe for 100 m^3/h with a flow velocity v = 2 m/s.
Result: DN 125.

b) Compressed air:
Pipe for 600 m^3/h with an air pressure of 6 bar and a flow velocity v = 8 m/s.
600 m^3/h is equivalent to 600/6 = 100 actual m^3/h.
Result: DN 65.

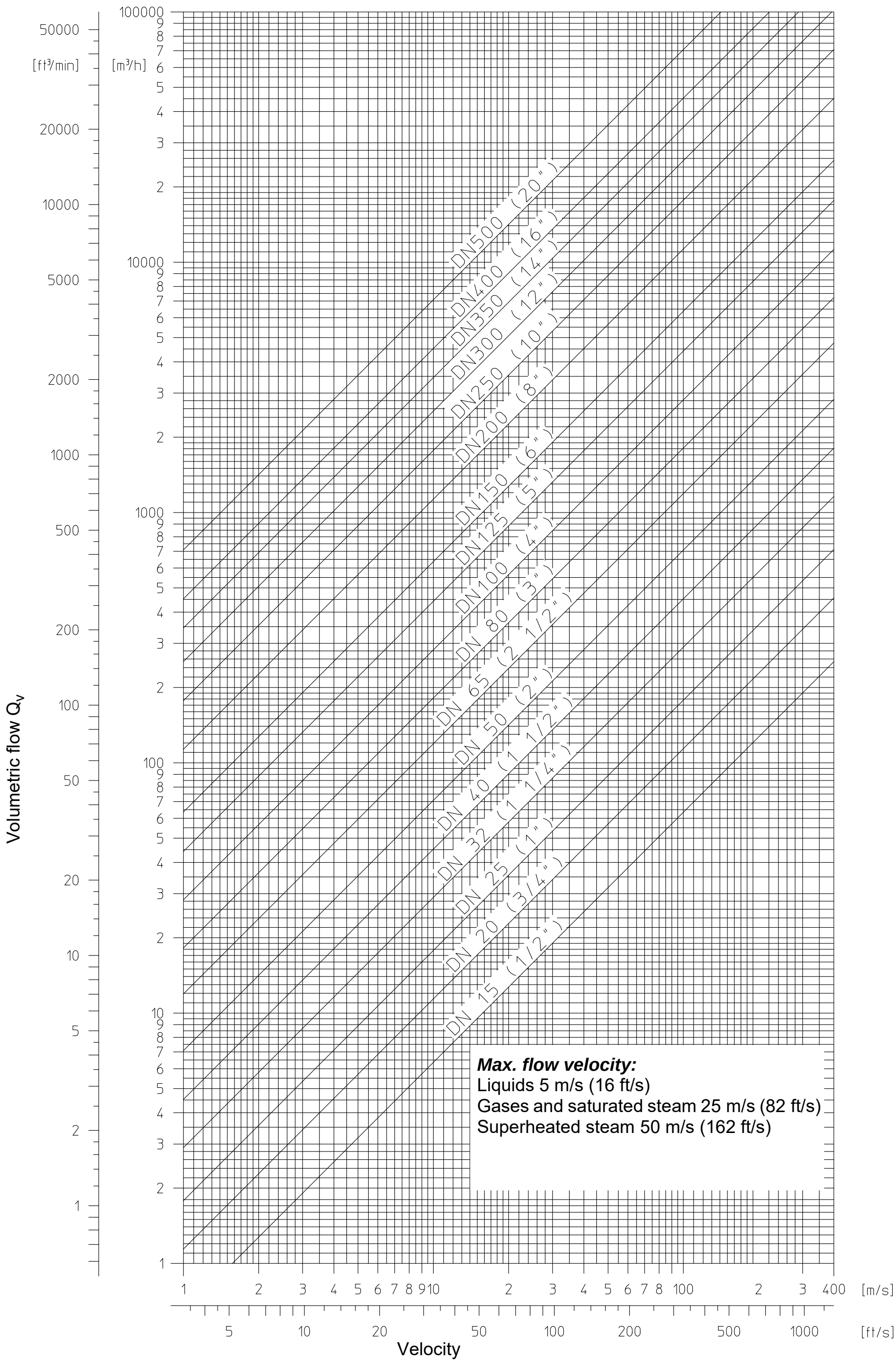


Fig. 4-9: Nomogram for determining the nominal diameter of water and air pipes

Determination of the nominal diameter of, and flow velocity for, steam pipes

Fig. 4-10 can be used to determine the flow velocity for a steam pipe based on the steam temperature and pressure as well as the nominal diameter. Conversely, the nominal diameter of the steam pipe can be derived from the nomogram and the flow velocity.

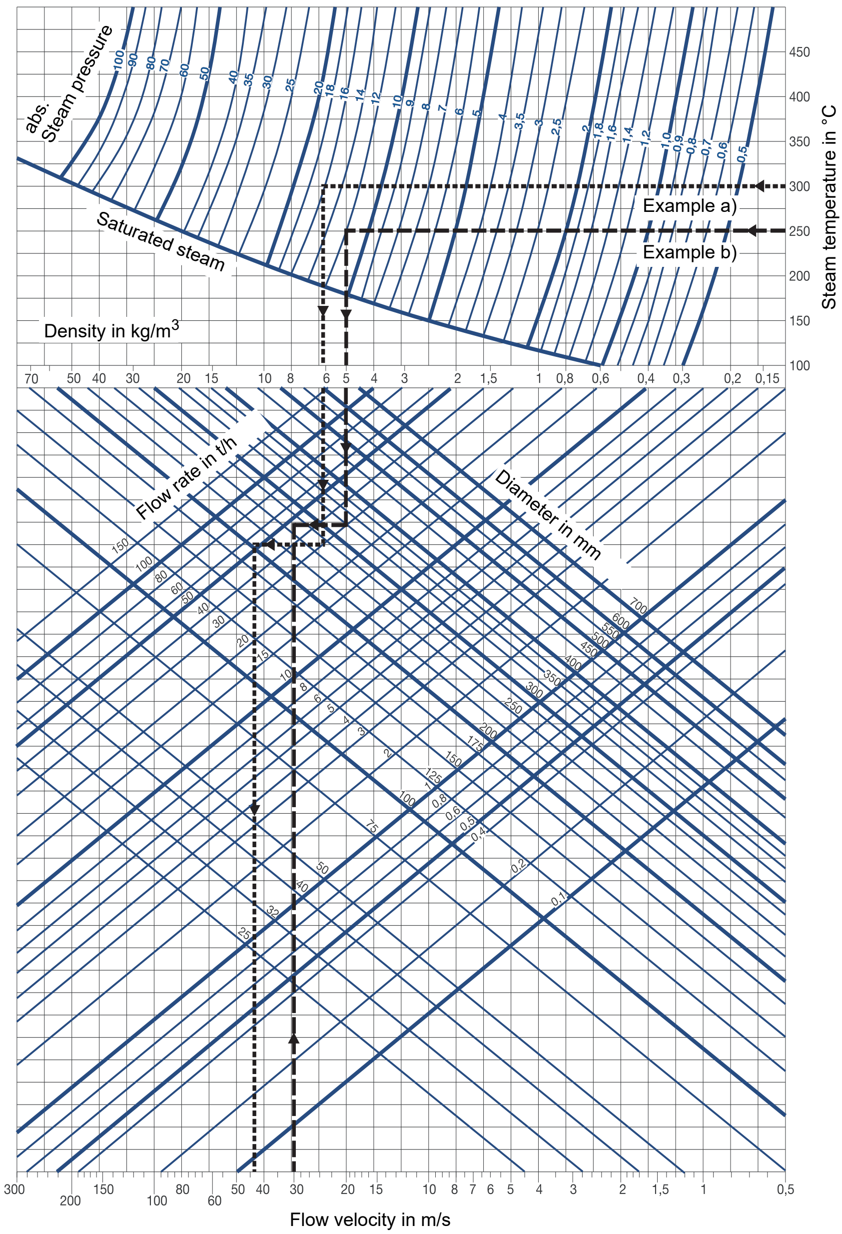


Fig. 4-10: Nomogram for determining the nominal diameter of, and flow velocity for, saturated steam pipes

a) Determination of the flow velocity (example):

Superheated steam, 30 t/h, 300 °C, 16 bar, DN 200.

Follow the dashed line horizontally from 300 °C up to the 16 bar line. Then continue vertically downward from this point as far as the 30 t/h flow rate line and from there leftward to the DN 200 nominal diameter line. Next, continue vertically to the flow velocity lines and read off the velocity at 43 m/s.

b) Determination of the nominal diameter (example):

Superheated steam, 30 t/h, 250 °C, 12 bar, flow velocity 30 m/s.

Follow the dashed line horizontally from 250 °C up to the 12 bar line. Then continue vertically downward from this point as far as the 30 t/h flow rate line. Plot a horizontal line to the left and right, then starting at 30 m/s on the flow velocity scale follow a vertical line until it intersects the plotted horizontal line. The nominal diameter can be read off at the point of intersection: DN 300.

For saturated steam, start at the pressure curve in the top part of the nomogram and follow this curve up to the saturated steam curve, then continue downward to the bottom part of the nomogram.

Determination of the nominal diameter of condensate pipes

The nominal diameter of condensate pipes can be determined directly using the table in Fig. 4-11.

Pressure (bar abs.)	**Pressure at the end of the condensate pipe (bar absolute)**												
	1	**2**	**3**	**4**	**5**	**6**	**7**	**8**	**9**	**10**	**12**	**15**	**18**
1													
2	15.4												
3	19.5	8.9											
4	22.2	11.8	6.5										
5	24.2	13.7	8.7	5.1									
6	25.0	14.5	9.6	6.2	4.9								
7	25.7	15.1	10.2	7.0	4.4	2.6							
8	28.2	17.3	12.5	9.4	7.1	5.9	4.9						
9	28.7	18.2	13.3	10.3	8.1	6.7	5.9	2.9					
10	30.0	18.9	13.9	10.9	8.7	7.3	6.5	4.0	2.6				
12	31.4	20.1	15.0	12.0	9.8	8.6	7.6	5.5	4.4	3.4			
15	33.2	21.6	16.4	13.2	11.1	9.8	8.8	6.8	5.9	5.1	3.5		
18	34.6	22.8	17.5	14.3	12.1	10.7	9.8	7.9	7.0	6.2	4.8	3.1	
20	35.4	23.4	18.1	14.9	12.7	11.3	10.3	8.6	7.6	6.8	5.4	3.7	2.1

Fig. 4-11: Determination of the nominal diameter of condensate pipes

To determine the nominal diameter of a pipe using the value obtained with the table, this value must be multiplied by the corresponding flow rate factor in the table below.

kg/h	**100**	**200**	**300**	**400**	**500**	**600**	**700**	**800**	**900**	**1000**	**1500**	**2000**	**3000**
Factor	1.0	1.4	1.7	2.0	2.2	2.4	2.6	2.8	3.0	3.2	3.9	4.5	5.5

The table is based on the formula explained in *Chapter 6.0 Condensate Management* as well as on a flash steam flow velocity of 10 m/s and a condensate flow rate of 100 kg/h.

$$D = \sqrt{\frac{354 \times Q}{v}}$$

where:

D = Diameter of the condensate pipe in mm

Q = Flow rate in m^3/h

v = Maximum velocity in m/s

Example 1:

Condensate flow rate: *500 kg/h*

Steam pressure: *4 bar*

Back pressure: *atmospheric (1 bar)*

Nominal diameter: *?*

In Fig. 4-11, look up the value for 4 bar steam pressure in the 1 bar back pressure column = 22.2 bar (abs). Read off the factor 2.2 under 500 kg/h in the bottom table. Multiply the value 22.2 by the factor 2.2. Result: 22.2 x 2.2 = 48.8 mm, in other words DN 50. If the required velocity is less than 10 m/s, e.g. 5 m/s, the calculated diameter must be multiplied by $\sqrt{10/5} \approx 1.4$ *.*
In this case, DN 50 becomes DN 80, for example.

Example 2:

Condensate flow rate: *1500 kg/h*

Steam pressure: *9 bar*

Back pressure: *2 bar in the condensate system*

Nominal diameter: *?*

In Fig. 4-11, look up the value for 9 bar steam pressure in the 2 bar back pressure column = 18.2 bar (abs). Read off the factor 3.9 under 1500 kg/h in the bottom table.

Multiply the value 18.2 by the factor 3.9. Result: 18.2 x 3.9 = 71 mm. Depending on the conditions on site (long pipe, varying peak load), select either DN 80 or – if the pipe is short with a constant condensate flow rate – DN 65.

5.0 Pipe Drainage

5.1 Introduction

If steam condenses in a pipe, the condensate must be discharged immediately it is formed for the following reasons:

- Condensate in steam pipes has disastrous consequences for turbine blades.
- Condensate in steam pipes leads to heat loss.
- Condensate in pipes results in corrosion.
- Condensate in steam pipes leads to erosion, for example at pipe bends.
- Condensate is a cause of water hammer in steam pipes.

Drainage is essential to remove condensate from pipes. This applies not only when a steam pipe system is actually operating but also during start-up. This chapter describes suitable drainage measures as well as the valves required to implement them. Further information about the consequences of water hammer can be found in the *Chapter 6.0 Condensate Management*.

5.2 Drainage of steam pipes

The technical design of steam pipes requires a continuous downward gradient (approx. 0.5 %) in the direction of flow. The drain points in a properly insulated pipe should be a maximum of 75 metres apart. An additional drain point must be provided before the start of each rising pipe section. It is not sufficient to discharge the condensate simply by installing a tapping in the main pipe. A DN 200 steam pipe with a DN 25 tapping cannot be drained because most of the condensate is entrained beyond the tap.

A drain pocket represents the only way to drain a steam pipe reliably. Fig. 5-1 shows a drain pocket in a straight pipe, while Fig. 5-2 depicts a rising pipe bend.

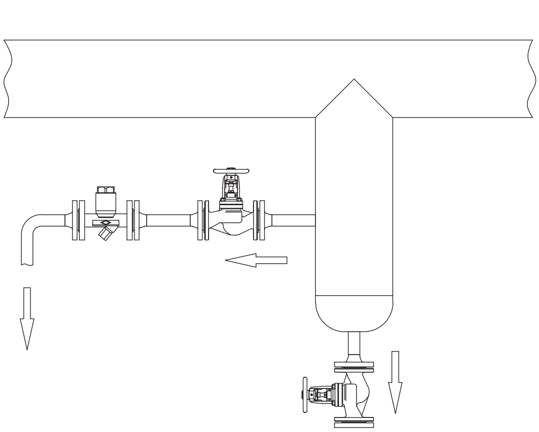


Fig. 5-1:
Drainage in a straight pipe

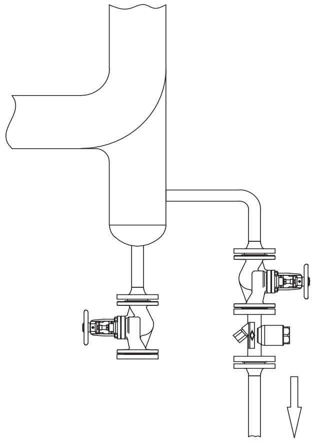


Fig. 5-2:
Drainage in a bend

The dimensions of the drain pocket should be determined as shown in Fig. 5-3.

$D_2 = 0{,}75\ D_1$
$H = 2{,}5\ D_1$
(up to max. 500 mm)

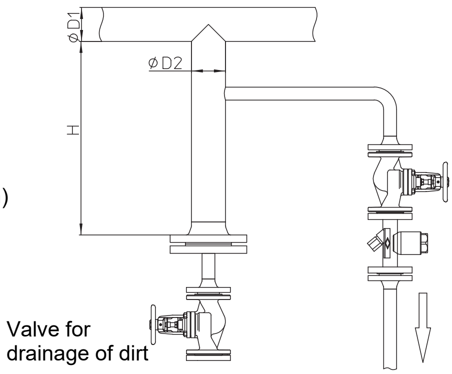


Fig. 5-3: Dimensions of a drain pocket

The steam trap must be connected approximately 50 mm above the bottom and mounted on the side of the pocket. The bottom of the drain pocket then acts as a strainer that protects the steam trap against debris in the pipe. Drain pockets with a continuous blowdown device have the advantage that significant amounts of debris can be removed very easily. For simple drainage (Fig. 5-4), the steam trap should be mounted in the branch and the manual drain point in the vertical pipe section. In this case, the straight pipe acts as a strainer.

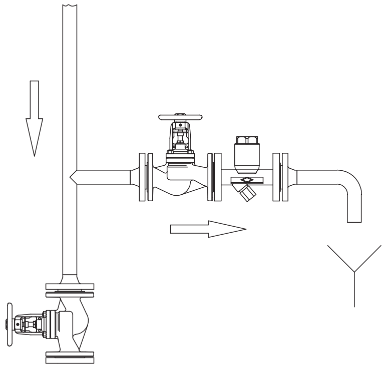


Fig. 5-4: Simple drainage with strainer

If the end of a pipe needs to be drained, a pipe bend is used as the drain port (Fig. 5-5). Instead of a pipe bend, the steam trap can also be mounted directly to the end flange. It is important to ensure that the trap is connected level with the bottom of the pipe rather than centrally.

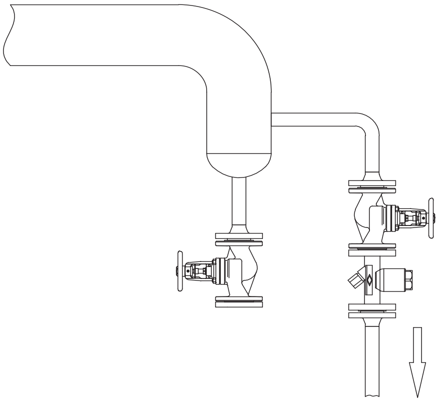


Fig. 5-5: Drainage at the end of a pipe

Particularly in critical equipment, separators are often used to prevent condensate from entering the equipment. Steam jet compressors and vacuum pumps are especially sensitive to entrained condensate drops. Fig. 5-6 shows a cyclone separator. The water drops are spun against the wall by the centrifugal force and collect at the lowest point, where the steam trap is located.

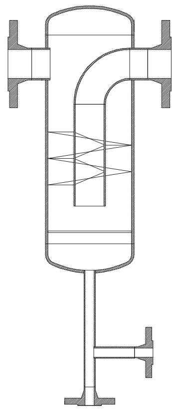


Fig. 5-6: Cyclone separator

Fig. 5-7 shows several typical drain points.

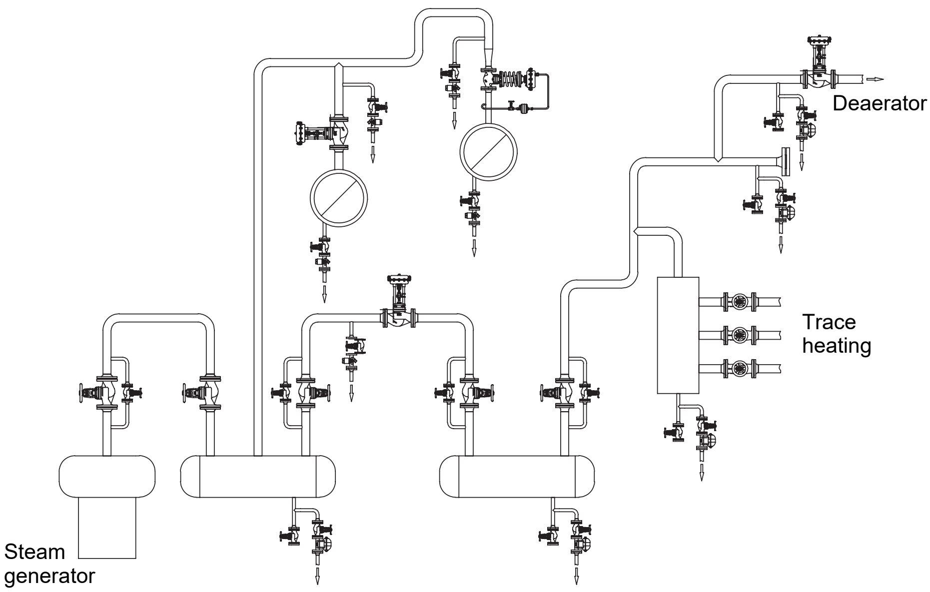


Fig. 5-7: Typical drain points

The most important drain points are as follows:

- Drain point upstream of the control valve
- Drain point on a steam header
- Drain point in a steam pipe to the steam turbine
- Drain point downstream of an injection valve for steam temperature control
- Drain point at the end of a pipe
- Drain point upstream of a rising pipe bend
- Drain point upstream of a steam jet system

Drain points are essential in underpasses or at the transition from a high pipe bridge down to ground level and back up again to another pipe bridge. Eccentric pipe reducers are often installed at changes in pipe diameter. The underside of the pipe and the underside of the eccentric reducer must be located in the same plane to prevent dams from forming (Fig. 5-8).

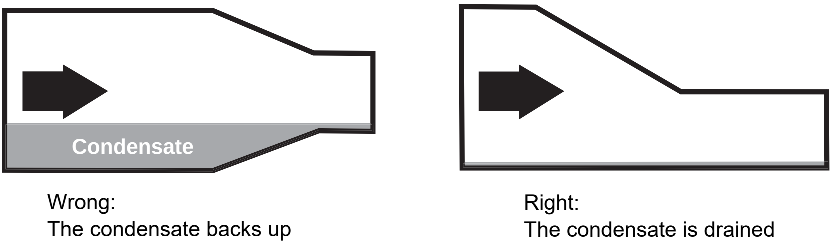


Fig. 5-8: Connection of an eccentric pipe section

5.3 Putting steam pipes into service

Considerable care must be taken when steam pipes are put into service. The large mass of cold steel requires a relatively long time to reach saturated steam temperature. One metre of DN 150 pipe corresponds to a mass of 15 kg needing to be heated. When selecting steam traps, you should remember that this maximum condensate flow rate also has to be drained off at low pressures.

All drain points must initially be brought on-line by opening the globe valves upstream and downstream of the steam traps. Next, the main valve on the steam header must be slowly opened to the 'cracked open' position. Wait until condensate emerges with steam at each drain point. The pressure in the pipe can then be gradually increased by successively closing all the drain points.

Attention:

The steam that exits at the drain points could well be flashing condensate!

When all the drain points are closed, any condensate that forms is only able to escape through the steam traps. The traps must remain on-line for some time before it is safe to assume that all condensate has been completely drained from the pipe.

The system is now at operating temperature.

5.4 Removing steam pipes from service

When a system of steam pipes is removed from service, it is important to remember that the steam in the pipes condenses. If the steam pipe is not ventilated, a vacuum will form inside it. Low-pressure steam pipes with thin walls and a large diameter have a tendency to become deformed as a result of vacuum. Completely emptying a product vessels (e.g. chemical containers) is another potentially for a vacuum situation.

5.5 Start-up drain valves

The amount of condensate that builds up during heating is 8 to 10 times more on average than the quantity that collects in a properly insulated pipe during normal service. If a steam trap is rated for the amount of condensate that occurs during the heating phase, it will be 10 times larger than the size that is required for normal operation. Conversely, if a trap is designed for the amount of condensate that collects during normal operation, it will be much too small to drain off the condensate that builds up as a result of heating. An automatic start-up drain valve provides the perfect solution to this dilemma (Fig. 5-9). This valve combines a steam pressure-controlled function with a temperature-controlled function. The temperature control part is taken care of by the bimetallic assembly.

When the steam pipe is put into service, a compression spring holds the valve open to enable air and cold condensate to be drained off up to a closing pressure of 1.5 bar. If the pressure is greater than 1.5 bar, the drain valve remains closed. When the steam pipe is removed from service, the valve opens as soon as the pressure falls below 1.5 bar. The pipe is then completely drained without a vacuum forming.

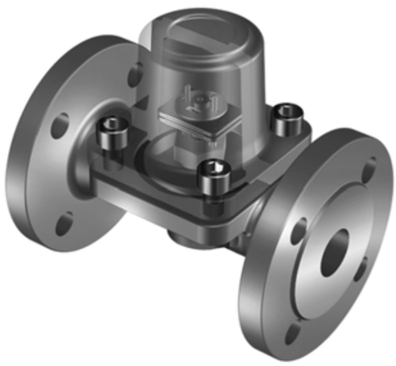


Fig. 5-9: Automatic start-up drain valve, ARI Type CONA® 665

In production facilities where the steam boiler plants are shut down at weekends, automatic start-up drain valve enable the steam pipes to be put into service faster and eliminates water hammer. Using the boiler operator's experience, the steam boiler may be maintained at a pressure slightly less than the 1.5 bar closing pressure of the valve. The valve opens and the steam flashes. The large amount of condensate that has collected is drained off without the need for cyclic inspections and without having to open or close any bypass valves. Start-up drain valves help increase productivity by reducing the time to start up and shut down a plant. Production outages due to water hammer or defective components are a thing of the past.

Start-up drain valves are also suitable for condensate drainage if the apparatus is started up at such a low steam pressure that the condensate cannot be returned to the closed condensate system.

When a cold plant is put into service, the condensate is initially drained off by means of the drain valves. The steam trap connected in parallel does not begin to discharge to the condensate system until the steam pressure has risen sufficiently.

5.6 How to prevent freezing

Room air heaters that draw in fresh air have a tendency to freeze if the outdoor temperature falls below 0 °C and the plant is not under positive steam pressure. A small amount of condensate always remains in the heater batteries when the plant is removed from service. If the start-up drain valve is installed underneath the room air heater, it can drain off residual condensate when the heater is shut down.

5.7 Draining remote steam pipes

In many cases, condensate from remote steam pipes is not returned to the boiler house, for example because there is no condensate pipe. If one is installed, it is often completely filled with cold condensate, so that newly supplied condensate is able to flash.

Recommendation:

It is advisable not to discharge the condensate into a plastic sewer because PVC is not dimensionally stable at extreme water temperatures and the cement in the joints could be attacked.

If there is no cost-effective way to return the condensate, the best alternative is to connect the steam trap via a simple sandpit containing gravel by means of an open standpipe. The flash steam is then discharged into the open and the condensate drains into the gravel. Freezing is reliably prevented, especially in winter.

5.8 Steam traps for pipe drainage

Selecting the right steam trap is not always easy. Each type of trap has its own specific advantages, but it can also have disadvantages if used incorrectly. The choice is mainly determined by the operating mode and the situation in the field.

Drainage with thermostatic steam traps

Thermostatic steam traps guarantee good air venting during start-up. As long as the condensate temperature is lower than the saturated steam temperature, the trap remains fully open and condensate and air are discharged quickly. Thermostatic steam traps adapt automatically to increases in pressure or temperature.

They close again as soon as the condensate temperature approaches the saturated steam temperature. The trap opens when the condensate cools down to a temperature approximately 10 ºC lower than the saturated steam temperature. Condensate backs up while the steam trap is closed. It is advisable not to insulate the globe valve upstream of the steam trap or the steam trap itself, in order to avoid unnecessarily prolonging the time between the trap's "closed" and "open" states. Many pipes are uninsulated as much as a metre upstream of the steam trap.

Drainage with inverted bucket steam traps

Inverted bucket steam traps work independently of the condensate temperature. Drainage of the collected condensate is controlled according to the level without backing-up. However, inverted bucket traps are relatively slow when it comes to venting air, so that the start-up process may be delayed. If superheated steam reaches the trap, the water seal could boil empty. If it loses its water seal and remains open, steam is lost. This can happen in practice if the connecting line between the steam pipe and the steam trap is too short or if the pressure drops in the steam pipe. It is advisable not to insulate the steam trap or the upstream globe valve.

Drainage with thermodynamic steam traps

Thermodynamic steam traps work independently of the condensate temperature. The valve disc inside the trap closes before saturated steam has a chance to exit at high velocity. A steam cushion forms above the valve disc, forcing it down and closing the steam trap. When this steam cushion condenses, the valve disc can no longer be closed against system pressure. It sometimes happens when a steam pipe is put into service that air from the pipe is bound over the valve disc. The trapped air prevents the steam trap from opening correctly. Vent holes drilled in the disc to permit air venting can result in leaking steam.

Drainage with ball float steam traps

Ball float steam traps work independently of the condensate temperature. Drainage of the collected condensate is continuous and controlled according to the water level inside the trap. Good ball float traps are fitted with an air vent.

If superheated steam reaches the trap, the chamber can boil dry. In contrast to the inverted bucket principle, however, no steam is lost by the ball float trap because the float sinks to the bottom of the housing and the steam trap valve remains closed.

Drainage with fixed orifice steam traps

Fixed orifice steam traps (also known as venturi traps) are designed to allow a precisely calculated amount of condensate to flow under constant, defined pressure conditions. The diameter of the drain orifice is exactly rated for this flow.

Since the diameter is constant, the orifice trap is unable to respond to variations in the operating parameters. Steam may be lost as a result. Fixed orifice steam traps cannot react to the significant differences in flow rate between start-up and normal operation because they lack the variable valve system of other trap types. The use of these traps to drain steam pipes is not recommended for this reason. The operating principle of the various steam trap types is described in detail in *Chapter 7.0 Steam Traps*.

6.0 Condensate Management

6.1 Condensate as a cost factor

There are several reasons why around 25 % of the valuable condensate is not returned to the energy process via the boiler house.

- Users are unaware of the potential money saving.
- Users are unaware of the influence of steam traps on the overall energy balance.

Calculation of the condensate loss

The amount of condensate that is lost may be approximated by the amount of make-up water supplied to the boiler. This value can be read off directly on the water meter downstream of the softener or the demineralisation plant.

If this is not possible, you can also calculate the percentage of returned condensate. This calculation is based on the conductivity or the chloride content of the make-up water and the feedwater.

$$\text{Percentage of condensate returned} = \frac{Cl_{mw} - Cl_{fw}}{Cl_{mw}} \times 100\%$$

Cl_{mw} = Chloride content (or conductivity) of the make-up water
Cl_{fw} = Chloride content (or conductivity) of the feedwater (excluding the deaerator)

Example:
Cl_{mw} = 50 mg/l
Cl_{fw} = 30 mg/l

$$\textit{Continuous blowdown percentage} = \frac{50 - 30}{50} \times 100\% = 40\%$$

Condensate costs

Condensate of (saturated) steam at a pressure of 10 bar has a heat content (enthalpy according to the steam table at 10 bar) of 762 MJ/tonne. Assuming a gas price of €7.50 /GJ and a boiler efficiency of 90 %, the heat content of the condensate alone costs about €6.35 /tonne. If the steam pressure is 3 bar, the costs for the heat content (561 MJ/tonne) of the condensate amount to €4.20 /tonne. If the condensate is not returned to the boiler house, this figure is increased by water costs of approximately €1.- /m^3 for treatment and effluent costs.

Flash steam costs

Flash steam is often discharged into the atmosphere without thinking. The heat content that is wasted is 2675 MJ/tonne.

Assuming a boiler efficiency of 90 % and a price of €7.50 /GJ for natural gas, one tonne of discharged flash steam costs as much as €21.60.

Once again, water costs of approximately €1.- /m^3 must be added to this figure.

6.2 Efficient condensate systems

Calculating efficient condensate systems and the energy recovery from condensate is quite complicated.
The fact that condensate does not behave like water, but rather resembles a two-phase flow (flash steam with condensate), is not properly appreciated. This is a common cause of many condensate and transport problems, e.g. in particular:

- Incorrectly sized pipes
- Unacceptable two-phase flow velocities
- Incorrect choice of the steam trap type
- Poor heat transfer efficiency if condensate backs up in the heat exchanger
- Pipe leakage due to corrosion, erosion or water hammer
- Inexpert installation
- Merging of several condensate pipes into one
- Condensate lift downstream of a steam trap

Incorrectly calculated pipes

The most frequent mistake is made when the condensate system is designed. The pipe sizes are erroneously calculated based on a pure water flow rather than the actual two-phase flows. The discharged condensate flashes because the pressure downstream of the steam trap is lower than in the heat exchanger.

The flash steam that is produced has a volume 300 to 1500 times greater than that of the original condensate. It is therefore essential to consider the volume of the flash steam when calculating the condensate pipe.

Percentage of flash steam

The percentage of flash steam can be determined as follows:

a) Calculation

Percentage of flash steam $= \dfrac{h_1 - h_2}{r} \times 100\%$

where:
h_{f1} = Heat content of the condensate upstream of the steam trap
h_{f2} = Heat content of the condensate corresponding to the pressure in the condensate system
h_{fg} = Latent heat of vaporisation corresponding to the pressure in the condensate system

Example:
Pressure upstream of the steam trap 8 bar, h_{f1} = 721 kJ/kg
Pressure in the condensate system 3 bar, h_{f2} = 561 kJ/kg
Latent heat of vaporisation at 3 bar, h_{fg2} = 2163 kJ/kg

Percentage of flash steam $= \dfrac{721 - 561}{2163} \times 100\% = 7.4\%$

b) Using a diagram

The percentage of flash steam can also be determined using a diagram (refer to Fig. 6-2).

c) Using a rule of thumb

Percentage of flash steam = temperature difference across the steam trap x 0.2.

Example:
Steam pressure upstream of the steam trap *10 bar (condensate temperature = 180 °C)*
Pressure in the condensate system *5 bar (temperature = 150 °C)*
Percentage of flash steam = *(180 - 150) x 0.2 = 6 %*

If steam flashes at 10 bar into an atmospheric condensate system (100 °C), the flash percentage is (180 - 100) x 0.2 = 16 %.

Steam at 5 bar has a specific volume of 0.375 m^3/kg, whereas the specific volume of atmospheric steam (1 bar) is 1.69 m^3/kg.

In other words, if 1000 kg/h of condensed 10 bar steam is fed to a 5 bar condensate system, 60 kg/h of flash steam with a volume of 60 x 0.375 = 22.5 m^3/h is produced.

160 kg/h of flash steam with a volume of 160 x 1.69 = 271 m^3/h is produced if the condensate is supplied to an atmospheric system.

Only the volume of the flash steam is important for calculating the pipe diameter. The volume of the condensate is so small compared to that of the flash steam that it can be ignored for the purpose of the calculation (Fig. 6-1).

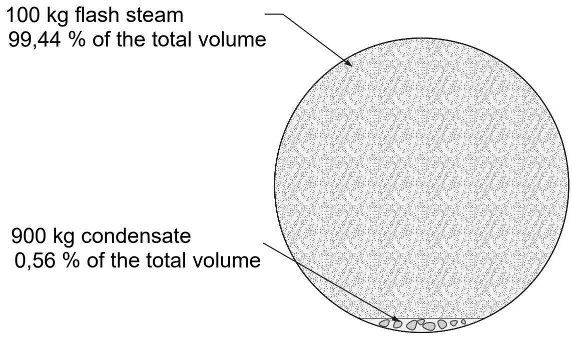


Fig. 6-1: Ratio of flash steam to condensate volume

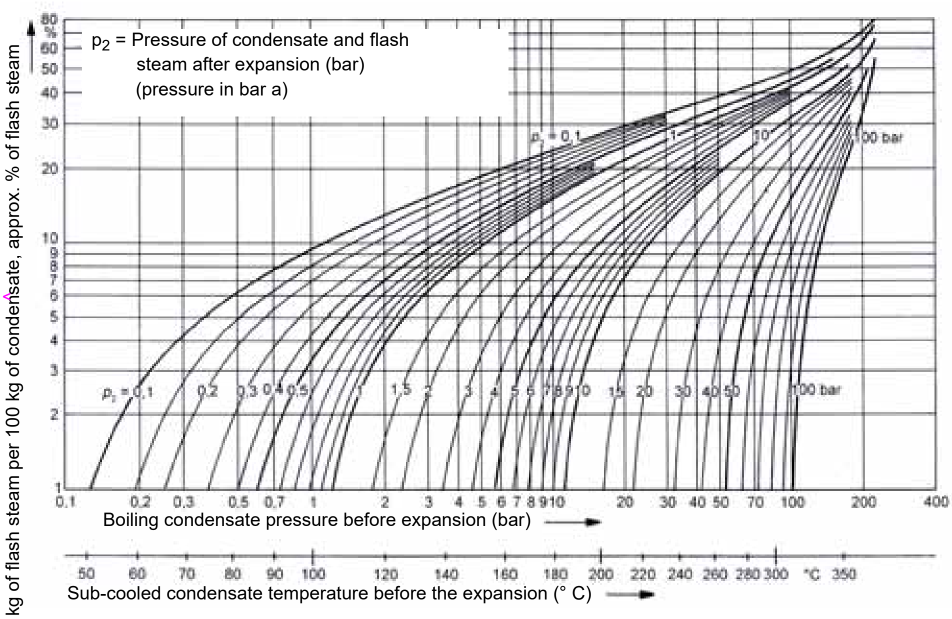


Fig. 6-2: Quantity of Flash Steam Graph

Example 1:

Condensate at 10 bar saturation temperature flashes into an atmospheric condensate system. The percentage of flash steam is 15.2 %.

According to the rule of thumb, this is equivalent to (180 - 100) x 0.2 = 16 %.

Example 2:

Condensate at 5 bar saturation temperature flashes into a 2 bar condensate system.

The percentage of flash steam is 6 %.

According to the rule of thumb, this is equivalent to (152 - 120) x 0.2 = 6 %.

Unacceptable two-phase flow velocities

Downstream of the steam trap there is a mixture of flash steam and condensate in the condensate pipe, with 95 % of the total volume typically accounted for by steam.

The condensate is entrained at the same velocity as the flash steam. Whereas the flash steam continues to flow smoothly when it reaches a pipe bend, the water particles have a tendency to keep moving in the same direction owing to their greater specific density. As a result, they hit the bend in the pipe, initially creating a barrier and then suddenly shooting through it like a wave (the correct technical term for this phenomenon is "plug flow"). Erosion occurs owing to this unacceptably high flow velocity. It can be assumed that around 75 % of all leakage in condensate pipes is on bends.

A maximum velocity of 20 m/s is recommended for condensate pipes in the literature on steam traps. In worst-case conditions, this value can be critical. The water droplets in a continuous flow can have an impact velocity of more than 60 km/h.

Apart from ball float steam traps, all other trap types work intermittently. It is not uncommon for an intermittently operating steam trap to remain shut for up to half its service time (and possibly even longer). If the condensate is discharged in bursts, the velocity can increase to 120 km/h for short periods.

Efficient velocity

To ensure operation without erosion, approximately 10 m/s should be taken as a starting basis when calculating condensate pipe sizes with float-type steam traps or 6-8 m/s for systems with other trap types. Individual calculations provide further information.

Recommendation:

10 m/s for ball float steam traps or approx. 6-8 m/s for other trap types

Which calculation method is not acceptable?

Unfortunately, it is common to base these calculations on water. The result is then corrected and a permissible velocity of 0.3 to 0.5 m/s is assumed.

The correction is reasonably accurate at pressures up to around 3 bar, even though adequate account has still not been taken of the flash steam volume. The nominal diameter of the pipe can deviate by up to three sizes, depending on the condensate flow rate:
Example: Size DN 40 is required instead of DN 80.

Important:
The majority of problems in condensate systems are attributable not to defective steam traps but to incorrectly calculated condensate pipes. If a condensate header does not contain any flash steam, the diameter is determined by calculating the condensate pipe as a water pipe with a permissible water velocity of 1.5 to 2 m/s. The volume of the flash steam must be considered when calculating condensate pipes.

WARNING:

A condensate pipe is NOT the same as a water pipe.

Determination of the condensate pipe diameter

The fundamental formula for calculating the diameter of a condensate pipe is as follows:

$$Q = \frac{1}{4}\pi \times D^2 \times v$$

where:

Q = Water, gas or steam flow rate in m^3/s
D = Inside diameter of the pipe in mm
v = Flow velocity in m/s

In practice, it is advisable to specify the flow rate in kg/h and the pipe diameter in mm. The formula must therefore be adapted as follows in order to calculate the required diameter ($Q = 1/4\ \pi \times D^2 \times v$):

$$D = \sqrt{\frac{354 \times Q}{v}}$$

where:

D = Diameter of the condensate pipe in mm
Q = Condensate flow rate in kg/h
X = Proportion of flash steam (decimal)
v_g = Specific volume (m^3/kg)

$$D = \sqrt{\frac{354 \times Q(kg/h) \times X \times v_g(m^3/kg)}{v(m/s)}}$$

Example:

1000 kg/h of condensed steam at 11 bar (h_f = 781 kJ/kg) flashing in the 4 bar condensate system (h_f = 604 kJ/kg, v_g = 0.4622 m^3/kg and h_{fg} = 2133 kJ/kg)

The percentage of flash steam is as follows:
(781 - 604)/2133 x 100 % = 8.3 % or 83 kg/h

The volume of flash steam is as follows:
83 x 0.4622 = 38 m^3/h

97.6 % of the volume of a condensate / flash steam mixture is accounted for by flash steam. The corresponding pipe diameter for a velocity of 8 m/s is as follows:

$$D = \sqrt{\frac{354 \times 1000 \times 0.08 \times 0.4622}{8}} = 40\ mm\text{, in other words DN 40}$$

Assuming discharge into an atmospheric condensate system (v_g = 1,694 m^3/kg),
(781 - 418)/2258 x 100 % = 16 % of the condensate, in other words 160 kg/h, flashes.
In this case, the condensate pipe will have the following diameter:

$$D = \sqrt{\frac{354 \times 1000 \times 0.16 \times 1.694}{8}} = 110\ mm$$

A size DN 100 or DN 125 pipe should be selected, depending on the local situation.

Simple determination of the condensate pipe diameter

information is presented in table form in Fig. 6-3.

The first column shows the steam pressure while the horizontal indicates the back pressure in the condensate system. The table specifies the pipe diameter for 100 kg/h at defined pressure drops and a flow velocity of 10 m/s. If the required velocity is less than 10 m/s, e.g. 5 m/s, the calculated diameter must be multiplied by $\sqrt{10/5} \approx 1.4$. In this case, DN 50 becomes DN 80, for example.

WARNING:

Identical pipe sizes up- and down-stream of a steam trap may be an are an indication of incorrect sizing.

The nominal diameter of condensate pipes can be determined directly using the table in Fig. 6-3.

Steam pressure (bar abs.)	**Pressure at the end of the condensate pipe (bar absolute)**												
	1	**2**	**3**	**4**	**5**	**6**	**7**	**8**	**9**	**10**	**12**	**15**	**18**
1													
2	15.4												
3	19.5	8.9											
4	22.2	11.8	6.5										
5	24.2	13.7	8.7	5.1									
6	25.0	14.5	9.6	6.2	4.9								
7	25.7	15.1	10.2	7.0	4.4	2.6							
8	28.2	17.3	12.5	9.4	7.1	5.9	4.9						
9	28.7	18.2	13.3	10.3	8.1	6.7	5.9	2.9					
10	30.0	18.9	13.9	10.9	8.7	7.3	6.5	4.0	2.6				
12	31.4	20.1	15.0	12.0	9.8	8.6	7.6	5.5	4.4	3.4			
15	33.2	21.6	16.4	13.2	11.1	9.8	8.8	6.8	5.9	5.1	3.5		
18	34.6	22.8	17.5	14.3	12.1	10.7	9.8	7.9	7.0	6.2	4.8	3.1	
20	35.4	23.4	18.1	14.9	12.7	11.3	10.3	8.6	7.6	6.8	5.4	3.7	2.1
To determine the nominal diameter of a pipe using the value obtained with the table, this value must be multiplied by the corresponding flow rate factor in the table below.													
kg/h	**100**	**200**	**300**	**400**	**500**	**600**	**700**	**800**	**900**	**1000**	**1500**	**2000**	**3000**
Factor	1.0	1.4	1.7	2.0	2.2	2.4	2.6	2.8	3.0	3.2	3.9	4.5	5.5

Fig. 6-3: Nominal diameter of condensate pipes

If necessary, the diameters should be corrected according to the actual condensate flow rate using the factors given in Fig. 6-3.

Example:
Steam pressure = 9 bar, condensate flow rate = 1500 kg/h
Back pressure in the condensate system = 2 bar

The value 18.2 can be read off in Fig. 6-3 for 9 bar steam pressure and 2 bar back pressure. The correction table shows the factor 3.9 for 1500 kg/h. The pipe should therefore have a diameter of 18.2 x 3.9 = 71 mm. A size DN 80 pipe should be selected. A size DN 65 pipe might also be suitable, depending on the local conditions (short pipe, continuous flow).

Condensate discharge when multiple drain points are connected to a single header

If several condensate pipes flow into a single header and from there to the flash vessel, the pipe diameter must be adapted to take account of this situation.

Fig. 6-4 shows three different condensate pipe sizes (DN 80, DN 50 and DN 40) with a single header.

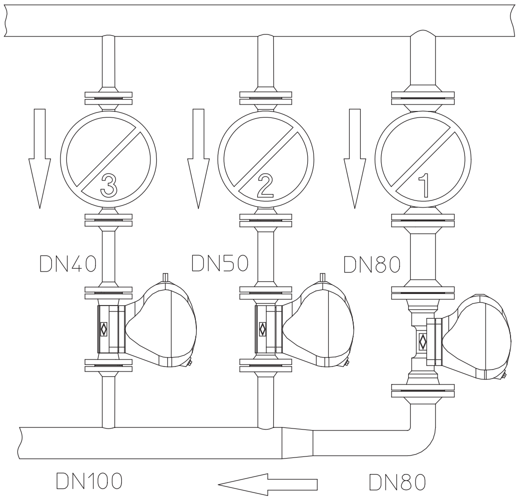


Fig. 6-4: Drainage from multiple heat exchangers into a single header

The condensate pipe from heat exchanger 1 is size DN 80 up to the point where heat exchanger 2 is connected. The diameter of the downstream condensate pipe section must be calculated as follows:

$$D = \sqrt{(80^2 + 50^2)} = 94 \text{ mm}, \text{ in other words DN 100}$$

Heat exchanger 3 has a size DN 40 drain at the next connection point.
The diameter of the final pipe section is as follows:

$$D = \sqrt{(94^2 + 40^2)} = 102 \text{ mm} \quad \text{(no need for further adaption)}$$

High condensate header

If the condensate header is higher than the trap of the steam user, for example at a pipe bridge, the condensate must overcome the gravity head from the user to the header.

The condensate is accompanied by flash steam bubbles, between which so-called "condensate plugs" are formed. These plugs fall through the steam bubbles due to gravity, causing small water hammer pulses. This process may be repeated several times.

To avoid this phenomenon, a flow velocity 12 to 15 m/s higher should be selected in rising condensate pipes. The condensate can then be drained without any problems.

Incorrect choice of the steam trap type

It is unfortunately a common occurrence for inappropriate steam trap types to be selected for certain applications or for the traps to be incorrectly installed. One common mistake, for instance, is to use thermostatic steam traps as level controllers or to install thermodynamic traps in a system with high back pressure (> 60 % of the steam pressure).

If the traps or the control valve trim are replaced, you must take account of the size of the controller for the steam trap. Controllers in high pressure traps have smaller orifices than low pressure traps. This means that a high pressure trap may not be adequately sized for a low pressure application.

Please note: The capacity of a steam trap varies according to the pressure difference. For useful tips concerning the most suitable trap type for individual requirements and applications, refer to *Chapter 7.0 Steam Traps*.

Poor heat transfer if condensate backs up in the heat exchanger

If the condensate drainage in the heat exchanger stalls (backs up), the heat transfer surface of the process apparatus (e.g. the heat exchanger, trace heater or jacket) is completely or partially flooded with condensate. No steam, or not enough steam, is able to condense on the heat transfer with the result that no heat transfer, is compromised.

At a steam-side temperature control condensate backup is probably detectable, whilst comparing the temperature in front of the steam trap with the steam temperature behind the control valve regulating the heat exchanger. If condensate is being drained correctly, these two temperatures are very similar. A small temperature difference is permissible; the term 'stall' is used if this difference increases to between 15 and 25 ºC.

The temperature differences can be measured with a surface or infrared thermometer. If the infrared method is used, both measuring points should have the same colour. It is possible to measure a higher temperature under the same conditions on carbon steel than on polished, stainless steel or on surfaces with an aluminium structure.

> **Important:**
>
> Surface temperature measurements are comparative measurements! It is therefore advisable to measure the surface temperature at the steam inlet of the heat exchanger and at the inlet of the steam trap and then compare the two values:

The main causes of stall are as follows:

- The capacity of the steam trap is too small.
- The pressure difference across the steam trap is too low because the pressure in the condensate system is too high. Possible reasons: Pipe diameter too small, live steam loss from steam traps, bypass valve open.

- As set-point is approached, low steam temperature (and hence pressure) steam is required in the heat exchanger. This pressure may not be sufficient to overcome the back pressure in the condensate system.
- The filter in the steam trap or the upstream strainer is clogged.
- The heat exchangers are incorrectly sized (generally oversized, with too great a heat transfer area). Too much pressure is lost, so that the pressure upstream of the trap is unacceptably low.

Pipe leakage due to corrosion, erosion or water hammer

Condensate pipes are renowned for leakage. There are three main causes:

- Erosion
- Corrosion
- Water hammer

Erosion

Erosion is caused by high velocities of the condensate / flash steam mixture. Entrained water particles act like an abrasive cleaner in the pipe bend. The change of flow direction in valves results in strong turbulences that damage not just the plug and seat but also the valve body. The mixture velocity can be reduced by increasing the pipe diameter.

Erosion in condensate systems is usually accompanied by corrosion if the condensate is aggressive. To prevent erosion, the pressure in the condensate system is sometimes increased artificially by installing an excess pressure valve at the end of the condensate pipe, for example upstream of the inlet into the condensate vessel. Less flash steam is produced in the header as a result: it is important to ensure that the pressure maintained by the valve does not reduce, or even halt, the capacity of the steam trap. This will reduce plant performance.

Corrosion

Corrosion in condensate systems is mainly caused by the oxygen (O_2) or carbon dioxide (CO_2) contained in the steam.
Oxygen corrosion occurs as pitting or shallow abrasion if condensate collects in an atmospheric vessel. As long as a steam cushion forms in the collection vessel, no oxygen can penetrate. However, if sub-cooled condensate reaches this vessel, ambient air will enter via the vent. Preventing the ingress of oxygen into the condensate vessel is described later in this chapter.

Carbonic acid corrosion in condensate systems is mainly encountered when the feedwater is only treated with a softener. The feedwater contains carbon dioxide compounds, such as free carbon dioxide (CO_2) from the ambient air or carbonic acid bound to the water (H_2CO_3). Only the free carbon dioxide is expelled in the deaerator. The bound carbonic acid is only split into carbonic acid gas and water at a higher temperature in the boiler. The carbonic acid gas is supplied to the user together with the steam and is then dissolved in the condensate. The pH may be reduced to less than 4.

Corrosion is particularly likely at those points where condensate cools down and is aerated. Carbonic acid corrosion is visible on the pipe or apparatus bottom and is uniformly distributed over a wide area. Chemicals that increase the pH (neutralising agents) are often added as a preventive measure. Amines that build up a protective layer can also be metered into the condensate system. However, they only make sense in the part of the system that actually carries condensate.
The latter method can occasionally lead to unpleasant side-effects in pipes and steam boilers, for example if old corrosion residues are detached and filters or drain valves become clogged as a result. If amines are used, the information provided by the supplier must always be observed.

Corrosion residues can also become detached if the feedwater is treated by demineralisation instead of softening.

If a steam or condensate system is temporarily removed from service, for example at weekends, it is advisable to provide draining at the lowest points in the system, e.g. by installing drain valves.

Inexpert installation

A few examples of inexpert installation are described in the following.

If a thermostatic steam trap is insulated, it works slowly because an insulated supply pipe and globe valve prevent essential cooling. Thermostatic or thermodynamic steam traps should never be insulated for this reason. A perforated plate is often mounted as a shield in such cases in the interests of work safety.

Steam traps that are installed in the wrong direction of flow are a particularly common mistake.

When a density sensitive trap (inverted bucket or ball float) with a threaded connection is installed, it is vital to ensure that the level control can operate in the correct plane. If the trap is turned away from the perpendicular in the pipe axis, for instance, or if it is installed at an angle, the level control will no longer be able to move freely.

6.3 Water hammer

A distinction is made between hydraulic and thermal water hammer.

Hydraulic water hammer

Hydraulic water hammer occurs, for example, if a globe valve is suddenly closed during normal flow through a water pipe. The water flow immediately comes to a standstill and the velocity energy of the moving water is converted to destructive pressure energy. The liquid downstream of the closed globe valve is still in motion and continues to flow for a while. However, since no more liquid is flowing in its wake, a vacuum is created in this section of the pipe. As soon as the flow velocity drops to zero downstream of the valve, the movement is reversed under the influence of the vacuum. A water hammer pulse occurs in the direction of the globe valve as a result.

These pressure waves may be repeated several times, during which the system pressure can significantly exceed the normal service pressure. Depending on the magnitude of the pressure peaks, the water hammer can have a destructive effect on system components, for example gaskets may be blown out.

Thermal water hammer

Thermal water hammer is mainly a problem in condensate systems. Sub-cooled condensate is fed in a condensate pipe that is partially filled with flash steam. The flash steam condenses. The volume of the flash steam is abruptly reduced and a vacuum is created locally. As a consequence of this vacuum, condensate flows towards it from all sides at high velocity. Once again, a water hammer pulse occurs here when the vacuum is removed and the liquid comes to a halt.

Sub-cooled condensate must never be fed into a pipe containing flashing condensate!

Water hammer is a common phenomenon at condensate drains and in process apparatus that is controlled on the condensate side. A condensate cooler installed downstream of a heat exchanger also increases the risk of water hammer. The condensate drained from this kind of configuration is always sub-cooled. However, sub-cooled condensate is only allowed to be returned via a separate line and sprayed into the condensate vessel from above. Another alternative would be to mix it with warm condensate using a perforated supply pipe (nozzle pipe).

Avoiding or reducing the risk of water hammer

In addition to the above-mentioned causes, condensate discharge from a drain trap into an atmospheric condensate header also results in water hammer. Imploding steam bubbles close to the port lead to serious damage to the pipe material.

Water hammer is additionally possible when a cold plant is put into service. It is a good idea to drain the condensate using a start-up drain valve while the plant is heating up. One positive side-effect of start-up drain valves is that if a plant stoppage occurs, the residual condensate is automatically drained and the system aerated.

It is advisable to feed condensate in on the top side of the header, if possible in the parallel flow direction as is usual with steam pipes (Fig. 6-5).

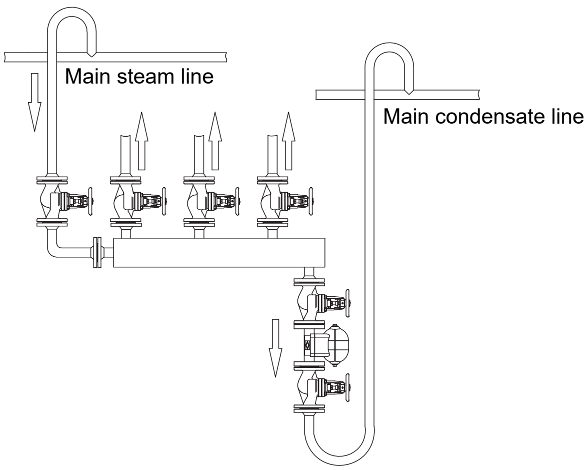


Fig. 6-5: Condensate inlet on the top side of the header

If condensate must be supplied from a low level to a higher condensate pipe, water hammer can be avoided by installing a condensate lock as a damping mechanism. The steam cushion that is formed in the top part of the condensate lock cushions pressure waves (refer to Fig. 6-6).

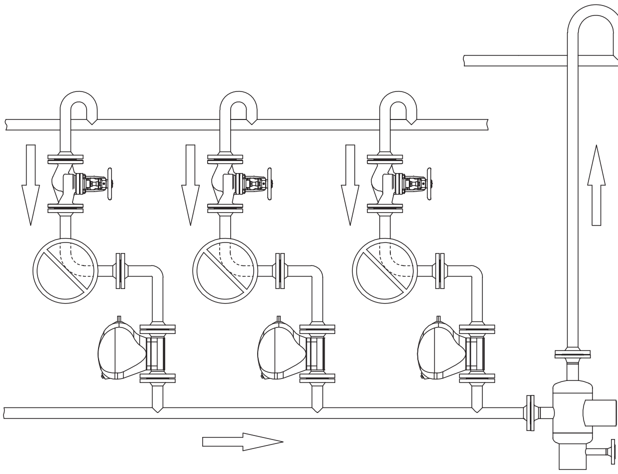


Fig. 6-6: Condensate lock used as a damping mechanism

6.4 Condensate flashing

Introduction

The temperature of the condensate that forms is identical to the saturation temperature of the steam, *for example 152 °C at 5 bar*.

Fig. 6-7 shows a typical installation***:***

- *Heat exchanger controlled on the steam side*
- *5 bar steam pressure downstream of the control valve*
- *Condensate temperature upstream of the trap = 152 °C*
- *Heat content of the condensate = 640 kJ/kg*
- *Condensate discharged into the atmosphere*

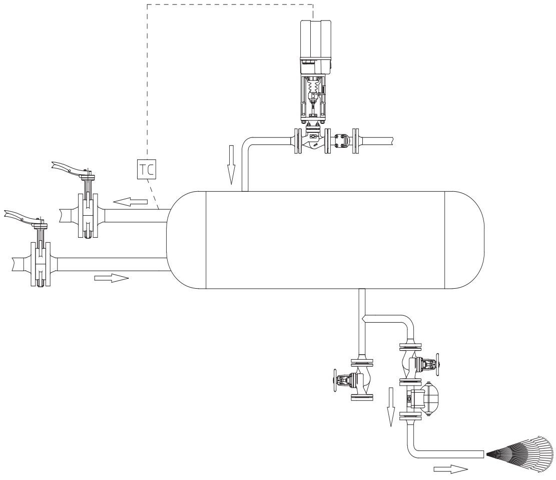


Fig. 6-7: Tube bundle heat exchanger

When it exits from the steam trap, the condensate is abruptly flashed from 5 bar to atmospheric pressure. The maximum temperature of water or condensate in the atmospheric state can be up to 100 ºC with a heat content of 420 kJ/kg.

The heat content of the 5 bar condensed steam, on the other hand, is 640 kJ/kg. In this example, the excess heat contained in the condensate amounts to 640 - 420 = 220 kJ/kg. This surplus results in partial evaporation of the condensate, also referred to as flashing.

Evaporation = flashing

In this example, the flash steam percentage can be calculated as follows:
220/2258 x 100 % = 9.7 % (Enthalpy of Evaporation (h_{fg}) at 1 bar = 2258 kJ/kg)

If this flash steam is supplied to a steam system at low pressure, it is possible to restrict the losses. How to utilise the flash steam, for example in order to save costs, is described in the next section.

Heat recovery with flash steam

The recovery of energy from flash steam facilitates a significant reduction in costs, as well as CO_2 emissions. The potential benefit of heat recovery is described in the following with the help of two diagrams and calculations. Fig. 6-8 shows how the flash steam from the condensate is discharged into the atmosphere by several heat exchangers operating at 10 bar.

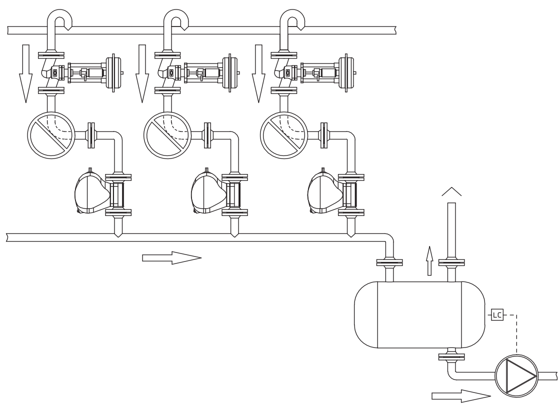


Fig. 6-8: Discharge of the flash steam into the atmosphere

Saturation temperature of 10 bar steam:	180 ºC
Percentage of flash steam:	(180 - 100) x 0.2 = 16 %
Live steam flow rate:	1500 kg/h
Loss due to flash steam:	240 kg/h

Assuming a gas price of €7.50 /GJ, this represents a yearly loss of €38,400.- for 8000 hours operation.

Fig. 6-9 shows how flash steam is produced and supplied to a low-pressure steam system.

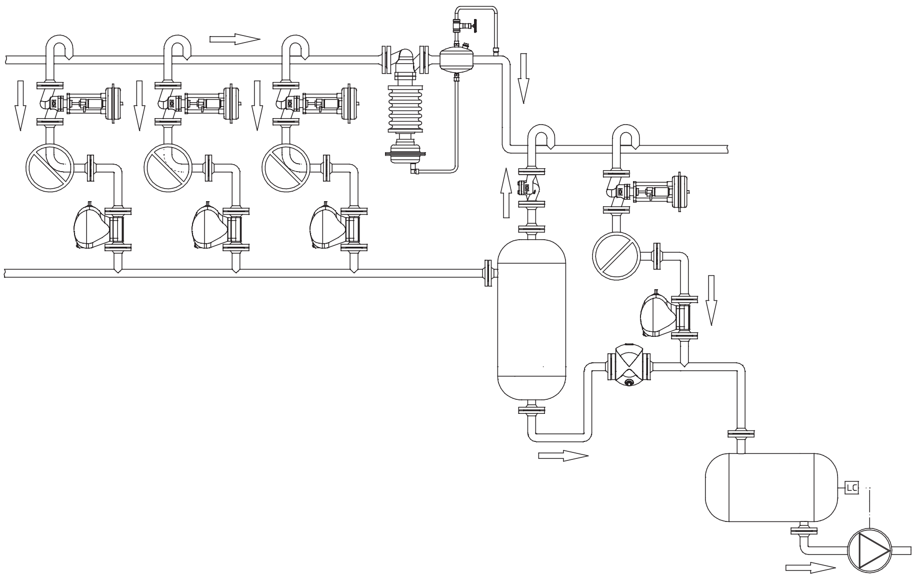


Fig. 6-9: Recovery of the flash steam in a low-pressure steam system

1500 kg/h of condensed steam is flashed from 10 bar to 2.5 bar (127 °C saturation temperature) into a downstream steam system.

The amount of flash steam formed is as follows: (180 - 127) x 0.2 = 10.6 % or 160 kg/h.
The residual condensate (1340 kg/h) is flashed to the atmosphere in the second flash vessel. (127 - 100) = 5.4 % or 72 kg/h is discharged in the process.

The discharge steam loss is reduced to 72 kg/h with a value of €11,500.- /year instead of €38,400.- – an annual cash saving of €26,900.- .

6.5 Condensate flash vessels

There are two different types of flash vessel (or flash tank):

- Atmospheric flash vessels
- Pressure expansion vessels

Atmospheric flash vessels

Atmospheric flash vessels are used to flash condensate from low-pressure steam systems. The flash steam is discharged into the atmosphere because the low steam pressure is unsuitable for commercial applications. If there is no condensate, or if the temperature of the condensate remains below 100 ºC, ambient air can enter the vessel freely and the condensate is aerated. The risk of oxygen corrosion (pitting) is increased in the vessel and in the downstream pipes. A safety valve that regulates the pressure to 0.2 bar can be installed in the discharge pipe to prevent the ingress of air. If not enough condensate is supplied, a small amount of live steam is blown in via a control valve to prevent vacuum in the vessel and ensure that slight gauge pressure is maintained.

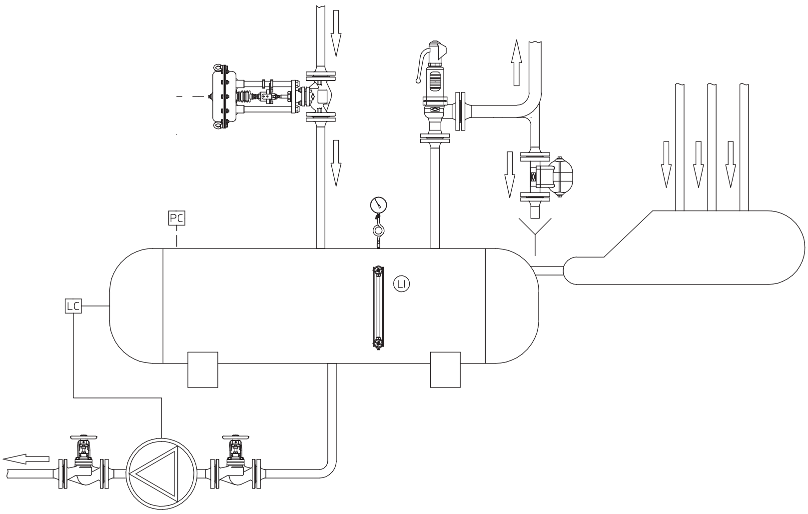


Fig. 6-10: Flash vessel with steam cushion

Pressure expansion vessels

The mixture of condensate and flash steam collects in the pressure expansion vessel (Fig. 6-11), where the steam is separated from the condensate. The flash steam is supplied to a low-pressure steam system while the condensate is returned to the boiler house.

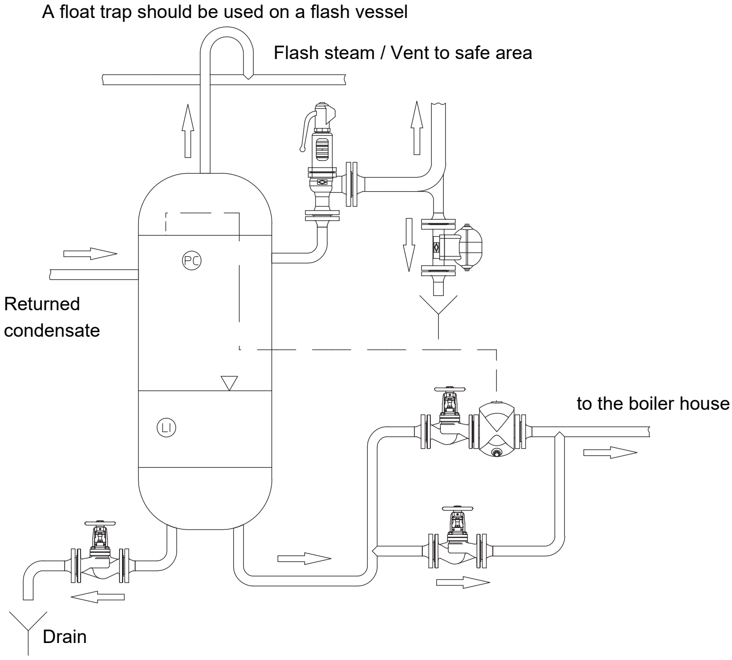


Fig. 6-11: Pressure expansion vessel

Requirements specified for pressure expansion vessels

Pressure expansion vessels are connected to the condensate system on the steam side. Strictly speaking, the term "expansion vessel" is inaccurate. The condensate is flashed as soon as it exits from the steam trap, providing it is supplied to a system with a lower pressure than the steam pressure in the heat exchanger. The separation of the condensate from the flash steam takes place in the flash steam vessel.

The sizing of the vessel with its connection ports and its protection have to meet various requirements in order to guarantee the corresponding quality of the flash steam. The flash steam that is fed into the steam system must not be wet – if it is, condensate will be entrained with it. To guarantee optimal separation of the condensate from the flash steam, the vessel should ideally work like a cyclone. The mixture of condensate and flash steam must be supplied tangentially above the water level for this purpose. Owing to the high specific mass of the water, it is spun against the vessel wall and most of the flash steam is separated from the condensate.

An additional separation can be achieved with the calculation by assuming a maximum steam velocity in the vessel of approximately 1 m/s. In addition, the path travelled by the steam to the outlet point should be as long as possible in order to ensure a sufficient residence time. Due to the low steam velocity and the high specific mass of the condensate, the separation is almost perfect and the flash steam reaches up to 97 % dryness.

A float-type steam trap must be used to drain a flash vessel. In the event of sudden pressure relief, for example if there is an abrupt increase in steam consumption in the low-pressure steam system, the pressure expansion vessel boils empty. The condensate in the vessel is completely flashed as a result. To avoid this problem, the float-type steam trap should be mounted higher than the supply pipe. This manometric loop ensures that the trap is always full of condensate. Flash steam is consequently unable to reach the trap and the service life is significantly extended.

The pressure expansion vessel must be fitted with a safety valve! The set pressure is determined by the maximum permissible pressure load at the maximum vessel temperature. The vessel must be capable of discharging both the maximum possible flash steam flow rate from the maximum inflow of condensate and the steam flow through any open steam traps.

6.6 Multi-stage condensate flashing

Different steam pressures can occur in a plant with several steam systems. If this is the case, it is advisable to use multi-stage flashing.

Fig. 6-12 shows how condensate from a 20 bar steam system is flashed in a flash vessel connected to an 11 bar steam system. The condensate from the 11 bar heat exchanger is then flashed in a flash vessel connected to the 3 bar steam system. The condensate from the 3 bar heat exchanger is supplied to an atmospheric flash vessel, from which flash steam is discharged into the open. This steam cloud can be reduced if required by means of a heat exchanger with cold condensate or feedwater, for example using a vapour cooler.

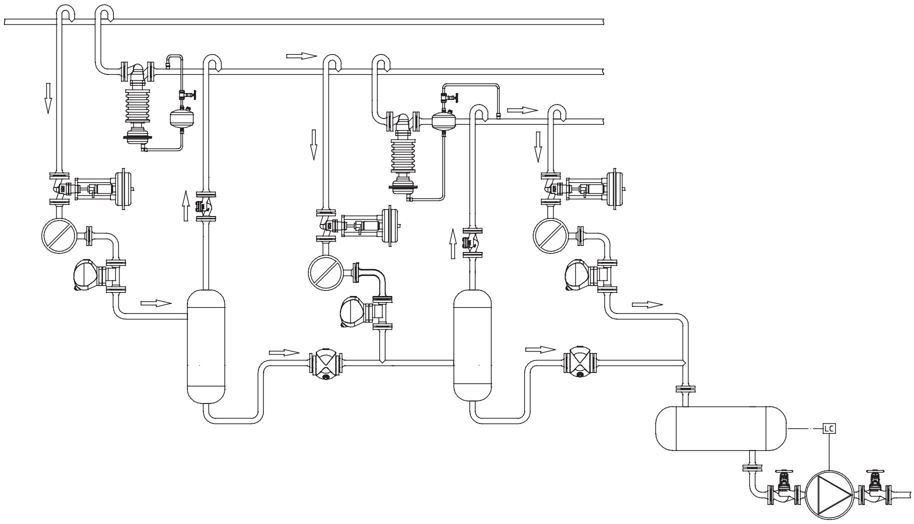


Fig. 6-12: Multi-stage flashing

6.7 How to avoid the flash steam cloud

The flash steam cloud that inevitably emerges from an atmospheric flash vessel unfortunately cannot be avoided altogether. However, there are several tricks that can be employed to reduce it. Some of them are very effective, while others occasionally resorted to in practice are technically irresponsible. A few of the most suitable methods for reducing the steam cloud are described in the following.

Condensing with feedwater or condensate

One proven technique for condensing the flash steam cloud without water hammer is to spray feedwater or cold condensate into the vent pipe. The cloud can be partially or completely suppressed in this way. Sub-cooled condensate produces better results than hot condensate. The following points must be taken into account:

- There must be no water hammer
- The temperature of the make-up water / condensate mixture to the deaerator must not exceed 95 ºC

Another condition is that the flow velocity in the vent pipe must not be too high. It is consequently advisable to select the pipe section to lead into the vent pipe with a larger diameter. This section should have a diameter 2.5 times that of the vent pipe and be between 1 and 1.5 metres long.

Spraying the water in a cascaded condenser counter to the flow of flash steam, with modulating control of the water, represents a particularly elegant solution (Fig. 6-13).

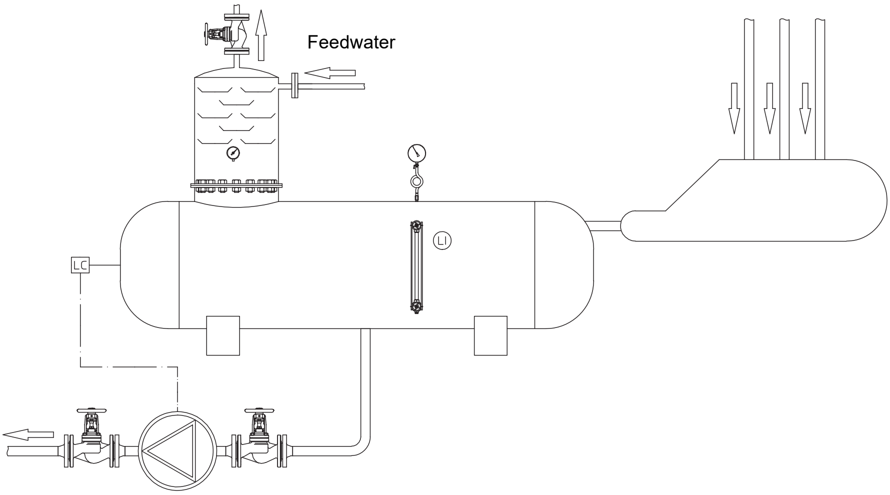


Fig. 6-13: Reduction of flash steam by spraying with feedwater or condensate

Reducing the steam pressure

In steam users without temperature control, the steam pressure should be reduced to the minimum permissible value. The lower the steam pressure and the closer the saturation temperature to 100 ºC, the lower the percentage of flash steam.

Example:
A trace heater (product temperature to be maintained = 80 ºC) requires 500 kg/h of steam.
Steam pressure = 6 bar, saturation temperature = 159 ºC
Condensate drainage in an atmospheric condensate system
Total amount of flash steam lost = (159 - 100) x 0.2 = 11.8 % or 59 kg/h

However, a steam pressure of 2 bar is sufficient to maintain the product at a temperature of 80 ºC. If a pressure reducing valve (Fig. 6-14) were to be installed to reduce 6 bar steam pressure to 2 bar, the flash steam loss would be only 4 % instead of 11.8 %. An annual saving of 312 tonnes of steam, equivalent to €6,300.- , could be achieved in this way. The payback period would be less than twelve months.

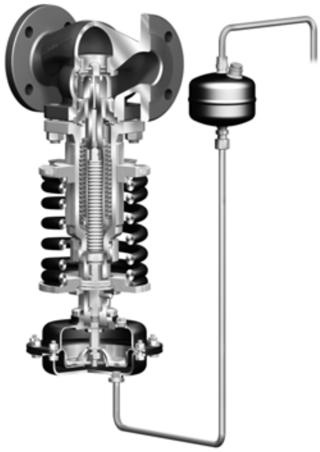


Fig. 6-14: Pressure reducing valve, ARI Type PREDU®

Steam traps with additional sub-cooling

Steam traps with variable sub-cooling should be used for trace heaters. They help reduce flashing and hence afford better protection against frost.

Thermostatic steam traps are ideal for this task. Bimetallic traps feature adjustable condensate sub-cooling. Balanced pressure steam traps may be fitted with have different capsules with different sub-cooling characteristics for greater flexibility.

Thermostatic traps do not open when the saturation temperature of the steam is reached, but will remain closed for a while longer, so that condensate backs up. The trap only opens when the temperature drops to 30 °C to 40 °C below saturation.
The condensate that has backed up continues to exchange heat with the product pipe.

Fig. 6-15 shows a steam trap that has a sub-cooling characteristic. Thermostatic steam traps with a membrane regulator are often used for trace heating (Fig. 6-16).

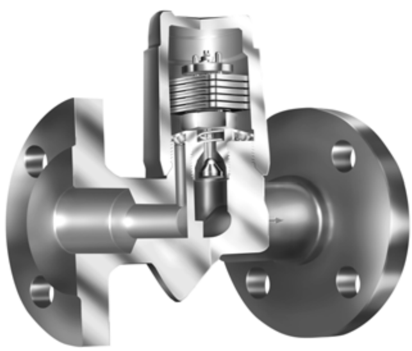


Fig. 6-15: Variable bimetallic steam trap, ARI Type CONA®B

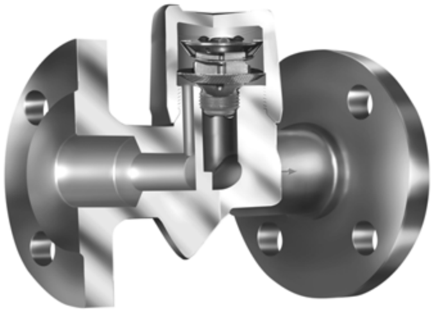


Fig. 6-16: Steam trap with a sub-cooling capsule, ARI Type CONA®M

> **Warning:**
>
> Sub-cooling is not recommended for draining steam pipes or for draining condensate from a heat exchanger.

Fig. 6-17 shows a 7 bar trace heater without sub-cooling while Fig. 6-18 depicts a steam trap that has been set to 40 ºC sub-cooling.

Without sub-cooling, (165 - 100) x 0.2 = 13 % of the flash steam is lost.
With sub-cooling, the figure is reduced to (125 - 110) x 0.2 = 5 %.

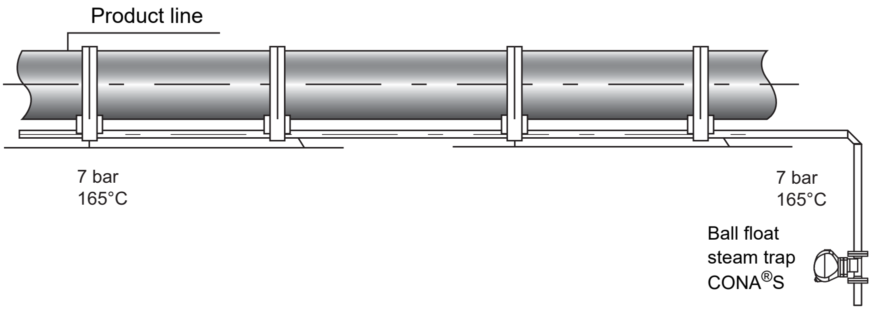


Fig. 6-17: Trace heater and steam trap without sub-cooling

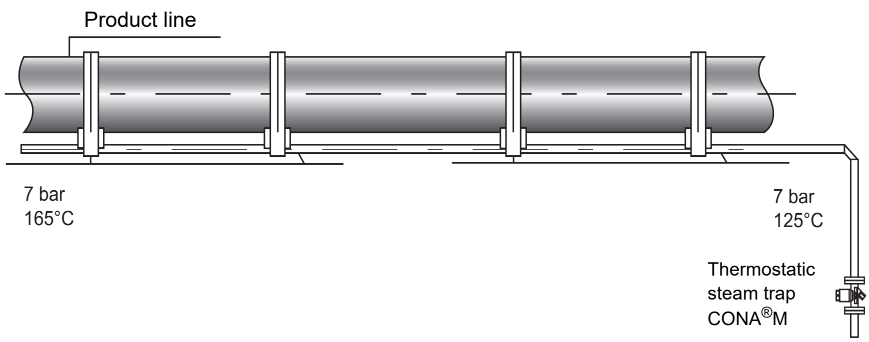


Fig. 6-18: Trace heater and steam trap with 40 °C sub-cooling

The saving for a trace heater used for frost protect systems is considerable, especially if the condensate is discharged to the atmosphere. With trace heaters like the ones in Fig. 6-17 and Fig. 6-18, which consume 2 t/h of steam during 4000 h/a in service, the total saving amounts to around €14,000.- per year.

Cooling the condensate

At process temperatures below 100 ºC (e.g. in air heaters for drying plant or central heating systems), the condensate can be cooled very efficiently by connecting a condensate cooler in series with the heat exchanger.

However, mistakes are often made here in the implementation. Fig. 6-19 shows two faulty designs.

In design A, the condensate is flashed downstream of the steam trap. The mixture of flash steam and condensate hits the tube plate of the condensate cooler at too high a velocity. The result: erosion and leakage.

In design B, the condensate is not cooled at all. The heat exchange surface is simply enlarged due to the series connection.

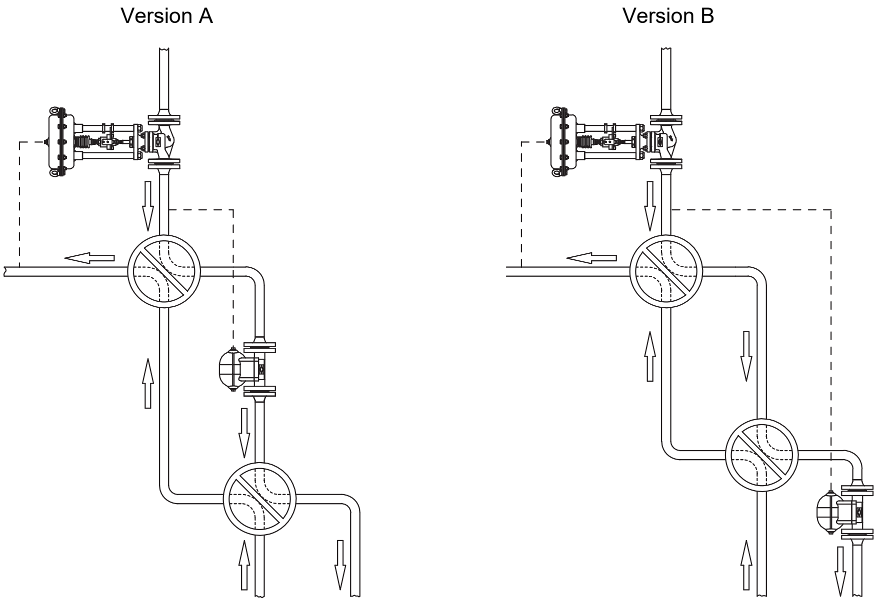


Fig. 6-19: Faulty designs for condensate cooling

Fig. 6-20 shows a more appropriate design. A float-type steam trap is installed downstream of the condensate cooler, but at a slightly higher elevation. The trap can therefore be used to control the water level and the cooler is always full of condensate. The cooled condensate should be supplied to the condensate vessel or the boiler house in a separate pipe. If not, there is an acute risk of water hammer in case of discharge into a condensate pipe with flash steam.

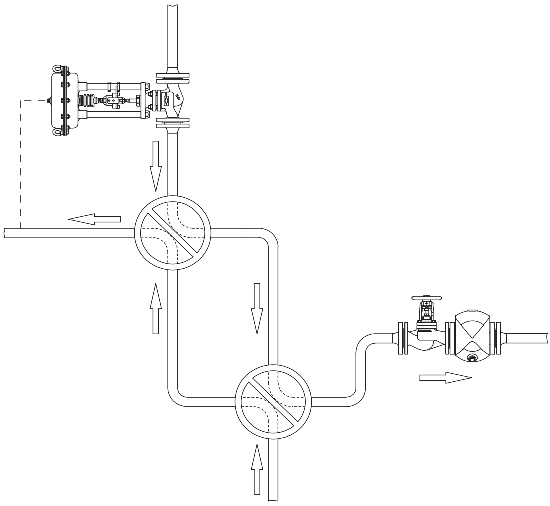


Fig. 6-20: Optimally designed condensate cooling

Thermal Compressors

Vapour compression with ejectors is occasionally used if sufficient amounts of live steam are available (Fig. 6-21). An ejector allows the flash steam to be compressed from 1 bar to 2.5 bar by using 10 bar live steam. In this example, 3 kg of live steam is required to compress 1 kg of flash steam.

This method should only be chosen if a large amount of steam has to be reduced from a higher pressure to a lower pressure.

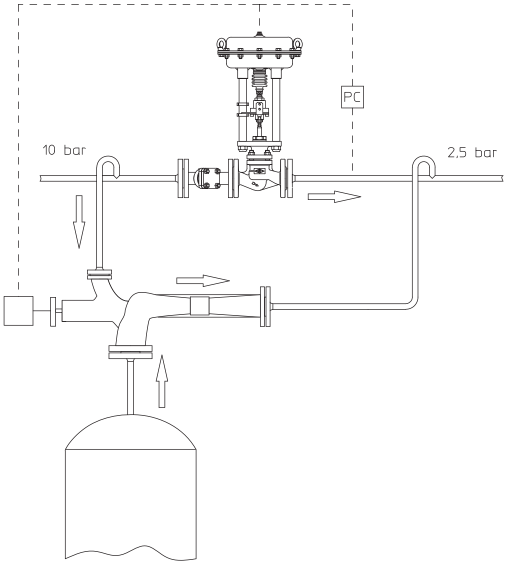


Fig. 6-21: Thermal compression

Thermal compression is often used for drying cylinders in the paper industry. The main reason for its popularity is that when condensate is drained from the cylinders, a small amount of live steam is deliberately allowed to escape together with the flash steam to enable the condensate to be drained from the drying cylinders. This mixture of flash and leakage steam is increased to the required pressure (generally only a slight amount) after each drying stage.

Typical planning errors

The same unsuitable methods are often selected to avoid the steam cloud, unfortunately with similarly unsatisfactory results in the majority of cases.

Condensate vessel without a vent pipe

The basic idea behind this method is that if there is no vent pipe, there is also no flash steam. As long as the mixture temperature of the condensate streams to the vessel remains below 100 ºC, no steam is generated. In the worst case, a vacuum is created inside the vessel, although this is not a problem providing the system is designed correctly (Fig. 6-22).

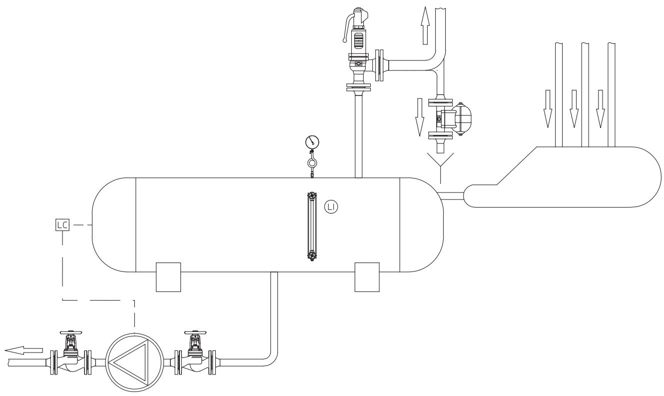


Fig. 6-22: Condensate vessel without a vent pipe

If, on the other hand, condensate reaches the condensate vessel at a temperature of more than 100 ºC, some of it will flash to steam. The pressure inside the vessel then gradually increases to the vapour pressure of the hot condensate. It is therefore essential to mount a safety valve on the condensate vessel. Ultimately, the pressure in the vessel is almost identical to that in the heat exchanger. The pressure difference is then too small and only an very small amount of condensate can be discharged to the vessel. The heat exchanger fills up with water as a result (i.e. it "floods"). The condensate pump is only able to transport a very small amount of condensate (compared to the amount of flash steam produced).

Other measures

Two other measures are commonly implemented to prevent the steam cloud:

- The mixture of returning condensate and flash steam is fed in below the condensate level.
- The mixture is sprayed into the vessel's vent pipe.

However, neither of these solutions have any effect on the vessel's heat balance and the flash steam is not reduced in size.

7.0 Steam Traps

7.1 Introduction

The enthalpy of steam and condensate was discussed in detail in *Chapter 1.0 Heat Engineering Concepts*. It was made clear that condensate discharge has a major influence on the efficiency of a steam system and that its importance should not be underestimated. Steam traps form a vital link between efficient steam usage and the condensate system. Together with a condensate system that meets all the specified requirements, they represent the crucial key to cost-effective steam use.

7.2 Steam trap specifications

Steam traps should have the following properties and satisfy the following requirements:

- Efficient and problem-free discharging of condensate without loss of live steam.
- Immediate discharge of air and inert gases when the steam system is started up.

The steam trap should have these properties at all times, even if the steam or back pressure fluctuates, and it should meet the specified requirements throughout the pressure range. All steam traps should additionally comply with the following demands:

- The seat and plug of the steam trap must withstand the abrasive action of the flashing condensate.
- The steam trap should be compact and take up as little space as possible in order to restrict heat emission losses to a minimum.
- The steam trap should be robust, reliable and insensitive to water hammer.

In many cases, the desirable properties are so diverse that they are impossible to reconcile:

- A steam trap is required to keep the steam space of a heat exchanger free of condensate.
- On the other hand, it can be useful to allow backing-up (waterlogging) in the heat exchanger in order to utilise the sensible heat of the condensate (e.g. in trace heaters for heating instruments by setting the steam trap to sub-cooling).
- Discharging of large amounts of condensate at low differential pressures (e.g. for heating and draining a steam pipe on start-up).

- Discharging of small amounts of condensate without steam loss at high differential pressures (e.g. for draining a steam pipe in service).

There are numerous applications where the requirements defined for the steam traps are actually mutually exclusive. In short, there is no such thing as a universal steam trap to suit every situation.

7.3 Classification of steam traps

The three basic types of steam trap are classified in DIN EN 26704:

- Mechanical steam traps – controlled by the condensate level
- Thermostatic steam traps – controlled by the condensate temperature
- Thermodynamic steam traps – controlled by changes in the fluid state

7.4 Mechanical steam traps

The most important members of this group are:

- Float-type steam traps (sealed float)
- Inverted bucket steam traps (float open at the bottom)

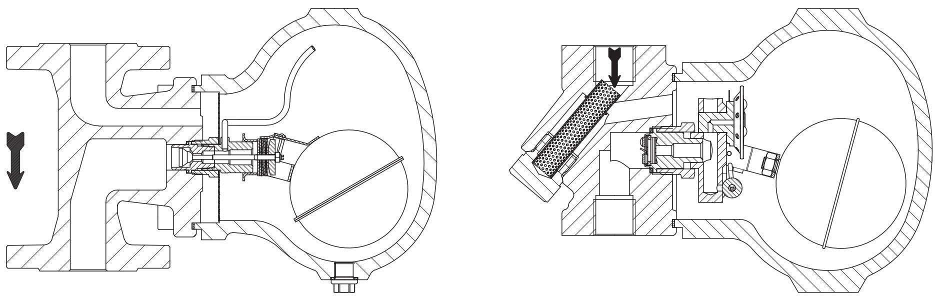


Fig. 7-1: Operation of a float-type steam trap,
ARI Type CONA®S 631 / CONA®SC 634

The operating principle of a float-type steam trap (Fig. 7-1) is based on the density difference between steam and condensate and on the buoyancy force of a ball float. The float opens and closes the drain valve or the drain gate by means of a system of lever arms. The water level in the trap rises as the condensate builds up. The float rises with the condensate and the drain valve opens; when the condensate level falls again, the float sinks and the valve closes. The float trap thus acts as a modulating level controller that guarantees continuous and immediate discharging of condensate. The drain orifice should be drilled as low down as possible, depending on the model. The seat and plug are always located below the minimum condensate level; this provides a water seal which gives protection against live steam loss. This latter property and the ability to discharge condensate continuously are the most important advantages of this trap type. The temperature of the condensate in the steam trap is the same as the saturated steam temperature of the corresponding pressure at the outlet of the heat exchanger and the inlet of the trap. No steam is lost, even if the trap is completely emptied, because in this case the float goes to the bottom of its travel, and the drain valve is closed.

Pressure range

The majority of float traps have a limited range as regards the maximum permissible pressure difference. As a result, not every trap will operate efficiently in a particular differential pressure range. If the differential pressure is too high, the buoyancy force and the length of the lever arm will be insufficient to open and close the plug. A smaller seat bore must be selected for higher maximum differential pressures, to ensure that the valve can still be operated at high pressures. Float-type steam traps are offered with different regulators (seat areas) for different pressure ranges for this reason.

Fig. 7-2 shows the interaction of the buoyancy force of the float with the closing force of the valve.

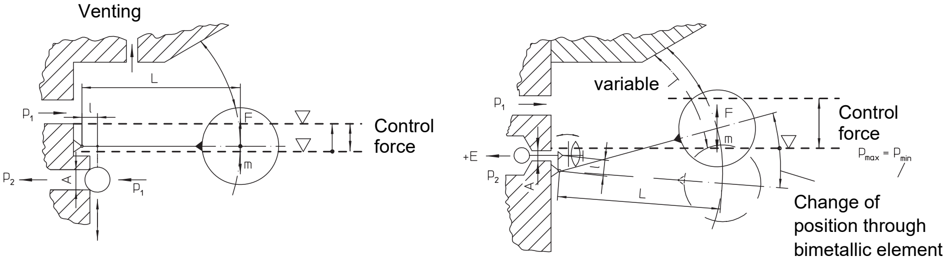


Fig. 7-2: Interaction of the forces in a float trap

Where:

P_1 = Pressure in the steam trap
P_2 = Pressure downstream of the steam trap
A = Cross sectional area of the discharge port
m = Mass of the float

F = Buoyancy force (Archimedes)
L = Lever length up to the float
l = Lever length up to the valve

CONA®SC 634: In order to open the valve, the moment created by the buoyancy force of the float and the lever length must be greater than the moment acting on the valve plug and seat area as a result of the pressure difference.

CONA®S 631: In order to close the valve, the moment created by the mass of the float and the lever length must be greater than the moment acting on the valve plug and seat area as a result of the pressure difference.

This relationship is expressed by the following formula:

$$(F - m) \times L > (P_1 - P_2) \times A \times l$$

When specifying the steam traps for a system, it is important to remember that the difference between the normal flow rate and the maximum possible flow rate can often be considerable. When the system is started up, the steam consumption – and hence the amount of condensate – can be high at comparatively low pressures. On the other hand, there may be relatively little condensate at high pressures during the normal process.

The specification for a float-type steam trap should always take account of the following aspects:

- When the (float) trap is ordered, the maximum amount of condensate to be discharged should be specified in addition to the service pressure and the differential pressure.

- The trap should be rated for this peak capacity. For example, it must have a capacity of 60/10 x 400 = 2400 kg/h in order to handle a maximum flow of 400 kg of condensate in 10 minutes.
- If a defective DN 50 trap needs to be replaced, it is not sufficient simply to take another DN 50 trap from the store. The size and designation of the controller must be checked as well, because it is they that determine the seat area and hence the trap's capacity.

Air venting

Air venting is not only important because air can cause corrosion but also because air in the steam system has a negative effect on heat transfer.

Since there is always a minimum residual condensate level in a float trap and the obturator valve orifice is underwater, air that collects in the system upstream of the steam trap can only be discharged via the trap. Float traps for steam systems have an air venting function for this reason.

Continuous air venting

The simplest method of continuous air venting is to connect a bypass pipe from the inlet to the outlet of the steam trap or to drill a short-circuit path. The steady loss of steam is a small price to pay in return for eliminating all air from the system. However, this is only an acceptable solution if the steam that is discharged can be put to meaningful use with the help of a flash vessel. A needle valve can be installed in the bypass pipe to restrict the steam loss to a minimum (Fig. 7-3). This valve should only be opened when the system is started up or vented. If you forget to close it again, it will represent an additional source of leakage or faults.

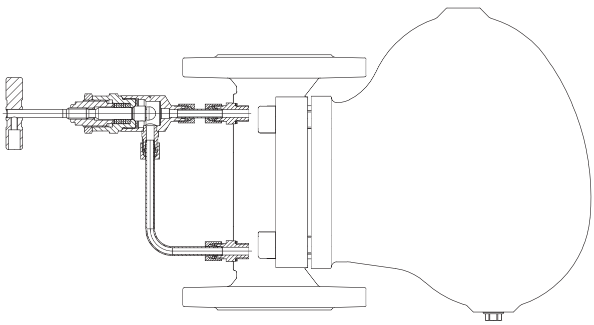


Fig. 7-3: Float trap with a needle valve for air venting, ARI Type CONA®S 631

Automatic air venting

The majority of manufacturers provide their float traps with automatic internal air venting in the form of thermostatic devices. Another alternative is to vent the trap by means of external, thermostatic traps mounted on the float trap body.

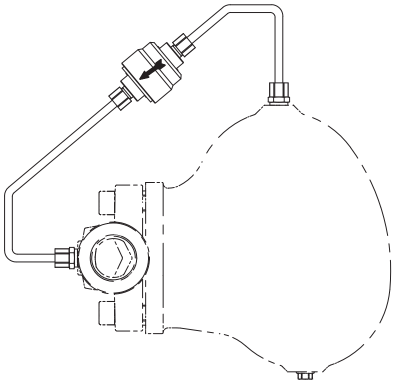


Fig. 7-4: Float trap with a external bimetallic air vent,
ARI Type CONA®S 631 with mounted ARI Type CONA®M 614

The float trap shown in Fig. 7-5 features an integrated thermostatic capsule for air venting. The capsule opens when cold and does not close until all air has been discharged and the corresponding saturation temperature almost reached.

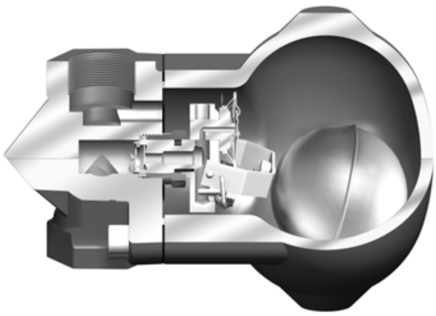


Fig. 7-5: Float trap with a bimetallic air vent (vertical installation),
ARI Type CONA®SC 634

The integrated thermostatic air vents are based on the principle that when air is present in the trap, the temperature of the air / steam mixture is lower than the saturation temperature of the corresponding steam pressure.

The float trap shown in Fig. 7-6 vents air automatically by means of a bimetallic element integrated into the valve mechanism.

In many cases, the buoyancy force of the ball float is not sufficient to overcome the pressure (and hence force) created by the pressure acting over the areas of the orifice. This problem has been solved in the configuration below where opening of the valve is assisted by steam pressure. The valve closes counter to the flow direction if condensate is forced back into the trap, in other words it doubles as a non-return valve. No condensate is returned to the heat exchanger from the condensate system if the plant is not in service.

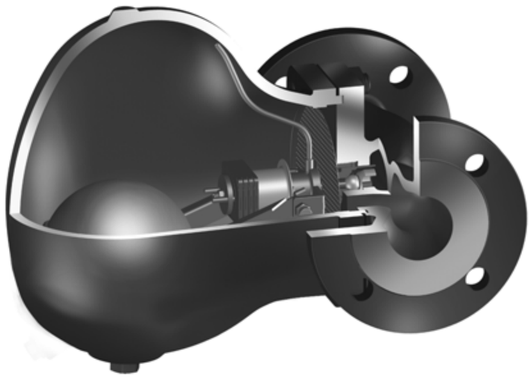


Fig. 7-6: Float trap with a bimetallic air vent (horizontal installation), ARI Type CONA®S 631

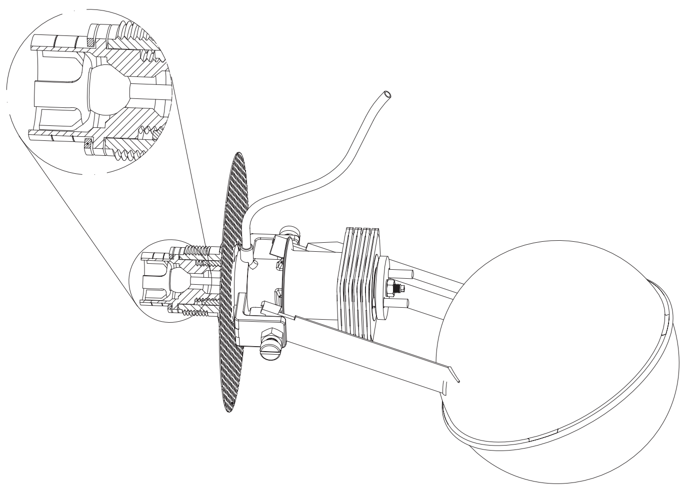


Fig. 7-7: Regulator unit with in-service air venting, ARI Type CONA®S 631

Applications

Float-type steam traps are usually more expensive to purchase than other types owing to their size and design. They are best suited for draining heat exchangers in process applications where the product temperature has to be precisely controlled within defined limits. The following three features represent their main advantages:

- The characteristic of a float trap follows the saturation curve, in other words the trap discharges the condensate boiling hot and without waterlogging (Fig. 7-8).
- A float trap modulates continuously according to the condensate level in the trap.

- A float trap is capable of discharging large amounts of condensate without losing steam.

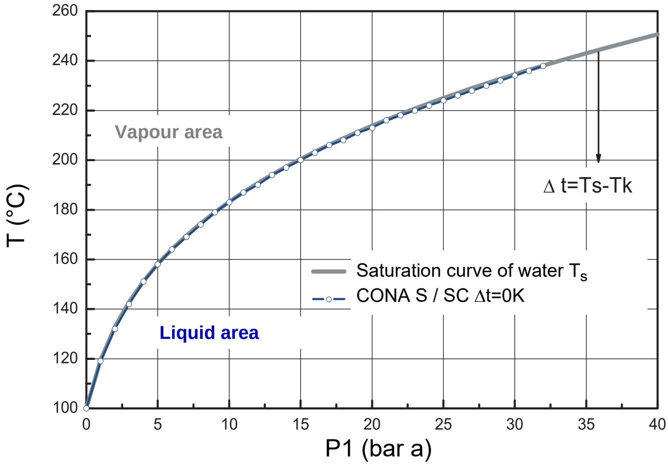


Fig. 7-8: Control characteristic of a float trap

Float traps are also suitable for critical drainage tasks or as level controllers in condensate flash vessels and condensate coolers.

Another point in favour of the float design is that its functionality is not in any way impaired by variations in the pressure or flow rate or higher back pressure. If a change in the operating state causes the differential pressure and the condensate flow rate to exceed the operating and performance limits of the installed regulator, it is possible to switch to another regulator with a different cross-section in the same trap body.

Installation requirements

Owing to the design of the float trap, there is always a small amount of residual liquid in the cover, with the associated risk of freezing when the system is removed from service. It is therefore advisable to provide the trap with an automatic drain valve, particularly in outdoor installations susceptible to frost or systems that are frequently shut down. (Fig. 7-9)

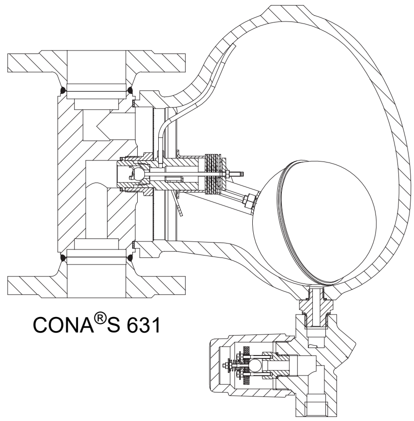


Fig. 7-9: Float trap with a start-up drain valve, ARI Type CONA®S 631 with CONA® 665

Both the float traps and the inlet and outlet pipes should be insulated to ensure safe working conditions, to prevent freezing and to reduce heat losses through radiation. On the other hand, insulating the supply pipe also creates a risk of a steam lock occurring in the pipe between the heat exchanger and the steam trap. The steam lock is formed when part of the condensate in the pipe evaporates upstream of the trap due to a sudden pressure drop in the exchanger. Especially if the piping arrangement is less than ideal and includes rising bends, these steam locks can be very stubborn and a cause of waterlogging in the heat exchanger. It is a good idea to install the float traps as close as possible to the steam user and to lay the supply pipe with a gradient in the direction of flow.

Float traps can also be used to drain compressed air systems. Owing to the low temperatures, a thermostatic air vent must be removed and plugged off for these applications otherwise compressed air would escape continuously.

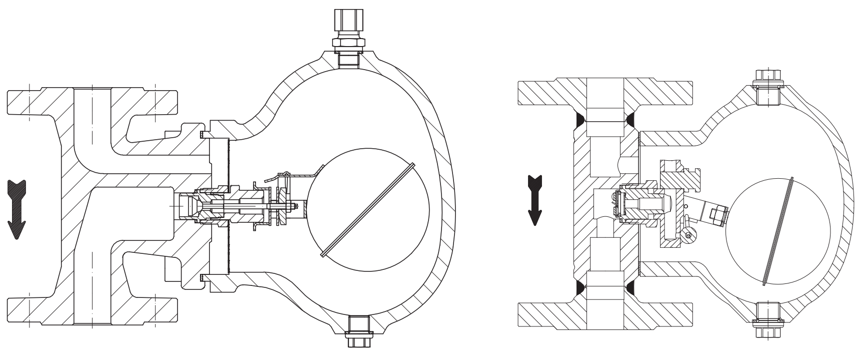


Fig. 7-10:Float trap with a start-up drain valve,
ARI Types CONA®S 630 and CONA®SC 636

Inverted bucket steam traps

Inverted bucket steam traps resemble a float trap that is open at the bottom. Like float traps, they discharge the condensate at saturation temperature though not continuously. Opening and closing of the outlet valve are directly controlled by the inverted bucket via a lever mechanism. A small air vent (bleed) hole is provided in the top of the bucket to prevent air from collecting underneath it.

Method of operation

To understand the operating principle of an inverted bucket steam trap, it is a good idea to start at the point in the operating cycle where the body of the trap is completely filled with condensate (refer to Fig. 7-11). The bucket rests on the bottom and the outlet valve is fully open. The pressure difference across the trap causes the condensate to be discharged directly into the header or the atmosphere. If steam flows under the bottom of the bucket, the bucket becomes buoyant, rises and starts to close the valve. The valve remains shut as long as the buoyancy force is greater than the weight of the bucket. When the weight exceeds the buoyancy force again, the bucket begins to sink and the outlet valve is opened.

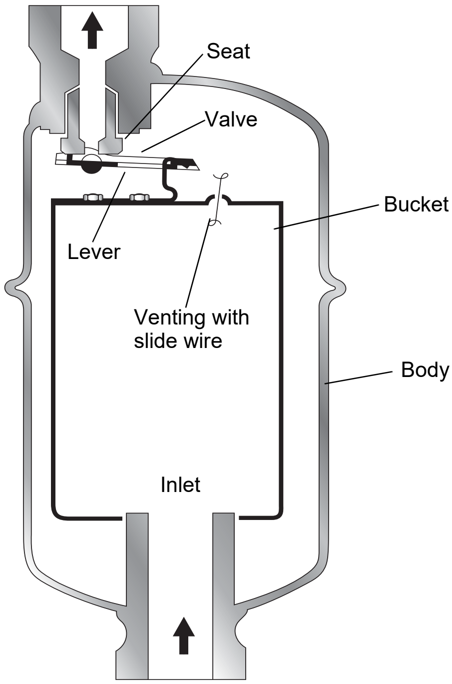


Fig. 7-11: Operation of an inverted bucket steam trap

Since it would take far too long for the steam bubble to condense underneath the bucket, a small hole is provided in the top to allow steam to escape into the top part of the body. When more condensate flows under the bottom of the bucket, the buoyancy force decreases and the bucket starts to sink. Finally, when the weight of the bucket exceeds the buoyancy force, the valve is opened by means of the lever arm and the condensate is discharged. The larger the pressure difference, the higher the level in the trap must rise in order to open the valve.

Starting up an inverted bucket steam trap

Extreme care is essential when starting up an inverted bucket steam trap. If there is no condensate inside the trap, as is the case when the system is put into service for the first time or following a repair, the bucket hangs right down and the outlet valve is fully open. A water seal, i.e. the presence of condensate in the body, is necessary for the inverted bucket to work. If the inlet valve upstream of the trap is opened when there is only steam in the pipe and no condensate, the steam will escape into the condensate system through the open trap. If the steam contains condensate, the latter is entrained due to the high velocity of the mixture and the condensate level in the trap builds up either very slowly or not at all. It is therefore advisable to collect the required amount of condensate in the trap prior to starting it up. This is done by opening the inlet valve but keeping the outlet valve downstream of the trap closed for the time being.

Steam wastage due to the absence of a water seal not only occurs when the trap is started up. If there is a sudden drop in pressure or if no condensate accumulates, the condensate that is needed to form the seal may flash to steam during operation. When the level falls, the bucket sinks to the bottom and the trap remains wrongly in the open position. It is hence a good idea to increase the length of the supply pipe to the trap slightly and to refrain from insulating it.

Applications

The main advantage of the inverted bucket steam trap is its insensitivity to dirt, because the outlet valve is located on the top of the trap. Dirt can thus collect on the bottom without damaging the valve seat or plug. The disadvantage of this trap type is that it can only vent air very slowly. Any air trapped under the bucket (for instance, when the system is started up) can only escape gradually through the small hole in the top. This is an important aspect in a system that is started up and shut down frequently. Live steam likewise escapes through the bleed hole during operation. The steam loss owing to the hole in the bucket is approximately 0.4 kg/h, equivalent to €65.- /year. Generally speaking, float-type steam traps – or bimetallic traps for pipe drainage and trace heating – should be preferred to inverted bucket traps if large amounts of condensate are involved, depending on the application.

7.5 Thermostatic steam traps

Introduction

This group comprises two main types:

- Steam traps controlled by a bimetallic element or thermoelastic action (bimetallic steam traps)
- Steam traps controlled by steam pressure (balanced pressure traps, liquid expansion traps)

Thermostatic steam traps are controlled according to the condensate temperature. A temperature difference must exist between the saturated steam and the condensate for a thermostatic trap to open and close. The valve only opens, in other words, after the condensate inside the trap has cooled down to a few degrees below the saturation temperature and closes before the temperature increases to that of the steam pressure. All thermostatic traps operate intermittently. The opening and closing temperature can be influenced by the trap's adjustment and design.

7.6 Bimetallic steam traps

Method of operation

The principle of a bimetallic steam trap is based on the thermoelastic action of bimetal elements. The bimetallic element consists of two strips of metal with different coefficients of expansion. The top strip has a higher coefficient than the bottom strip.

Fig. 7-12 shows two cold bimetal elements; the layers with the lower coefficient of expansion are facing one another. When the element is heated, these layers are deflected against one another (Fig. 7-13.). The higher the temperature, the greater the deflection.

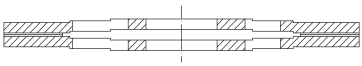


Fig. 7-12: Cold bimetallic plates

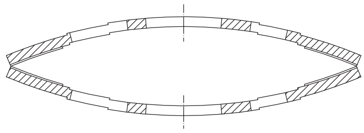


Fig. 7-13: Heated bimetallic plates

If several bimetal elements are stacked in pairs and connected by a valve stem, the changes in the temperature of the inflowing condensate cause the bimetal elements to be deflected, so that the plug opens and closes. Fig. 7-14 shows a cold bimetallic regulator. The bimetal elements are lying flat, one on top of the other, and the valve is fully open. Condensate and air are discharged swiftly under the influence of the pressure difference between the steam and condensate systems.

When the bimetal elements are heated by hot condensate, they are deflected and the stack expands. If the temperature rises to a few degrees below the saturation temperature, the expansion of the bimetal elements causes the valve to close (refer to Fig. 7-15). The closing force (F_S) of the bimetallic regulator is greater in this state than the opening force ($F_Ö$) generated by the system.

Principle of a bimetallic steam trap:

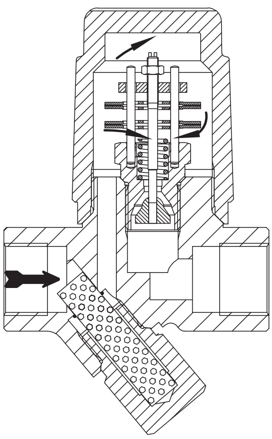


Fig. 7-14: Valve open

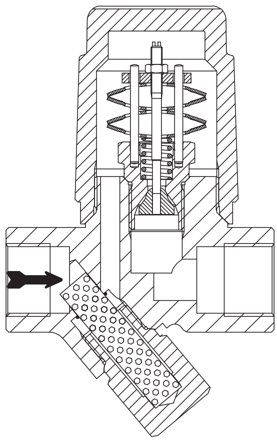


Fig. 7-15: Valve closed

The expansion of the bimetal elements thus not only results in the closure of the valve but also produces the force that is necessary to keep the valve permanently shut at saturation temperature. In the closed state, the force of the steam or system pressure acts on the valve and attempts to open it.

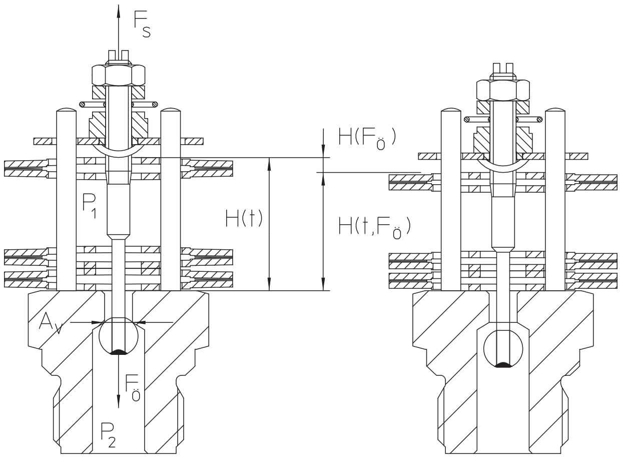


Fig. 7-16: Interaction of the forces in a bimetallic steam trap

The following is true: $\boldsymbol{F_ö = (p_1 - p_2) \times A_V}$

Where: $F_ö$ = Opening force of the valve
p_1 = Pressure upstream of the steam trap
p_2 = Back pressure in the condensate system
A_V = Active surface area of the valve

To keep the valve shut, the bimetal stack must apply a closing force (F_S) that is greater than the opening force ($F_Ö$). The closing force (F_S) decreases as the condensate cools down. When a defined temperature is reached, the system pressure acting on the valve area overcomes the closing force due to the expansion of the bimetal elements and causes the valve to open. Since the valve plug moves in the opening direction, the spring pressure of the bimetal stack also increases. A new balance is created between the opening and closing forces, depending on the condensate temperature. The majority of bimetallic steam traps are set to open at approximately 15 degrees below boiling point and close again at around 5 degrees below saturation temperature.

Influence of back pressure on the opening and closing torque

The factor (p_1 - p_2) in the above formula decreases as the back pressure in the condensate system increases. The opening force ($F_Ö$) is therefore reduced and the closing force (F_S) must also be reduced in order for the valve to open. The condensate in the steam trap must continue to cool down, so that the expansion of the bimetal elements decreases. The trap remains closed until this new balance is established. Put another way, an increase in back pressure is accompanied by increased condensate sub-cooling. "Sub-cooling" is defined as the difference between the condensate temperature and the saturation temperature. In contrast to a float-type steam trap, the characteristic of a bimetallic trap is influenced by the back pressure.

This is aptly illustrated by an example:
A bimetallic regulator is set to the standard sub-cooling temperature, i.e. approximately 15 degrees. Assuming a pressure p_1 = 10 bar (a) [T_s = 180 °C] upstream of the steam trap and atmospheric pressure [p_2 = 1 bar (a)] downstream of the trap, the trap will open at approx. 165 °C. If the back pressure rises to 4 bar (a) [T_s = 144 °C], the trap opens at 15 degrees below the saturation temperature (159 °C) for 10 - 4 = 6 bar, that is to say at approx. 144 °C.

This higher condensate sub-cooling at back pressure can be critical in certain applications (trace heating). In this case, it is advisable to correct the setting on the bimetallic regulator by adjusting the valve clearance. In other applications, the higher condensate sub-cooling brings with it the often desirable effect of reducing flashing. Owing to the smaller pressure difference at higher back pressure as well as the lower opening temperature, the percentage of flash steam downstream of the steam trap is smaller.

Combination of thermostatic and thermodynamic valve control

If the condensate in the steam trap cools down sufficiently for the opening force to start to overcome the closing force of the bimetal elements, the valve opens slightly. In order to open it further, the condensate needs to continue cooling. Since this is not normally the case, condensate will exit at very high velocity through the small gap between the valve plug and the seat. This hypercritical velocity will lead to erosion due to cavitation on the seat and plug, preventing the valve from closing tightly. Modern bimetallic steam traps get round this problem by combining the thermostatic with the thermodynamic principle. Chapter 4.0 Pipes includes a description of a full-lift safety valve. The stem of this valve is connected to a valve disc. If the system pressure rises enough to overcome the spring force that is keeping the valve shut, the outflowing medium lifts the disc and opens the valve completely. This principle is also employed in the regulator shown below (Fig. 7-17). The valve obturator takes the form of a plug rather than a simple ball. When the valve begins to open, the outflowing condensate hits the plug. The pressure inside the chamber (p_z) rises spontaneously owing to the small gap area. The plug acts like a piston, causing the valve to open fully. Condensate and dirt can thus be discharged over a larger effective area.

When the condensate temperature rises, the deflection of the bimetal elements increases and the valve begins to close. The flow velocity increases according to the thermodynamic principle (Bernoulli) in the ever smaller gap between the seat and the plug. The pressure inside the chamber (p_z) falls and the evaporating condensate downstream of the valve plug unloads the stem, thus supporting the closing movement through the increased deflection of the bimetal elements. The valve closes and remains closed until the condensate cools down and the cycle repeats. This intermittent mode of operation reduces the time for which condensate flows out at critical velocities between the seat and the plug, with the result that the service life of the bimetallic regulator is extended, especially at high service pressures.

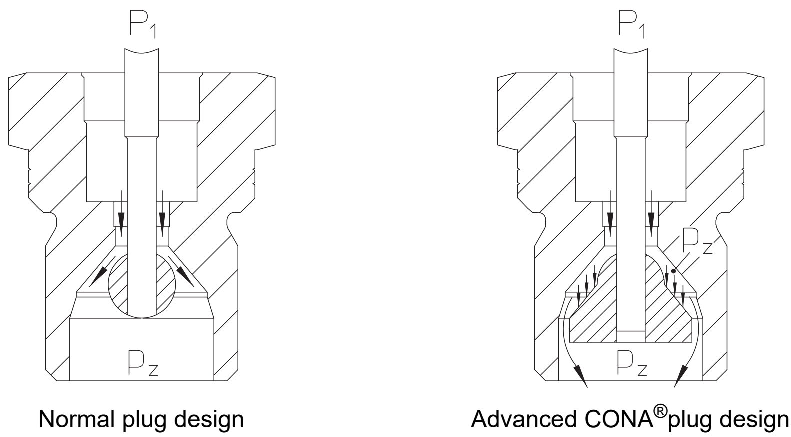


Fig. 7-17: Principle of thermodynamic valve opening

Design of the bimetal elements

Fig. 7-8 shows the control characteristic of a ball float trap. The opening characteristic coincides exactly with the saturation curve, in other words boiling hot condensate is discharged at the corresponding pressure. The opening characteristic of a bimetallic steam trap, on the other hand, is always a few degrees below the saturation curve. It can be seen from Fig. 7-18 that the characteristic curve of a simple bimetallic trap is linear rather than following the saturation curve. In practice, this means that sub-cooling is not constant over the pressure range. The linear characteristic of a simple bimetallic regulator is presented in Fig. 7-18 in relation to the saturation curve. The trap operates very close to the saturation temperature in the low and high pressure ranges but with relatively high sub-cooling in the mid-range area.

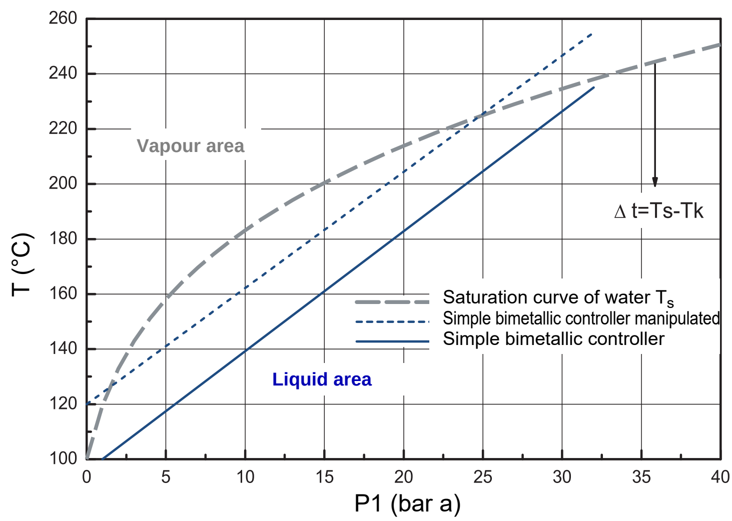
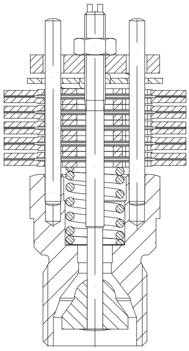


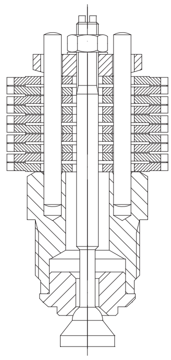
Fig. 7-18: Characteristic of a simple bimetallic steam trap

The characteristic can be shifted closer to the saturation curve or closer to sub-cooled condensate by altering the valve clearance. If it crosses the saturation curve, steam is lost in the areas above the curve. This risk is particularly great in the low pressure range as well as close to the maximum pressure.

The CONA®B bimetallic trap differs from this simple type in a number of key respects, offering intermittent operation, less wear, a longer service life and almost constant sub-cooling over the entire pressure range for which the regulator is designed (refer to Fig. 7-19).



PN 16-100



PN 160-630

Fig. 7-19: Bimetallic steam traps, ARI Type CONA®B

The bimetal elements in the trap shown here are not flat but stepped (Fig. 7-12, Fig. 7-13, Fig. 7-19). The stamped elements provide a larger heat contact surface (90 % compared to 55 % for flat elements), which means the trap reacts very quickly to temperature changes and are less sensitive to dirt. The stamped elements additionally support the spring characteristic of the bimetal stack.

By combining the thermostatic and thermodynamic principles, it is possible to achieve intermittent operation with spontaneous opening of the valve, followed by a long cooling phase. This function was described in detail in the previous section and the special design of the valve plug is illustrated in Fig. 7-19.

The integration of a spring in the bimetallic regulator results in a steeper characteristic in the low pressure range and guarantees almost constant sub-cooling throughout the pressure range of the regulator. The valve lift is calculated and adjusted so that the characteristic remains below the saturation curve and no live steam is lost (refer to Fig. 7-20).

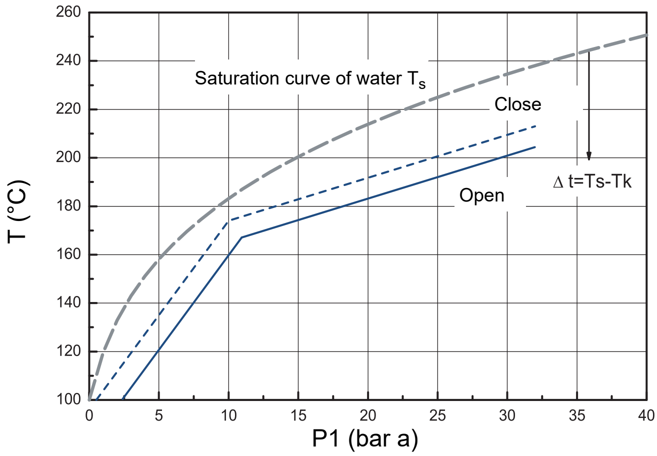


Fig. 7-20: p-t saturation curve with characteristic of a ARI Type CONA®B

Sub-cooling setting

Some designs allow the steam trap to be adjusted externally while it is operating. This is particularly important with simple bimetallic traps where the characteristic or the sub-cooling degree needs to be adapted to the respective service pressure. If this adjustment is carried out by a person without the appropriate technical know-how, it inevitably increases the risk of steam leakage and undesirable backing-up (waterlogging). Fig. 7-21 shows bimetallic steam traps that are configured for 15 Kelvin sub-cooling in the factory. These traps are designed to operate automatically with almost constant sub-cooling over the full pressure range. There is consequently no need to alter the setting during operation. Higher sub-cooling (30 Kelvin, 40 Kelvin, etc.) can be set in the factory if required.

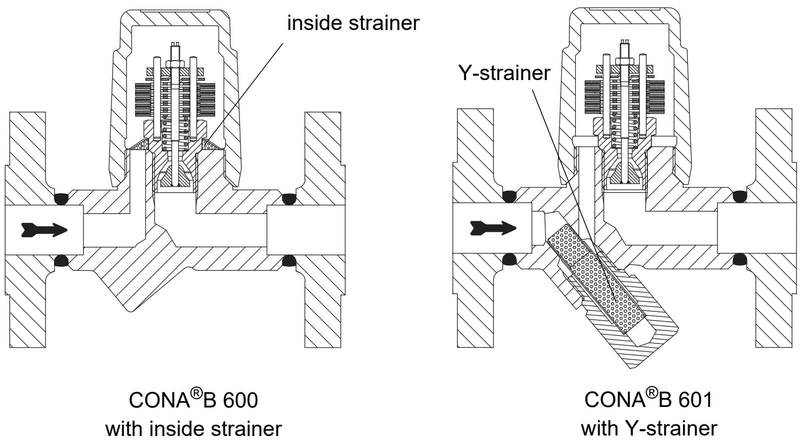


Fig. 7-21: Bimetallic steam traps with adjustable sub-cooling, ARI Types CONA®B 600/601

The option of setting a steam trap to different sub-cooling degrees has the advantage that the heat from the hot condensate can be utilised commercially, for example for trace heating processes. At the same time, multiple sub-cooling settings enable the percentage of flash steam to be reduced.

Tip:

Adjustable sub-cooling should be avoided with thermostatic traps, e.g. for draining heat exchangers or steam pipes. In the worst case, too much sub-cooling could cause condensate to back up right into the steam pipe or the heat exchanger, with water hammer as the result.

Check valve function

If the steam pressure in the heat exchanger drops below the pressure in the condensate system as a result of reduced heat demand, or if a user connected in parallel is shut down, condensate could flow back or be drawn into the exchanger. The bimetallic steam trap in Fig. 7-21 features an integrated check valve that reliably prevents return flows of this kind.

Air venting

A bimetallic steam trap is open in the cold state and remains open until the condensate temperature rises to a few degrees below the saturation temperature of the steam pressure. When a steam pipe, a trace heater or another steam user is started up, the air it contains is discharged swiftly together with the cold condensate. Bimetallic traps are thus suitable both for start-up venting and as thermostatic air vents in continuous operation.

Applications of bimetallic steam traps

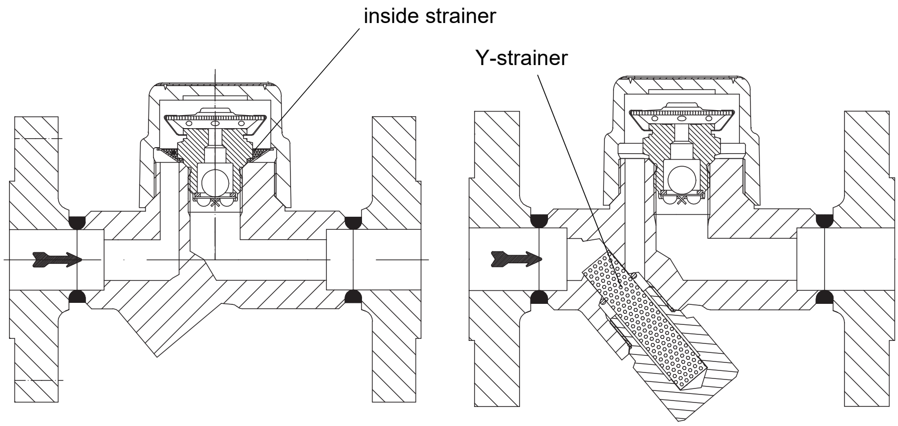
Bimetallic steam traps can be used for a wide range of drainage and venting tasks, such as pipe drainage, trace heaters, drainage systems, heat exchangers or air heaters, in which the condensate back-up due to the design does not impair the process or lead to water hammer. Traps with adjustable sub-cooling are sometimes recommended in trace heaters used for frost protection in order to utilise the sensible heat of the condensate. If a bimetallic trap is used to drain steam pipes, the globe valve upstream of the trap and the trap itself should not be insulated. Bimetallic steam traps are excellent air vents in heating systems with free air access that are frequently started up or shut down. They are also ideally suited as air vents when large steam spaces are put into service, for example in autoclaves. Bimetallic traps are extremely robust and insensitive to water hammer, making them an attractive alternative for all applications where water hammer would be a serious problem.

Steam traps controlled by steam pressure (balanced pressure traps)

The most important representatives of this thermostatic trap class are designed either with a membrane capsule for pressure balancing or with a bellows for liquid expansion. Only the membrane capsule type in common use today is described here.

Method of operation

The steam traps shown in Fig. 7-22 work according to the vaporisation principle. A control liquid is trapped inside a stainless steel capsule between the membrane and the capsule body. The boiling point of this liquid must be equal to or less than the boiling point of water at the corresponding pressure. The opening and closing movements are caused by imbalance between the condensate pressure in the steam trap and the steam pressure of the liquid in the membrane capsule. The liquid enclosed in the capsule vaporises as the temperature in the steam trap rises, resulting in a gauge pressure that forces the membrane against the valve seat and shuts the trap a few degrees below the saturation temperature. The trap opens again when the control liquid condenses as the condensate cools down.



CONA®M 611
with inside strainer

CONA®M 613
with Y-strainer

Fig. 7-22: Balanced pressure steam traps, ARI Types CONA®M 611/613

Fig. 7-23 a and Fig. 7-23 b illustrate the principle of a balanced pressure steam trap in more detail. The trap's opening and closing temperature at the corresponding pressure can be influenced by varying the composition of the control liquid. By altering the filling, in other words, it is possible to obtain membrane capsules for different sub-cooling degrees.

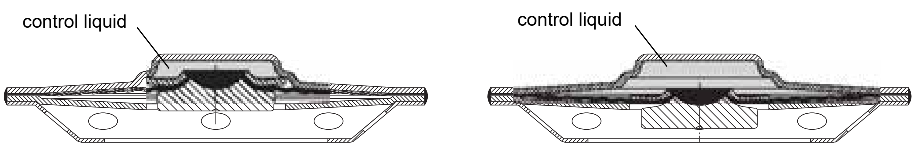


Fig. 7-23 a: For cold condensate
ARI Type CONA®M 611/613

Fig. 7-23 b: For hot condensate
ARI Type CONA®M 611/613

The membrane capsule with the standard filling opens and closes at 10 Kelvin below saturation temperature. This is also referred to as 10 Kelvin sub-cooling. Membrane capsules with higher sub-cooling are especially suitable for trace heaters and – as mentioned in the previous chapter – as a way to reduce flashing.

Membrane capsules for different condensate sub-cooling degrees

Various membrane capsules with different sub-cooling degrees and different fillings are available for the CONA®M series and can be replaced individually (refer to Fig. 7-24). Capsule 1 discharges the condensate at about 3 to 5 Kelvin below the saturated steam temperature and thus has a similar characteristic to mechanical traps. Capsule 2 has approximately 15 Kelvin and capsule 3 approximately 25 Kelvin sub-cooling.

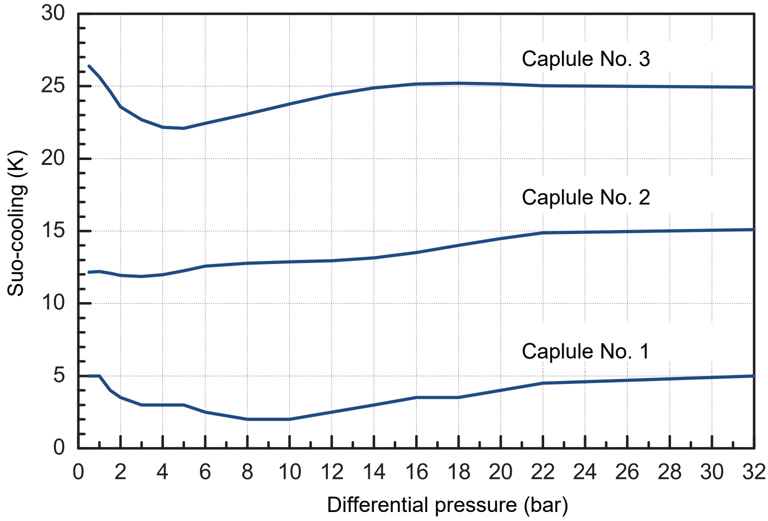


Fig. 7-24: Characteristic with different membrane capsules,
ARI Type CONA®M 611/613

The number of the capsule is stamped on the rating plate and on the capsule itself for easier identification and to avoid confusion.

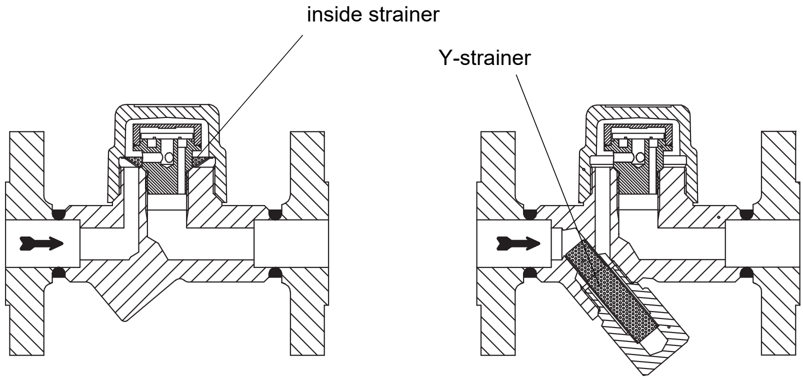
Applications

Balanced pressure steam traps are the solution of choice in trace heaters as well as for draining and venting small vessels with double jacket heating. They are fast and efficient air vents. They are also used as aerators for steam spaces (vacuum breakers) in models without a check valve function. Like their bimetallic counterparts, balanced pressure traps should not be insulated.

7.7 Thermodynamic steam traps

Steam traps with non-pressure balanced valve disc

Thermodynamic steam traps are controlled according to the thermodynamic state of the fluid. The simplest form of thermodynamic trap consists of no more than a body with an integrated valve seat, a valve disc and a screw cap (refer to Fig. 7-25).



CONA®TD 640
with inside strainer

CONA®TD 641
with Y-strainer

Fig. 7-25: Thermodynamic Steam Trap, ARI Type CONA®TD 640/641

The two seat rings in the body are hardened (like the valve disc) and their surface has a fine ground finish. The inlet orifice is centrally positioned in the seat. A small, circular duct is provided around this orifice, with three holes connected to the outlet orifice. When the valve disc is resting on the seat, it simultaneously closes both the inlet orifice and the duct with the outlet holes.

Method of operation

The principle of the thermodynamic steam trap is based on Bernoulli's law, which states that the sum of the static pressure (potential energy) and dynamic pressure (kinetic energy) is constant at all points in a fluid flow (gas or liquid). If the static (or gauge) pressure falls, the velocity (or dynamic pressure) rises and vice versa.

Pressure changes occur when condensate passes the valve disc at saturated steam temperature and is partially flashed due to the lower pressure in the condensate system. The operation of a thermodynamic steam trap is determined by the change in static and dynamic pressure. Its action can best be explained with the help of an example:

If cold condensate flows into the trap on start-up, the valve disc is forced upwards and the condensate is discharged via the outlet holes. The trap is fully open. As the start-up phase progresses (Fig. 7-26 a), the condensate becomes hotter and the pressure rises. Part of the static pressure is then converted to velocity in the chamber between the seat and the disc.

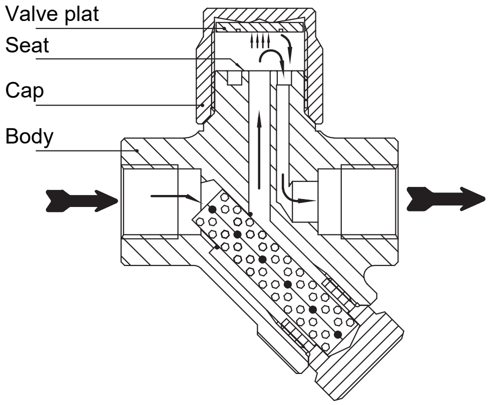


Fig. 7-26 a: "Pressure build-up" state

The increase in kinetic energy causes the pressure to drop and the condensate begins to flash. At the same time, there is a sharp rise in the volume and velocity. The condensate and the flash steam flow along the underside of the valve disc at high velocity. This velocity increase results in a pressure drop underneath the disc. Part of the flash steam enters the chamber above the disc and is drawn towards the inside of the cap.

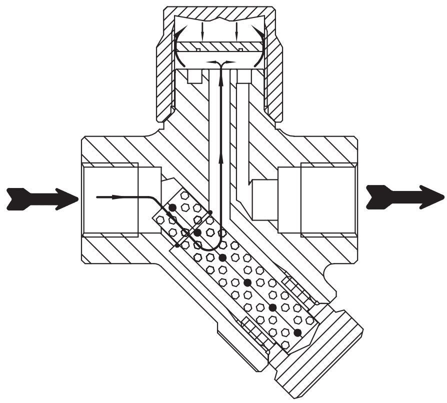


Fig. 7-26 b: "Flashing" state

The steam pressure that is built up above the valve disc forces it against the seat and shuts the valve (refer to Fig. 7-26 b). The trap remains closed under the influence of the difference between the active surface area above the disc and under it.

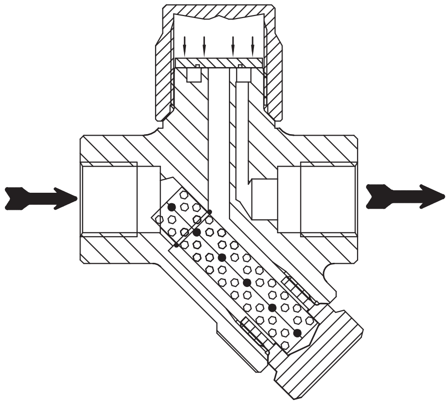


Fig. 7-26 c: "Closed" state

Heat is dissipated into the atmosphere via the cap of the body (refer to Fig. 7-26 c). As a result, the steam in the chamber above the valve disc condenses. The pressure above the disc decreases and is no longer sufficient to close it against system pressure. The valve disc opens and the cycle repeats.

Applications

Thermodynamic steam traps are likewise used for trace heaters and pipe drainage. They are insensitive to water hammer and suitable for a broad pressure range. However, the thermodynamic principle also suffers from several restrictions and disadvantages in practice:

- Thermodynamic traps should not be used if high back pressure occurs, i.e. greater than 60 % of the upstream pressure. The reason for this is that with a pressure difference less than this not enough condensate is flashed under the valve disc, so the pressure drop is not large enough to close the valve. Thermodynamic traps are not suitable for applications with fluctuating pressure, high back pressure or significant variations in the amount of accumulated condensate, as is often the case with heat exchangers that are regulated on the steam side.
- Thermodynamic traps are poor air vents. Air is contained both in the system and in the trap on start-up. According to Bernoulli's law, the valve disc is forced against the seat if air attempts to escape at high velocity. The trap remains closed and does not vent the air.
- Thermodynamic traps must be opened from time to time in order to renew the steam cushion above the disc and build up the required closing pressure. If the steam does not contain any condensate, for example with superheated steam, live steam will be released via the trap.
- With outdoor installations, the condensation of the steam cushion above the valve disc is easily influenced by the ambient conditions. Rain or wind will cause the opening frequency of a thermodynamic trap to increase, leading inevitably to wear.

7.8 Fixed orifice devices

Design and method of operation

Fig. 7-27 shows a fixed orifice device. The condensate enters the trap via a calibrated orifice and exits it again via an expansion chamber connected to the condensate pipe.

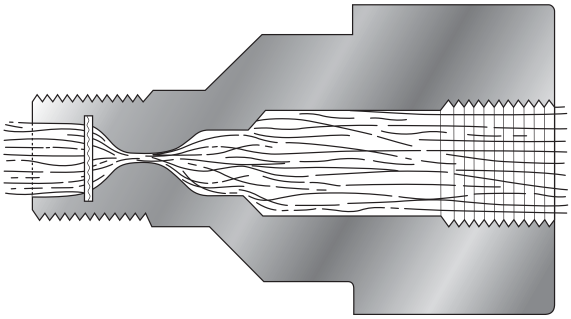


Fig. 7-27: Fixed orifice steam trap

The orifice plate and the expansion chamber are sized according to the maximum expected amount of condensate and the design service pressure (pressure upstream of the trap). Liquid flows freely through the orifice. Hot condensate that passes the orifice flashes in the expansion chamber. Owing to the enormous volume of the flash steam, the flow of steam and non-condensables is reduced and possibly prevented altogether. This blocking function is influenced by the dimensions of the expansion chamber and the length of the orifice. A labyrinth or multi-stage orifice can be obtained by connecting several chambers and orifice plates in series. Labyrinth and orifice traps are fixed (static) and cannot adapt to changing operating conditions. The fixed hole size means the maximum flow rate is restricted. If a larger amount of condensate builds up, waterlogging will occur. If the orifice is too large or if the steam does not contain much condensate, live steam will escape as well.

Applications

The potential applications for orifice traps are limited. This drainage method only works reliably under constant operating conditions. There should be little or no variation in the heat load, upstream pressure, pressure difference or condensate amount. Orifice traps must not be used to drain steam pipes. When a pipe is put into service, a large amount of condensate is formed at low steam pressure, so that an orifice with a large hole is required. Actually in service, on the other hand, only a small amount of condensate occurs at a higher steam pressure, so that a small hole suffices. Impurities must be reckoned with when a system is started up for the first time and the small orifices of this trap type can easily become blocked. Orifice traps are thus relatively susceptible to dirt. In addition, they have no automatic air venting function. Air must be vented manually on the heat exchanger, especially if the system is started up and shut down frequently.

8.0 Test Systems for Steam Traps

8.1 Introduction

The following factors are particularly important when testing the operation of steam traps:

- Live steam leakage
- Condensate back-up

8.2 Live steam leakage

Testing steam traps for loss of live steam by measuring the temperature at leaking traps is not as easy as it is often assumed to be. Unfortunately, a temperature measurement alone does not provide any indication of whether a trap passes steam or condensate at saturation temperature. With the exception of thermostatic traps, the temperature of the condensate directly upstream of the trap is always the same as the saturated steam temperature. If a float, inverted bucket or thermodynamic trap is used, the condensate temperature upstream of the trap is exactly identical to the saturation temperature of 10 bar saturated steam, in other words 180 ºC, for a heat exchanger with 10 bar live steam.
Temperature measurements on thermostatic steam traps are somewhat more complicated owing to their cyclic method of operation. A value close to saturation temperature is measured on the condensate side (return) immediately after the trap shuts. Slight sub-cooling can be determined as soon as it opens. If the back pressure in the condensate system is 5 bar, the saturation temperature is 152 ºC. If only steam (with a pressure of less than 30 bar) is allowed to pass by the trap, it can superheat in the condensate system. On the other hand, if just 4 % condensate is present in the steam, the amount of heat required to revaporise this condensate is so great that no further superheating is measured downstream of the trap.

8.3 Condensate back-up

If condensate backs up in a heat exchanger, part of the heat exchange surface is flooded. The area available for heat transfer is reduced and the condensate temperature upstream of the trap falls below the saturation temperature. If no condensate is discharged at all, the temperature drops to that of the product being heated. If condensate is discharged at less than saturation temperature, we refer to it as "sub-cooled" condensate.

Backing-up can occur in any type of steam trap as a result of:

- Pressure differential across the trap too small
- Strainer screens clogged
- Insufficient trap capacity

In a float-type steam trap, backing-up can also be caused by the ball float filling with condensate due to a leak, so that the valve remains closed. Inverted bucket traps are prone to backing-up if the bleed hole is obstructed. In a thermodynamic trap, the problem may arise if there is air above the valve disc. Thermostatic traps can be affected if the cooling leg is too short or the trap is incorrectly adjusted. Condensate back-up can be determined by measuring the temperature. Since there are not normally any test points upstream of the trap, the measurement is usually based on the surface temperature. These measurements have a tendency to be unreliable and errors are not uncommon. To restrict them to a minimum, it is a good idea to measure not only the surface temperature upstream of the trap but, if possible, also that of the steam pipe directly downstream of the steam control valve. By comparing the two measurements, you can then establish whether or not condensate has backed up upstream of the trap. If you use an infrared thermometer, the two test points must be made of the same material because measurements on stainless steel cannot be compared with the results for carbon steel, for instance.

If no condensate has backed up, the temperature upstream of a float, inverted bucket or thermodynamic trap is identical to the saturated steam temperature (with a tolerance of approximately 5 °C). Backing-up occurs if the temperature upstream of the steam trap is less than that downstream of the steam control valve and simultaneously less than the saturation temperature.

With thermostatic traps (bimetallic or balanced pressure), the temperature upstream of the trap varies from approximately saturation temperature on closing to about 15 Kelvin below saturation at the moment of opening (bimetallic). The temperature difference for a balanced pressure steam trap should be a maximum of 5 Kelvin. If the difference exceeds this value, we speak of "condensate back-up".

It should be noted that the sub-cooling degree of bimetallic traps increases with the back pressure in the condensate pipe. If a bimetallic trap is set to higher sub-cooling or a balanced pressure trap fitted with a sub-cooling capsule, the temperature profile will be completely different, of course.

8.4 Test methods

The following methods are used to test or monitor a steam trap's operation:

- Visual inspection
- Noise measurement / ultrasonic testing
- Conductivity measurement
- Calorimetric testing

Visual inspection

The operating behaviour of a trap that discharges into the atmosphere can usually be tested relatively easily by carrying out a visual inspection. Apart from float traps, all other trap types work intermittently, in other words they remain open for a while and are subsequently closed for a time. When a trap with a cyclic method of operation opens, the condensate / flash steam mixture is discharged. The trap shuts at the instant that hot condensate reaches a few degrees below saturation temperature. If the trap does not have a tight seal, a steam cloud will be visible at the outlet, giving the impression that steam is leaking. In this case, the trap must be replaced or cleaned. With the exception of float-type steam traps, almost all traps that drain into the atmosphere can be monitored very effectively in this way.

If the trap drains into a closed system, a drain device must be installed downstream of the trap to enable it to be tested in the manner described above. The drain valve opens and the condensate is intermittently discharged into the atmosphere. With an inverted bucket steam trap, the method of operation can change to continuous drainage, for example if the load is too low. A thermodynamic trap that opens and shuts more than five times a minute is not working correctly and is passing steam. A method for determining steam losses is described at the end of this chapter. Note that all live steam leakage of just 1 kg/h represents a financial loss of €200.- over the course of a year.

Visual inspections are certainly not the most reliable way to assess a steam trap's operation. In many cases, especially when pipes are drained into the atmosphere, flashing is incorrectly interpreted. Every time the trap opens, the flash steam appears at the outlet as a steam cloud. This condensate, which is flashed to atmospheric pressure and subsequently revaporised, are erroneously thought to be leaked live steam. Steam traps are often set to a higher sub-cooling degree in order to reduce flashing. This is not recommended, however, because sub-cooling the condensate excessively can lead to water hammer damage.

Sight glasses

Sight glasses for observing fluid flow should normally be installed upstream of the steam trap. If the trap is operating correctly, the amount of condensate will be indicated by the water level in the sight glass (Fig. 8-1 a). If backing-up occurs inside the trap, the sight glass will be completely filled with condensate (Fig. 8-1 b). In a trap that is leaking live steam, the condensate level will be forced below the level of the sight glass inlet lug (Fig. 8-1 c).

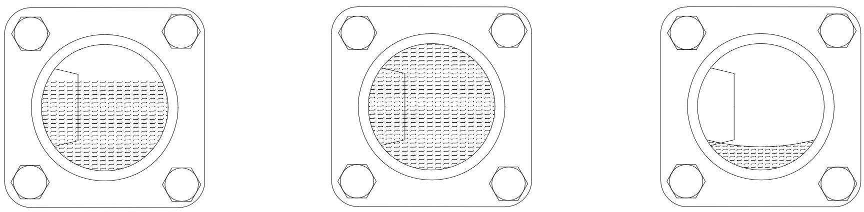


Fig. 8-1 a:
Normal operation

Fig. 8-1 b:
Condensate back-up

Fig. 8-1 c:
Live steam leakage

If the sight glass is mounted downstream of the trap on the condensate side, it is not always easy to distinguish between live steam loss and flashing. In both cases, a mixture of steam and condensate will be visible in the viewing window. The principal disadvantage of sight glasses and water level indicators is that they constitute an additional source of leakage and that the window may become clouded after a long period in operation or under the influence of chemicals in the condensate. The arrangement of the steam traps is another problem. They are normally installed at the lowest point or possibly on a pipe bridge, which means the sight glasses are comparatively inaccessible and difficult to observe.

Noise measurement / ultrasonic testing

Ultrasonic test devices measure the high-frequency vibrations produced by steam, gases or liquids flowing through the traps using a contact transducer. These vibrations have a frequency that is inaudible to the human ear. Fig. 8-3 shows a simple ultrasonic testing device.

This detector allows the opening and closing movement of an intermittently operating trap to be precisely reconstructed. With continuously operating float traps, the sound level can be taken as a benchmark. The assessment of the sound level is subjective and a certain level of experience is essential.

The trap can be assumed to be operating correctly if the ultrasonic signal is close to a limit value and modulates slightly. The limit value varies according to the steam pressure and the trap type (Fig. 8-2).

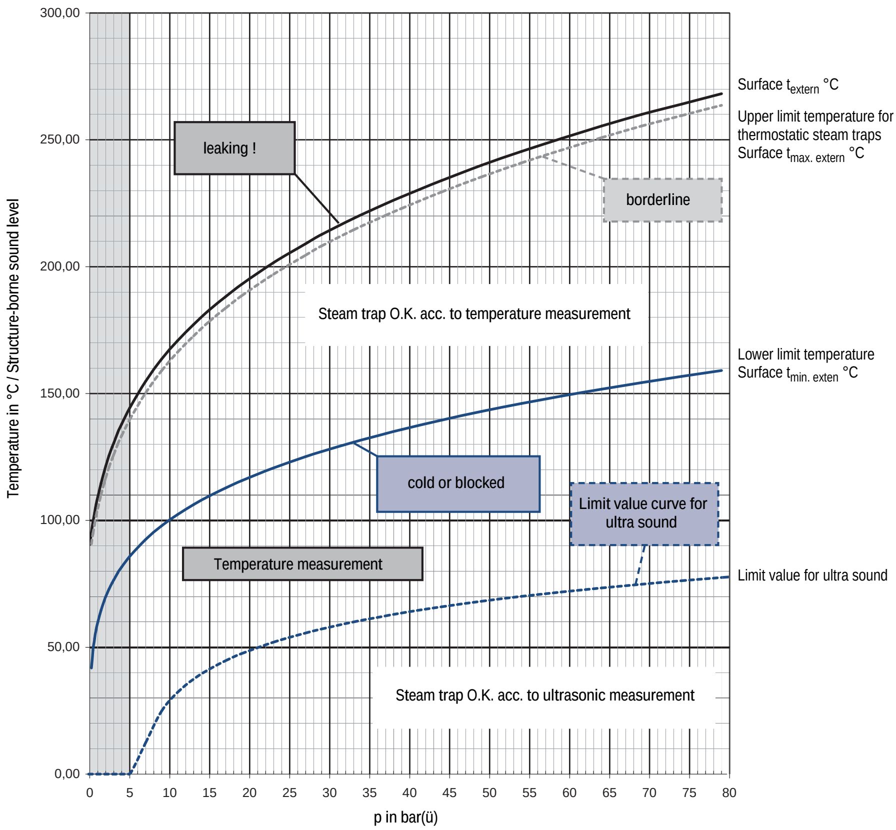


Fig. 8-2: Limit value curve

If the limit value is exceeded, the trap is probably leaking steam. The measured value is influenced by ambient noise and extraneous sound sources (steam pressure reducing valves, control valves, steam turbines). Ambient noise can usually be filtered out, except in the immediate vicinity of the trap, where it may be too loud.

In this case, the globe valve downstream of the trap can be shut for a while. Almost all of the ultrasonic level that is measured now can be attributed to the extraneous sound sources because there is nothing flowing through the trap. If a higher overall sound level is ascertained when the globe valve is opened, the difference compared to the previous level must be due to the trap.

The ultrasonic test itself often does not provide any indication of whether a steam trap is passing steam or condensate at a specific moment in time. It should be combined with a surface temperature measurement for this reason. A combination test device is shown in Fig. 8-3. In addition to the ultrasonic level, this tester also measures the surface temperature of the trap. The measured ultrasound and temperature values are visualised on the tester's display.

Fig. 8-3: Combined temperature / ultrasonic testing device, Type Multifunction tester

The measured values are compared with a limit value that varies according to the steam pressure and trap type. If they remain below this limit throughout the measurement phase, the trap is operating correctly and there is no leakage of live steam. If the limit value is exceeded, the trap is passing steam. The amount by which it is exceeded corresponds to the quantity of steam lost. All the values measured can be stored in the device and later processed on a PC. The operation of the various traps can thus be monitored over a long period of time. The temperature sensor compares the measured temperature with the saturation temperature to determine whether or not condensate has backed up upstream of the trap.

If the steam pressure downstream of the pressure control valve is known, the saturation temperature can be read off from the steam table. If it is not known, we recommend calculating the saturation temperature using the surface temperature measurement method described earlier in this chapter.

Assessment of the measured values for float traps

Assuming the trap is functioning correctly, only a float-type steam trap will produce a continuous ultrasonic signal because it discharges condensate continuously. If the values measured for a float trap (with a specific steam pressure) remain below the limit value, only condensate is being discharged. Fig. 8-4 shows a printout for a trap that is leaking live steam. To make sure that the high sound value is not due to pressure reducing valves or other apparatus in the vicinity, it is advisable to eliminate ambient noise using the method described above.

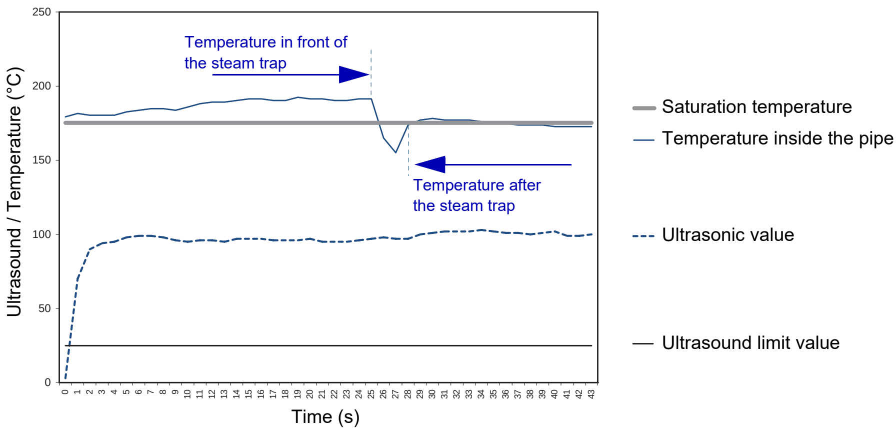


Fig. 8-4: Printout for a float trap that is leaking live steam

This signal exceeds the limit value throughout the measurement phase. There is no ambient noise that could distort the measured values. The surface temperature is identical to the saturation temperature. Conclusion: The trap is passing steam.

Assessment of the measured values for thermostatic traps

Thermostatic steam traps do not drain condensate continuously, so the measured ultrasonic signal has gaps (the same is also true when it comes to thermodynamic and inverted bucket traps). The curve recorded for a thermostatic trap during drainage is likely to be partly above and partly below the limit value. The fact that a measured value is higher than the limit value does not necessarily mean the trap is passing steam. Only if the measured value does not return to zero at the end of the trap cycle is it accurate to speak of steam loss. "Zero" means simply ambient noise, in other words the value that is determined when the trap is shut. The zero value based on ambient noise can sometimes be greater than zero. If the sound value measured for a thermostatic trap does not return to zero again at the end of the cycle, a comparative temperature measurement upstream of the trap can provide an indication of whether it is working correctly. If the temperature is close to or greater than the saturation temperature, the trap is passing steam. If a thermostatic trap has a thermodynamically amplified conical movement, the ultrasonic image is often extremely sharp. Fig. 8-5 and Fig. 8-6 respectively show the signal printout for a thermostatic trap that is operating correctly without and with thermodynamic amplification.

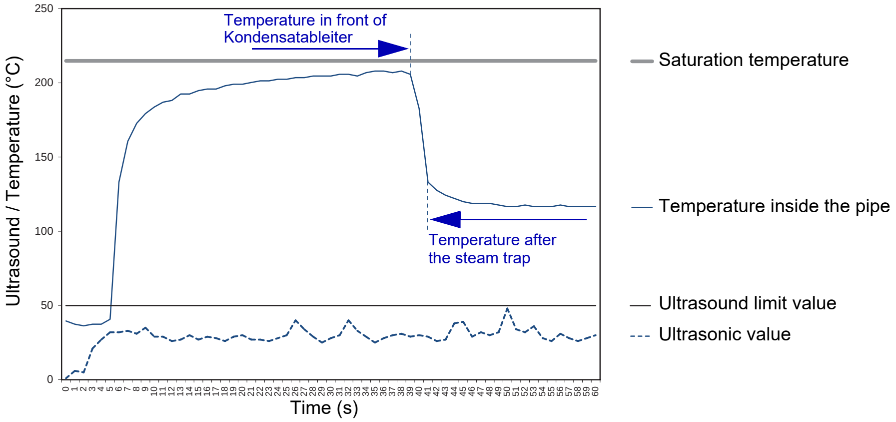


Fig. 8-5: Printout for an ultrasonic test of a thermic (bimetallic-) steam trap without thermodynamic amplification (continousliy operation)

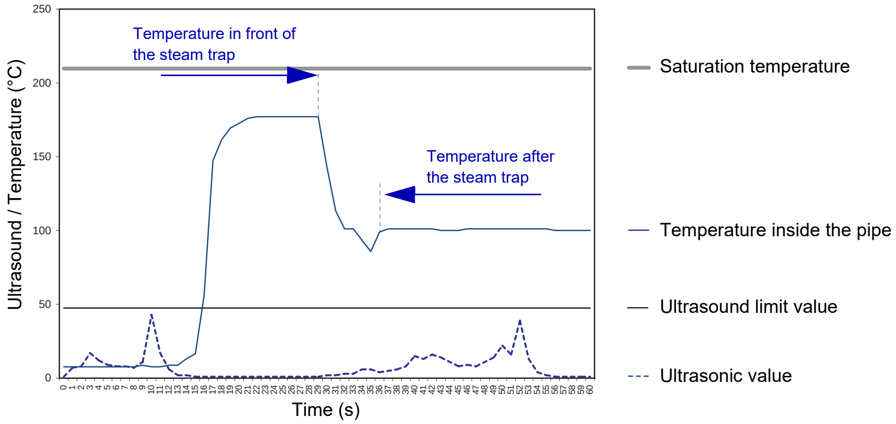


Fig. 8-6: Printout for an ultrasonic test of a thermic (bimetallic-) steam trap with thermodynamic amplification (intermittent operation)

Assessment of the measured values for inverted bucket traps

The cycle of an inverted bucket steam trap is very simple to reconstruct. The condensate is discharged in bursts and the measured value rises to 100 %. If steam is present in the trap, the ultrasonic value is quickly reduced to zero. If a constant, high signal is measured, the trap is passing live steam.

Assessment of the measured values for thermodynamic traps

The working cycle of thermodynamic traps can be reconstructed equally easily. A full-scale deflection is indicated when the trap opens. The sound value reaches its maximum. At the moment of closing, this value returns almost to zero. If a constant, high value is measured, the trap is passing steam. More than five cycles (opening / closing) a minute are a sign that the valve disc and seat are no longer able to shut sufficiently tightly and the trap is approaching the end of its life.

Conductivity measurement

The conductivity is measured in a test chamber containing a water seal that is located upstream of the trap.

The condensate continues to flow under the separating rib. A small hole for pressure compensation is provided in the top of the rib, so that a constant condensate level exists in both chambers and the test chamber is prevented from running dry. If the trap is operating correctly, the water seal remains intact. If it is passing steam, the latter forces out the seal completely.

A conductivity electrode is mounted in the test chamber and a measuring instrument connected to the electrode contacts. If the test chamber is filled with condensate, the conductivity is detected by the electrode and a green signal lamp lights up. If the test chamber has been completely blown out due to the leakage of live steam, the electrode will no longer be surrounded by condensate and the circuit is interrupted. The live steam loss is indicated by a red signal lamp.

In addition to manual conductivity test devices, systems for monitoring the operation of several traps on a central instrument board are also available.

The conductivity electrode can optionally be installed directly in the steam trap (Fig. 8-7). In this case, there is no need for a test chamber upstream of the trap and the measurement is much more straightforward.

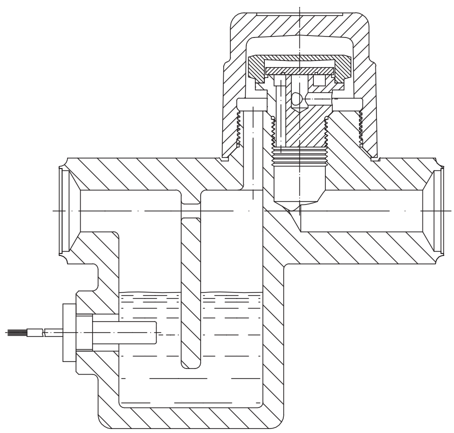


Fig. 8-7: Conductivity measurement in a steam trap

The conductivity measurement method is suitable for monitoring both continuously and intermittently operating steam traps. However, account must also be taken of the following factors and side-effects:

- The conductivity of high-purity condensate is so low that live steam loss is indicated (e.g. if fully demineralised water is used as feedwater for the steam boiler).
- Normal condensate contains impurities, and deposits on the electrodes convey false information.
- The conductivity measurement is meaningless if amines that form a protective layer are used to eliminate corrosion, because in this case protective layers will also build up on the electrodes.
- Condensate back-up is only indicated if a temperature sensor is installed in the test chamber.
- If the functional test upstream of the traps is carried out manually, there is a risk that the outlet will become dirty in the course of time, especially if the process frequently involves solids.
- The closing torque of some trap types is so near to the steam limit that the water seal may be completely forced out, even though there is no live steam.

Calorimetric testing

The calorimetric test method is based on the physical temperatures of gas (air) and steam that trigger a discharge function in the trap (Fig. 8-8).

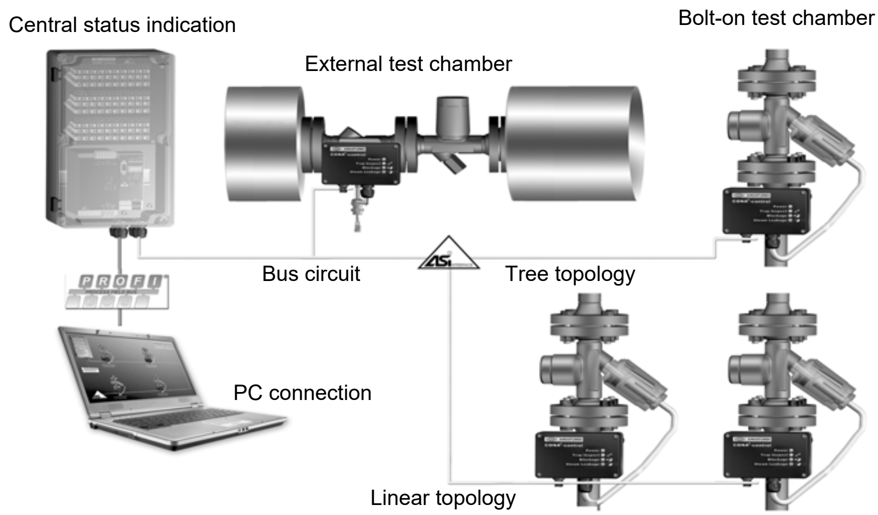


Fig. 8-8: Calorimetric test method, ARI Type CONA®control

The chamber is installed in front of the steam trap being monitored. The chamber has two temperature sensors, one of which is continually heated. The information provided by the sensors is processed and output to LED indicators ("Power“, "Inspect Trap“, "Blockage“ and "Steam leakage“) The chamber remains full of condensate (Fig. 8-9).

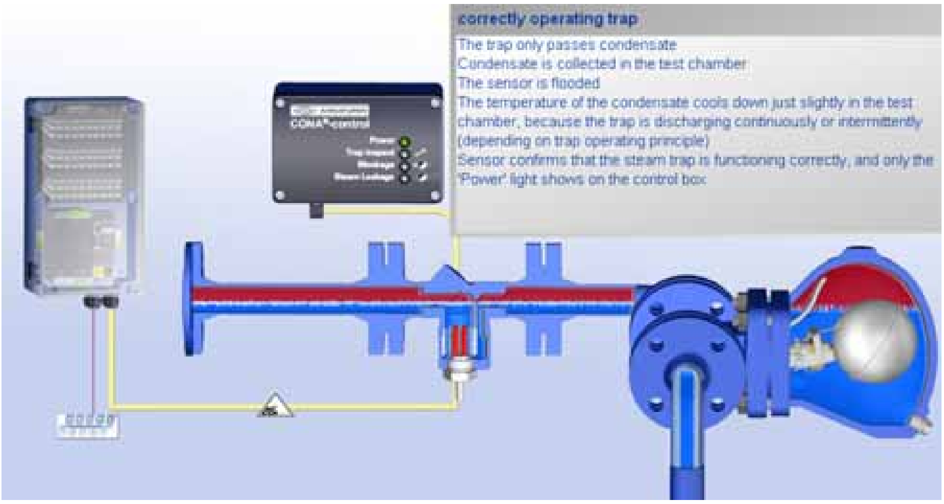


Fig. 8-9: Correctly operating trap, ARI Type CONA®control

The sensor is flooded. The condensate in the test chamber cools down slightly. The heated temperature sensor transfers part of its heat to the condensate. A defined temperature difference between the two sensors is not exceeded. Only the "Power" light is lit and the steam trap is functioning correctly. If the trap does not shut steam-tight, the sensor will be surrounded by steam (Fig. 8-10).

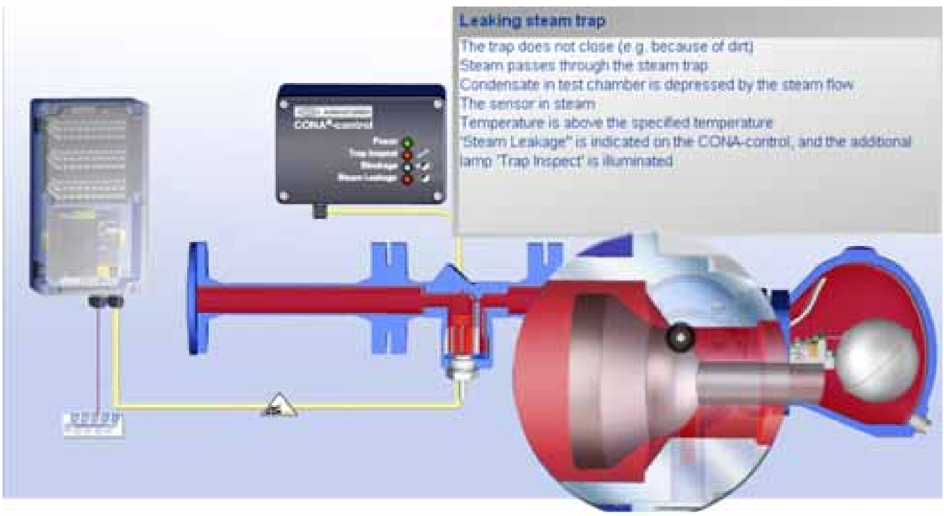


Fig. 8-10: Leaking trap, ARI Type CONA®control

The temperature gradient between the heated and unheated sensors changes and the setpoint specified for the temperature difference is exceeded. This indicates that the trap is passing steam, and the "Steam Leakage" LED is lit.

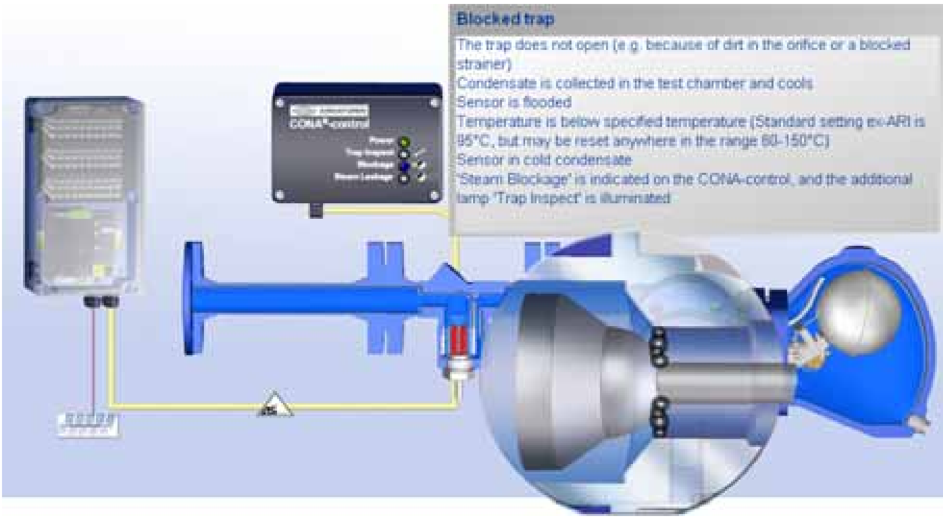


Fig. 8-11: Blocked trap, ARI Type CONA®control

If condensate back-up occurs upstream of the trap, the test chamber is flooded (Fig. 8-11).

The condensate in the test chamber cools down. If no condensate is discharged because a trap is blocked, the temperature falls below the set limit value (default: 95 °C) and the sensor reports cold condensate. The "Blockage" LED lights up.

When the system is shut down, the steam pressure in the system is reduced and the hot condensate in the test chamber and the pipe vaporises. The sensor is surrounded by steam as a result and the temperature falls below the set limit value. It reports cold condensate because the minimum limit was exceeded as well as steam leakage because both probes are surrounded by steam / gas and the temperature difference is too large. Both the "Blockage" and the "Steam Leakage" lamps light up.

The calorimetric system described here with external or bolt-on test chambers for steam traps is suitable for indicating states centrally. Deviations are readily visible, a defective trap can be identified easily thanks to bus technology and costly steam losses are eliminated without any wasted time. The calorimetric sensor detects not only steam leakage but also blocked traps. The system is independent of condensate conductivity as well as deposits on the sensors that are liable to distort the measurements (e.g. protective amine layers or magnetite).

Summary of test methods

Visual inspection
Suitable as a method for determining condensate back-up.
Can also be used with restrictions to establish live steam loss in thermostatic traps.

Noise measurement
Efficient way to monitor for live steam loss from intermittently operating traps.
Considerable experience is necessary to interpret and assess the operation of float-type steam traps.

Conductivity measurement
Only recommended for detecting live steam leakage under ideal conditions. Sensitive to magnetite deposits, protective amine layers and other boiler chemicals. Does not work with high-purity condensate. Only suitable for determining condensate back-up in combination with a temperature sensor.

Calorimetric testing
Suitable for all trap types without the drawbacks of other methods. The calorimetric sensor detects not only steam leakage but also blocked traps. The system measures the functional state at a given instant independently of condensate conductivity or deposits and may additionally communicate with sophisticated computer monitoring and management systems.

8.5 Determination of leakage losses

The amount of live steam that is lost due to leaking traps tends to be underestimated. The arguments put forward for investments in a permanent steam trap monitoring system are not always taken seriously. After drawing up a loss calculation, however, the figures speak for themselves and there is no doubt that money spent on system of this kind or on regular inspections is justified. Assuming a steam price of €22.- /tonne and a leakage rate of 10 kg/h, a total of €2,000.- is wasted annually, resulting in unnecessary emissions of 13000 kg of CO_2.

***Example from industry*:**

Approximately 85 steam traps, with 8500 hours in service, are installed on the site of a chemical plant. Some of them are float-type steam traps but the majority are bimetallic. Their operation used only to be monitored sporadically – if heat transfer problems occurred in the production process, the traps concerned were simply replaced without any attempt to identify the cause. The steam cloud that appeared on the condensate vessel was considered to be part of the "flash steam loss".

A decision was then taken to catalogue all traps and monitor their operation regularly by means of ultrasonic tests:

- *Condensate back-up was clearly established in eight traps, which were subsequently replaced.*
- *Steam leakage was discovered in 16 traps. A constant, high ultrasonic signal was recorded for some of these traps, while in other cases the signal failed to return to the initial value. The next step was to gauge the losses from four traps with a high signal and four with a signal that did not return to zero.*

It was agreed with the management of the plant that the highest (revealed later to be 60 kg/h) and lowest (revealed as 4 kg/h) measured values should be ignored for the purpose of the subsequent calculations. The average loss from the six remaining traps was then determined. This value was taken as representative for all 16 traps. The average loss per trap was 14 kg/h, making a total of 1900 tonnes of steam per year for the 16 traps together. Based on the present level of energy prices, this represents an annual loss of €42,000.- .

After being presented with the results of the survey, the management approved a budget for a permanent steam trap monitoring system with centralised evaluation. A handheld test unit was purchased to carry out temperature and ultrasonic tests on those traps that are too remote to be integrated in the central system. Special training in steam trap technology was provided to fitters and operators.

The traps that were found to be leaking steam were replaced within a week. The steam cloud on the condensate vessel is now visibly smaller. The reduction in the amount of make-up water matched the measured leakage quantity. Although gauging the leakage losses precisely upfront of this project was an arduous and time-consuming task, the outcome was extremely rewarding.

Gauging calculation (example):

Steam pressure in the heat exchanger	10 bar (saturation temperature = 180 °C)
Enthalpy of 10 bar saturated steam	h_g = 2776 kJ
Amount of condensate during the measurement	5 kg
Amount of water at the start of the measurement	20 litres at 10 °C
Measuring time	3 minutes
Condensate / water mixture temperature	62 °C
Heat content at the end of the measurement	25 x 62 x 4,2 = 6510 kJ
Heat content at the start of the measurement	20 x 10 x 4.2 = 840 kJ
Increase in heat content	5670 kJ

Without live steam loss the mixture temperature should be:
{(20 x 10) + (5 x 180)}/25 = 44 °C with a heat content of 25 x 44 x 4.2 = 4620 kJ

The temperature rise due to live steam leakage is thus 5670 - 4620 = 1050 kJ.
1050/2776 = 0.38 kg of live steam is entrained with the condensate during the three minutes of the measurement, equivalent to 7.6 kg/h. This represents an annual loss of 65 tonnes of steam, which based on an energy price of €7.50 /GJ works out at a total financial loss of €1,355.- . Conclusion: The investment in new steam traps is recuperated in only a few months!

9.0 Trace Heaters

9.1 Introduction

Pipes, which are operated at ambient temperatures, lose heat, regardless of how well the pipe is insulated. Trace heaters are used to raise or maintain the temperature of a product in the pipe, to protect it from frost, or to compensate for heat loss.

It is possible to distinguish between two types of trace heaters – Electrical or steam-powered. Trace heaters that use steam have an advantage over electrical heaters in that they are more cost effective to operate and can be used in hazardous areas without any problem. This chapter only deals with steam-powered trace heaters.

The design of a steam trace heater depends on the system so – depending on the heat loss – one or several smaller steam pipes (tracers) are laid along the product pipe and around, valves and fittings. The way in which the heater is fitted generally has a considerable influence on the heat output of a trace heater. The average thermal outputs are given in W/m K for some technical designs. This book does not deal with the details of how heat loss is calculated and how to specify the number of tracers. The calculation method is only explained using an example. For day-to-day practical purposes the number of tracer heating pipes is calculated on the basis of values established from past experience and tables for some operating situations checked in practical situations. This chapter also discusses the technical design of trace heaters and the selection of traps for some applications.

9.2 Trace heater classes

The required steam pressure and the number of tracers required to provide frost protection or for product trace heating are determined in different ways. When it comes to product trace heaters a distinction is made between light, medium-duty and heavy-duty trace heaters. Space heating is often advisable for certain purposes. The characteristic of a space heating system is that there is no direct contact between the tracer and the pipe. Heat is primarily transferred by radiating heat to the still air in the space between the pipe and the tracer. The air in turn transfers heat to the pipe by means of convection.

Trace heater classes:

- Frost protection:
  To prevent pipes from freezing, holding temperature 10 ºC.
- Space heating:
  Pipes with sensitive media (e.g. measuring lines), holding temperature 10 ºC.
- Light product trace heater:
  Raise and maintain temperature/below 30 ºC.
- Medium-duty product trace heater:
  Raise and maintain temperature/between 30 ºC and 80 ºC.
- Heavy-duty product trace heater:
  Raise and maintain temperature/above 80 ºC.

Copper, steel and stainless steel are materials of a suitable quality for tracers. The diameters generally used are 12 mm for copper and 15 mm for steel. The holding temperature is so critical on some products that conventional trace heaters cannot fulfil these criteria adequately. Heat transfer paste is used in such cases. In contrast to normal trace heaters, the tracer here is fitted on the top of the product pipe and the contact surface with the pipe wall is increased several times through the use of heat transfer paste. There are process situations where a steam jacket heating system around the process pipe is the best solution (jacket heating or jacketing).

9.3 Heat requirement of a pipe

The calculation of how much heat is lost by a pipe is based on the following formula:

$$\dot{Q}_V = k_1 \times l \times \Delta T \qquad [1]$$

$$\frac{\dot{Q}_V}{l} = k_1 \times \Delta T \qquad [2]$$

$$k_l = \frac{\pi}{\frac{1}{\alpha_j \times d_1} + \sum_j \frac{1}{2 \times \lambda_j} \times \ln\frac{d_{j+1}}{d_j} + \frac{1}{\alpha_a \times d_4}} \qquad [3]$$

$\dot{Q}_V$ = Heat loss in W

$\frac{\dot{Q}_V}{l}$ = Heat loss per meter in W/m

k_l = Heat transmission coefficient per running meter in W/m K

ΔT = Temperature difference between the product and the ambient air in Kelvin

l = Pipe length in m

[1], [2], [3] WAGNER, W.: Wasser und Wasserdampf im Anlagenbau. Vogel Buchverlag, Wurzburg, 2. Edition, 2011

The heat transmission coefficient is dependent on the coefficient of heat transfer (α_i) between the product and the pipe wall, the thermal conductivity (λ_1, λ_2, λ_3) of the pipe wall, insulation and of the plate and the heat transfer coefficient (α_a) between the insulation plate and the ambient air – see Fig. 9-1.

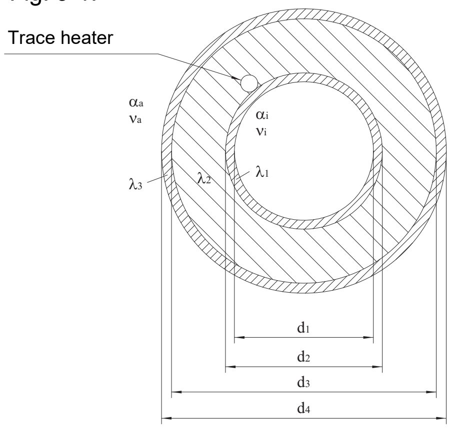


Fig. 9-1: Heat transfer factors

Sample calculation:

Where the lowest outside temperature T_2 is -15 °C, the 40 m long oil pipe DN 250 must be kept at a temperature T_1 of 50 °C. ΔT = 65 K

Heat transfer coefficient, oil to pipe wall α_i = 500 W/m^2 K
Heat transfer coefficient, insulation plate to air α_a = 10 W/m^2 K

Pipe DN 50

Inner diameter d_1 = 260.4 mm
Outside diameter d_2 = 273 mm
Thermal conductivity λ_1 = 60 W/m K

Insulation

Insulation thickness s_1 = 100 mm
Thermal conductivity λ_2 = 0.06 W/m K

Aluminum insulation plate

Wall thickness s_2 = 1.2 mm
Thermal conductivity λ_3 = 200 W/m K

Calculation objective:

Heat loss $[\dot{Q}_V \text{ in } W]$ *of the pipe*

Calculation:

$$k_l = \frac{\pi}{\frac{1}{500 \times 0.2604} + \frac{1}{2 \times 60} \times \ln\frac{0.273}{0.2604} + \frac{1}{2 \times 0.06} \times \ln\frac{0.473}{0.273} + \frac{1}{2 \times 200} \times \ln\frac{0.4754}{0.473} + \frac{1}{10 \times 0.4754}}$$

$$k_l = 0.6547 \text{ W/m K}$$

$$\dot{Q}_V = 0.6547 \times 40 \times (50 - (-15)) = 1702.2 \text{ W}$$

$$\frac{\dot{Q}_V}{l} = 0.6547 \times (50 - (-15)) = 42.56 \text{ W/m}$$

In order to keep the oil pipe at 50 °C when the outside temperature is -15 °C, compensation of 1702.2 W (42.56 W per running meter) is required from the trace heater tracer pipe.

9.4 Heat output of a trace heater

In order to prevent unacceptable cooling of a product pipe, the same amount of heat must be supplied by the trace heater as is lost through the insulation. It must be remembered that only part of the tracer surface loses heat through radiation but, in the space between the tracer and the pipe, heat is transferred through convection. In addition to this, part of the tracer surface is covered by insulation and cannot contribute effectively to heat transfer. What the above information means is that the heat output factors depend on how the tracer is fitted.

- ***Space heating***, Fig. 9-2: With this form of heating the tracer does not make contact with the product pipe and heat output is by means of convection. Average heat output is 0.35 W/m K.

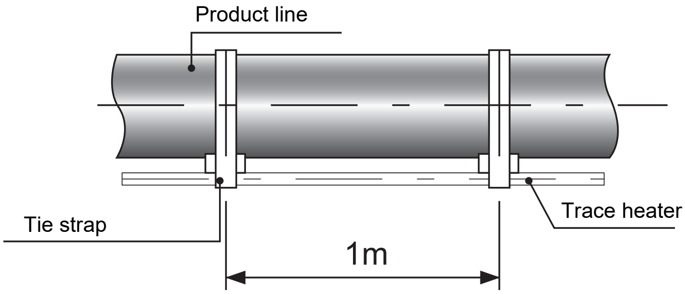


Fig. 9-2: Space heating

- ***Partial space heating***, Fig. 9-3: The tracer is in relatively close contact to the product pipe. Heat output is only partially via the product pipe. Heating has an average heat output of 0.7 W/m K.

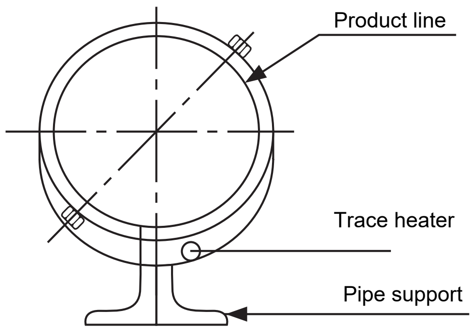


Fig. 9-3: Partial contact heating

- ***Contact heating***, Fig. 9-4: Almost complete contact between the tracer and the product pipe is achieved through the use of clamping bands. The maximum distance between bands is 600 mm. Heat output is almost exclusively via the product pipe. Average heat output is 1.1 W/m K.

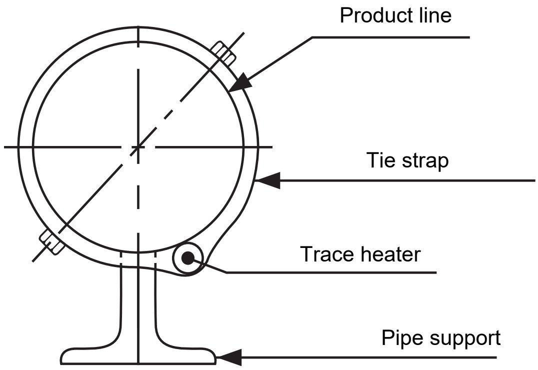


Fig. 9-4: Contact heating

- ***Heating via heat transfer paste***, Fig. 9-5, for the heavy trace heating. In order to increase the transfer of heat between the tracer and the product pipe, in addition to the clamping bands the heat conduction area is increased through the application of heat transfer paste. The average heat output can increase up to 5 W/m K.

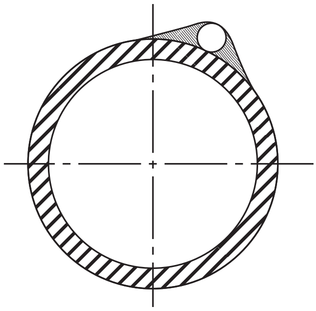


Fig. 9-5: Tracer with heat transfer paste

9.5 Calculating the number of tracers

As has already been mentioned, heat losses will not be calculated in detail here. The scope of this chapter should be sufficient to calculate the heat losses on the basis of values established from practical experience and using tables, Fig. 9-6 to Fig. 9-9. The tables are adjusted to steam pressure of 3.5 - 5.5 - 8.0 - 10 bar and for DN 15 steel tracers and DN 12 copper tracers. It is generally true that the saturated steam temperature must be at least 30 Kelvin higher than the required holding temperature.

Example: To keep the product temperature at 140 °C, steam pressure of 8 bar (170 °C) should be used for heating.

Tables

The left-hand column of the table indicates the holding temperature in ºC. The right-hand column “L” shows the maximum length of the tracer and “H” shows the maximum height difference, i.e. the number of metres which the tracer is allowed to rise (static head).

One tracer is sufficient for the areas in the table with a light-blue background (1). Two tracers are required for the areas (2) with a dark-blue background and three tracers are needed for the grey area (3).

Holding temperature °C	Number of tracers with steam pressure of 3.5 bar (138 °C)											
	Pipe diameter DN											
	25	40	50	80	100	150	200	250	300	350	L	H
10												
20											50	4
30												
40												
50				*1*								
60												
70						*2*					40	3
80												
90									*3*			
100												

Fig. 9-6: Approximated calculation of the number of tracers with steam pressure of 3.5 bar

Holding temperature °C	Number of tracers with steam pressure of 5.5 bar (155 °C)											
	Pipe diameter DN											
	25	40	50	80	100	150	200	250	300	350	L	H
10												
20											60	20
30												
40												
50				*1*								
60											60	12
70												
80								*2*				
90												
100											50	12
110										*3*		
120												

Fig. 9-7: Approximated calculation of the number of tracers with steam pressure of 5.5 bar

Holding temperature °C	Number of tracers with steam pressure of 8 bar (170 °C)											
	Pipe diameter DN											
	25	40	50	80	100	150	200	250	300	350	L	H
10												
20											70	30
30												
40												
50				*1*								
60												
70												
80											80	
90												
100												20
110							*2*					
120												
130												
140										*3*	50	
150												

Fig. 9-8: Approximated calculation of the number of tracers with steam pressure of 8 bar

Holding temperature °C	Number of tracers with steam pressure of 10 bar (180 °C)											
	Pipe diameter DN											
	25	40	50	80	100	150	200	250	300	350	L	H
10												
20											70	30
30												
40												
50				*1*								
60												
70											60	20
80												
90												
100												
110							*2*					
120												
130												
140										*3*	50	20
150												

Fig. 9-9: Approximated calculation of the number of tracers with steam pressure of 10 bar

The tables can be applied if insulation is of the thickness shown in the chart (Fig. 9-10) and the minimum outside temperature is -15 ºC. Some care must be taken as to how the tracer heater makes contact with the product pipe. The distance between clamping bands must not exceed 600 mm here. The tables are not suitable for determining how much time is required before a solidified or very cold product is pumpable again.

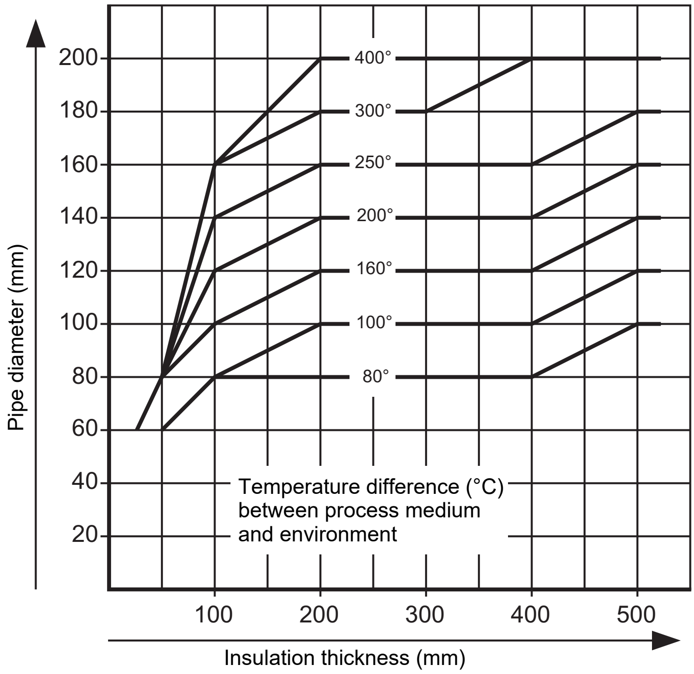


Fig. 9-10: Insulation thickness for tracer heater

Example for Fig. 9-6:
- Product pipe DN 100
- Holding temperature 40 °C
A tracer is required. Two tracers would be required for a holding temperature of 60 °C. The maximum length of the tracer is 50 m and the height difference to be bridged must not be more than 4 m.

Example for Fig. 9-7:
- Product pipe DN 100
- Holding temperature 60 °C
Only one tracer is required for a steam pressure of 5.5 bar. The maximum tracer length is 60 m and the height difference must not be more than 12 m.

Example for Fig. 9-8:
- Product pipe DN 200
- Holding temperature 80 °C
One tracer is sufficient at a steam pressure of 10 bar. In this case, consideration must be given to using two tracers, depending on the operating circumstances.

9.6 Methods of trace heating

Space heating

Space heating is used in systems where the product carried in the heated pipe must not get too hot. These include pipes filled with water, e.g. impulse lines for instruments, emergency showers or potable water pipes which must be kept frost free but without steam forming in the pipes. Plastic pipes used with trace heaters can become deformed if they come into direct contact with the steam tracers. The simplest way is to heat these pipes using an over-sized insulating shell, where the tracer will easily fit in the space between the shell and the pipe, Fig. 9-11. The insulating shell should be fixed to the product pipe with blocks.

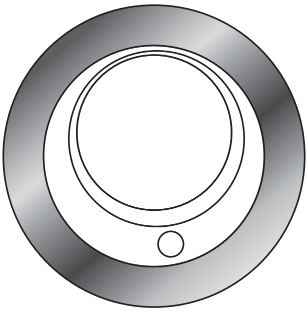


Fig. 9-11: Space heater with over-sized insulating shell

Designs using clamping bands are often seen. Here metal and plastic spacers or glass-fibre mats are used between the tracer and the product pipe in order to prevent the product pipe coming into direct contact with the tracer, Fig. 9-12.

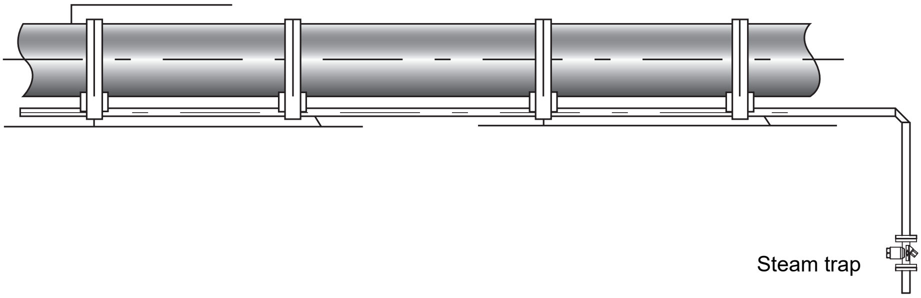


Fig. 9-12: Space heater with spacer

Heat transfer paste

Heat transfer paste is a thick paste with a high graphite content. The graphite makes the paste a very good conductor of heat. The paste is applied between the tracer and the product pipe, thus allowing the contact surface to be increased considerably. Clamping bands are used to clamp the tracer to the product pipe. Where heat transfer paste is used for the trace heater, nearly all heat is transferred by heat conduction. The tracers do not have to be fitted on the lower side of the product pipe. Installation is much easier if the tracer with the heat transfer paste is fitted on the top of the product pipe. The paste is applied in the gap between the product pipe and the tracer using a putty cartridge or a trowel. If a trowel is used, the paste is applied in a roof shape and fastened with clamping bands, Fig. 9-13.

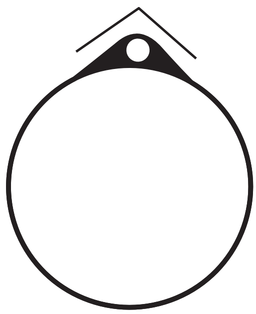


Fig. 9-13: Tracer with heat transfer paste in roof shape

Before the paste is applied, the trace heating should be checked to make sure it is free of leaks. Once the paste has been applied, leave the compound to cure slowly by supplying heat. If the compound is heated up too quickly, bubbles are formed in the paste and there is a risk that air may be trapped. In any case, the manufacturer's instructions must be followed down to the last detail.

Pre-formed flexible sections are also available and can be used in place of the heat transfer paste. Such a section is pushed over the tracer, covered with a channel section and tightened with clamping bands. In general, if heat transfer paste is used, the number of tracers can be reduced. It is possible that one tracer may be possible then. Heat transfer paste suppliers will provide detailed advice. In general, it is advisable to use more than one tracer for pipe diameters in excess of DN 250.

Half-pipe heating

A half-pipe heating arrangement is sometimes used for critical heating applications in order to improve heat transfer through the product pipe. In this case the tracer is welded on to the half-pipes, Fig. 9-14. Heat conductivity between the tracer and the half pipe is therefore maximised and the heat transfer area between the half pipe and the product pipe has been increased considerably. The half pipes are tightly bound to the product pipe with clamping bands. Heat transfer paste can also be applied to optimise heat transfer between the half pipe and the product pipe.

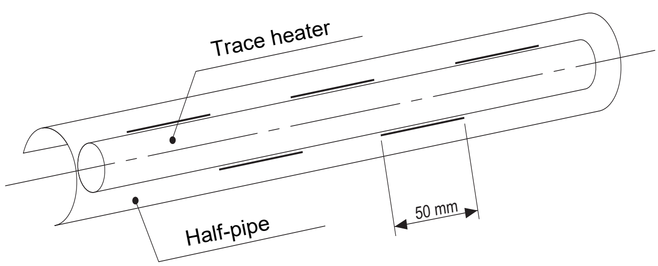


Fig. 9-14: Half-pipe heating

Jacket heating

Jacket heating (jacketing) is also an appropriate solution for heating situations where heat transfer is critical. This may, for instance, be the case if the saturation temperature of the available steam is less than 30 ºC greater than the holding temperature of the product. The critical 30 Kelvin limit can be moved a little because a steam jacket heating system is fitted around the product pipe. Steam is always supplied to the upper side of the jacket and the condensate is drained off at the bottom. If the product pipe is divided into several part sections using flanges, the jacket sections should be connected to each other at the steam and condensate sides, Fig. 9-15.

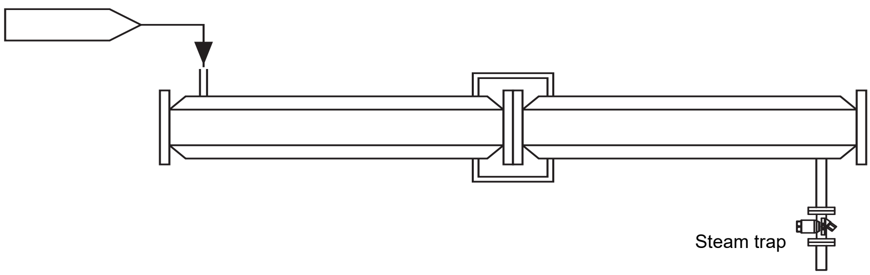


Fig. 9-15: Jacket heating

It is fairly common to connect the steam sections in series, but for each part section has its own condensate drain. As is the case with other heating methods, flanges, valves and fittings are the most sensitive parts because a great deal of heat can be lost, e.g. via thermal bridges. For this reason, there are drilled pairs of flanges, which connect steam and condensate from one section to another using couplings, thus maintaining the flange temperature.

9.7 Details about trace heaters

Fitting a tracer

As heat is mainly transferred by means of convection, the tracer must be secured below the product pipe, with the exception of tracers where heat transfer paste is used. It is advisable to secure the tracer at an angle of 15 degrees from the vertical. By doing this, it prevents any dirt lying on the base of the pipe from sticking inside the pipe, Fig. 9-16.

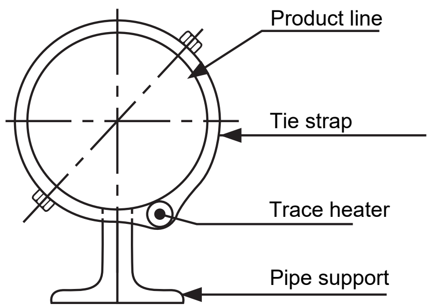


Fig. 9-16: Tracer 15 degrees from the vertical

If several trace heaters are required, the following method of installation should be adopted (Fig. 9-17).

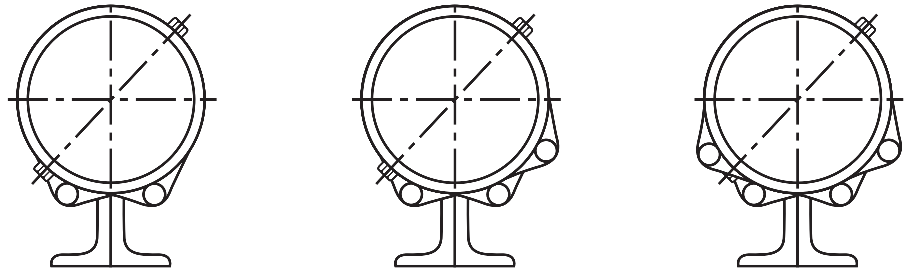


Fig. 9-17: Method of installation for several trace heaters

The position of the tracers is not as significant for tracers used in conjunction with heat transfer paste as for tracers where heat is transferred by means of conduction. The position is often adapted to the design of the system to take fitting considerations into account, Fig. 9-18.

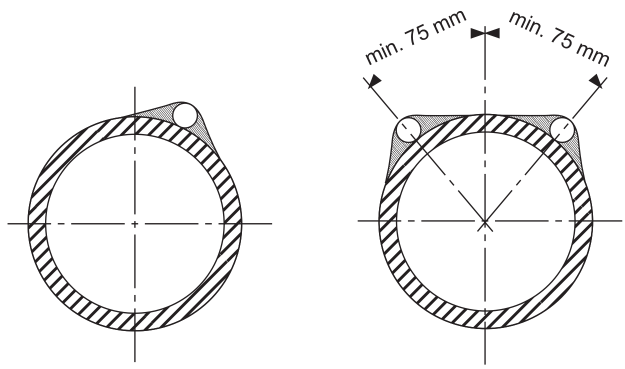


Fig. 9-18: Distribution when heat transfer paste is used

Height difference

Trace heaters are always laid starting at the bottom and working upwards. Rising sections are to be avoided, if possible. The permissible height different is also limited. The maximum permissible height difference is given in tables Fig. 9-6 to Fig. 9-9. Where there is a rising pipe bend in a trace heater, the condensate produced collects before the pipe rises. Because the speed of flow in the tracer is very low, no or hardly any condensate is transported upwards. This condensate is not discharged until the entire rising section is full of condensation.

The lengths of all rising sections must be added up to calculate the total resistance. Each metre a section rises increases the resistance by 0.1 bar. Back pressure in the condensate system must also be included, Fig. 9-19.

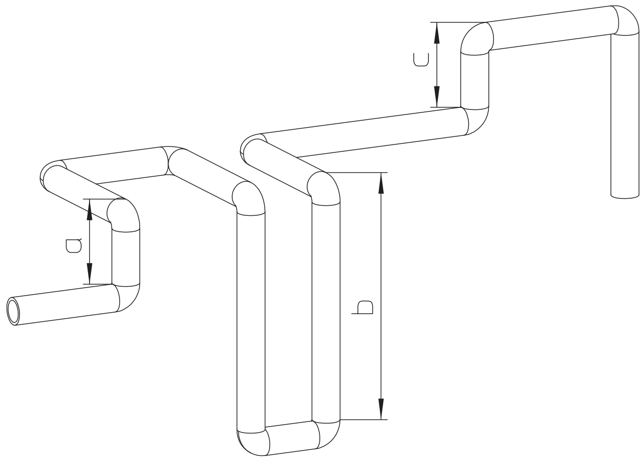


Fig. 9-19: Addition of rising pipe sections in a trace heater

Sometimes a vessel or a pipe is coiled around the tracer. Each of these coils must also be added to calculate the difference in height: 20 coils around a DN 250 pipe produce a height difference of 0.5 m.

Expansion loops in the trace heater

The tracer must be laid with expansion loops to balance out the expansion of a long trace heater section, in particular during start-up. Expansion of the tracer depends on the temperature and material and is calculated using the following formula:

$$\Delta l = l_0 \text{ x } a \text{ x } \Delta t$$

Δl = Expansion in mm
l_0 = Total length when cold in m
a = Coefficient of expansion mm/m K
Δt = Temperature difference between hot and cold in degrees K.

In practice, the temperature when hot is equivalent to the steam saturation temperature. The following rule of thumb applies: Expansion approx. 2 mm per m of tracer length. When flanges are fitted in the pipe at regular intervals, the expansion bends are fitted on a horizontal plane around these flanges, Fig. 9-20.

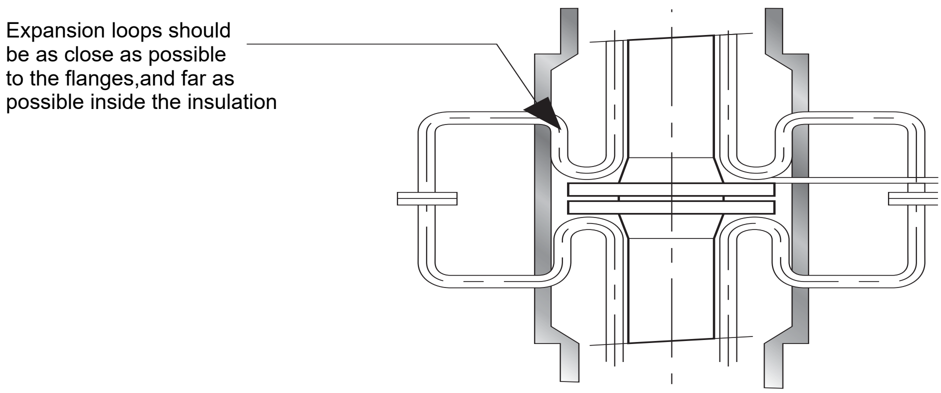


Fig. 9-20: Expansion loops around a flange

It is advisable to fit a coupling or a flange in the loop where pipe sections are regularly dismantled. Expansion loops should be fitted at intervals of 20 m in pipes without flanges. The span of each loop should be 300 mm, Fig. 9-21.

If there are several tracers on the same side of the pipe, the loops can be fitted inside each other: the loop on the outside will then be somewhat larger.

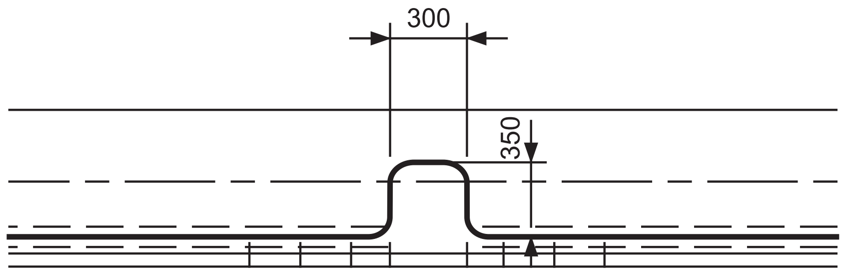


Fig. 9-21: Span of expansion loops

Heating up a control or globe valve

If a valve needs to be heated up, care should be taken to ensure that the valve can be maintained and removed. It is not particularly easy to equip a valve body with a 15 mm tracer so that a heater can still be said to be effective. If the function of the valve is critical to the process, for example because there is a risk that a product may solidify, the valve is often heated per 8 mm branch from the main tracer. The condensate is then drained via a separate trap, Fig. 9-22.

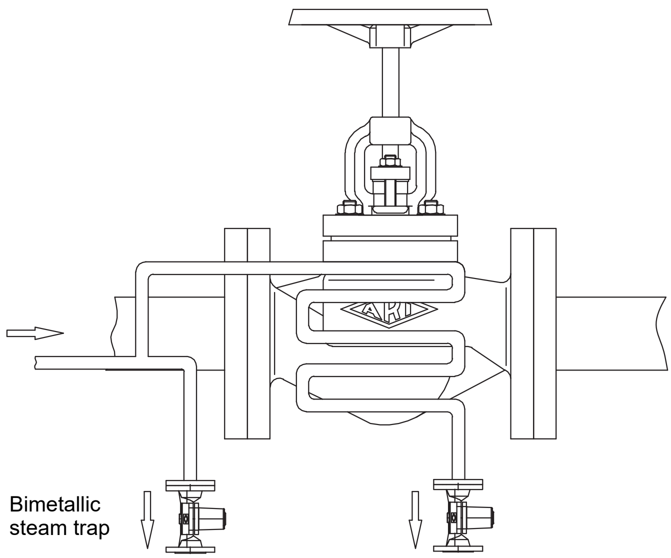


Fig. 9-22: Heating a valve using a loop from the main tracer

With flanged installations, it may be difficult to route a 15 mm pipe for optimum performance. An 8 mm off-take from the main pipe may provide an alternative solution, Fig. 9-23.

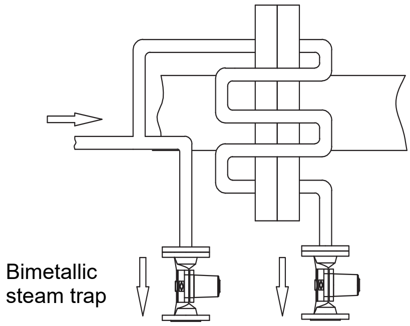


Fig. 9-23: Heating a flange with a branch from the main tracer

Heating a level controller

A level controller, see Fig. 9-24, can be heated via a coil. Because it is a vertically fitted coil here, the information on flow resistance does not apply in this instance.

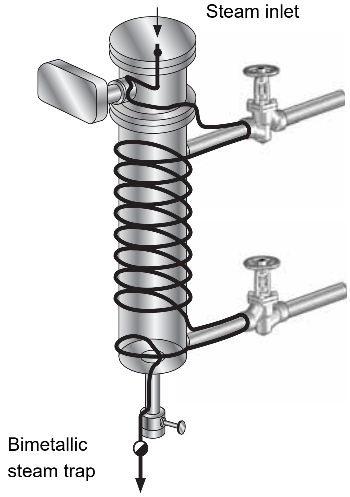


Fig. 9-24: Coil heating of a level controller

For media with a low boiling point, e.g. Water, measures should be taken to prevent boiling of the fluid in the main pipe. The heating coil should be carefully positioned a measured distance from the pipe; alternatively, insulation pads can be placed between the coil and the pipe. This is also true for level control, where the heating of the measuring column runs in the vertical direction, Fig. 9-25.

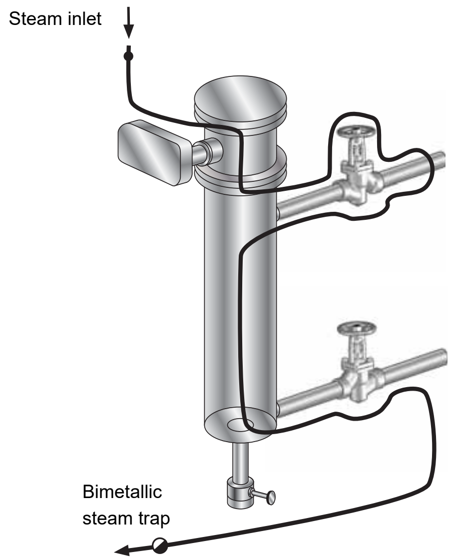


Fig. 9-25: Level controller heating in a longitudinal direction

Steam header and condensate collection stations

For processes which require a high level of trace heating, it is advisable to run the tracer connections via local header and collection stations, Fig. 9-26 and Fig. 9-27.

Here a certain system should be applied which involves assigning the same position and numbering to the tracer's steam globe valve as to the corresponding condensate globe valve on the collector. This provides a clear overview and avoids unnecessary searching if, for instance, the tracer has been taken out of service for repair work on apparatus or pipes.

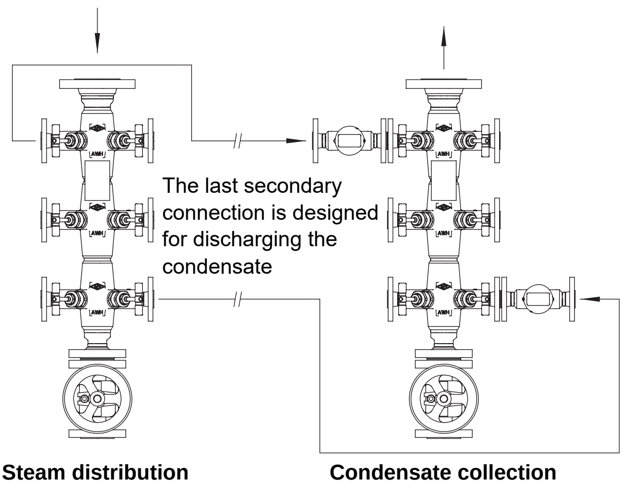


Fig. 9-26: ARI Type CODI®

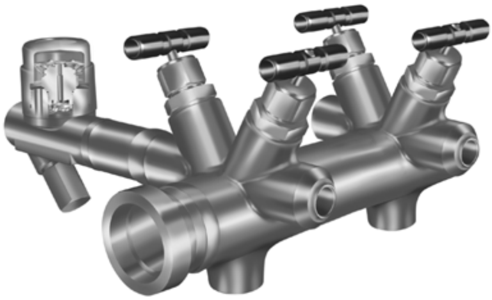


Fig. 9-27: Steam header station, ARI Type CODI®S

The condensate is drained from the condensate header via an internal pipe. This ensures that the header always stays warm, even if only one trace heater is in operation.

Steam trap per trace heater

At first glance, may appear to be an attractive solution to fit only one steam trap for both condensate drains, for instance when a pipe has two trace heaters. However, there is a risk here that the condensate will not be drained adequately, even if steam is supplied separately via two valves. The safety risk is that the flow resistance is slightly different, with the result that the pressure upstream of the trap is higher for the trace heater with the lowest resistance than for the trap with the highest resistance. As a result of this, the flow through the trace heater with the highest resistance (head loss to friction), is considerably impeded or even blocked.
Fig. 9-28 is to be seen as a warning notice!

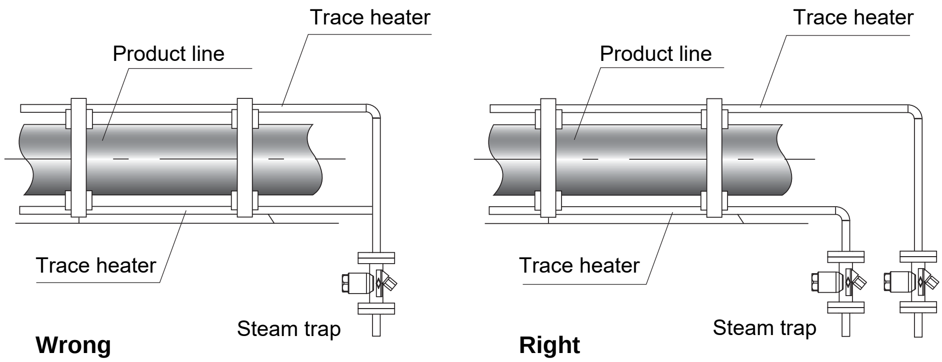


Fig. 9-28: Each trace heater has its own steam trap

Example from industry:
A chemical container has a 80 m long venting pipe to the air washer. The pipe has two copper trace heaters with one shared steam trap. One of the two tracers had a slight kink, which resulted in the through flow stagnating and in winter the condensate froze in the trace heater. The pipe burst and the steam escaped into the outside air. The pressure in the trace heater still operating then fell to such a low level that it no longer exceeded the back pressure in the condensate system. The second tracer froze like the venting pipe did. A vacuum developed in the vessel and it imploded.

In places where several tracers are bundled and routed to a collection station, stresses occur in the usually short connecting pipes, which can in turn lead to leaks. It is recommended to install a small expansion bend between the trap and the tracer. A 12 mm annealed copper pipe is generally used for this. Fitting this expansion bend prevents the trace heater being exposed to too much stress, thus preventing a leak. The bend can also be fitted upstream of the trap. A thin armoured hose can also be used instead of a copper expansion bend. These expansion bends are often also used for steam connections.

9.8 Putting a trace heater into service

One of the most important procedures before putting a trace heater system into service is to blow through the tracers with steam to ensure that these are free of dirt. Once the main steam pipes and the supply pipes have been blown through, the same procedure is carried out with the trace heater. The steam trap should be disconnected before this is done. The tracers are blown through, one after the other, with the valves fully open. A check can be made to ensure this cleaning operation has been effective by placing a metal plate in front of the outlet orifice. The sound of the particles of dirt making contact can be clearly heard. Once the steam trap has been refitted, the tracer heaters should be checked to ensure that they are air tight before the insulation is attached. Also, the couplings and flange connections should be re-tightened once they are warm.

9.9 Selecting a steam trap

In principle, all types of trap are suitable for discharging condensate in tracer heaters. A type can be selected depending on factors such as steam pressure, holding temperature, the possibility of recovering condensate and the fitting dimensions. The factor of the fitting dimensions alone shows that compact and lightweight steam traps are preferred in tracer heaters. Inverted bucket or float traps allow considerably more extensive condensate collection stations. In contrast to this, thermodynamic traps or balanced pressure steam traps have a compact design. It is quite important to ascertain whether the flash steam that has formed downstream of the trap can still be used or if the holding temperature of the product is so low that a trap adjusted to sub-cooling (or a trap with a sub-cooling membrane) is to be used.

Holding temperature below 100 ºC

In the case of trace heaters for pipes with a holding temperature below 100 ºC, e.g. in the case of frost protection, it can be sensible to select an adjustable trap, so the condensate is not discharged at the saturation temperature but at a temperature a few degrees lower. *Chapter 6.0 Condensate Management* (*Item 6.7 How to avoid the flash steam cloud*) shows a sample calculation. In this example (tracer heating for frost protection) the saving is calculated with a balanced pressure steam trap at 40 degrees sub-cooling instead of a trap set at saturation temperature. In order to avoid water hammer, it is advisable not to route condensate from a trap set for sub-cooling or equipped with a sub-cooling membrane to the same collection station as condensate with saturation temperature.

Critical holding temperature

If the temperature differential between the steam and the product is less than 30 Kelvin, the trap should be set to saturation temperature. This also applies to process applications with unstable steam pressure or if the holding temperature for the process application is considered to be critical. For such applications, it is possible to fall back on the selection of thermodynamic, membrane capsule or bimetallic traps without an adjustment facility and sub-cooling. When using bimetallic traps for critical process applications, it should be taken into account that cooling increases with back pressure for these traps. In contrast, balanced pressure steam traps are not sensitive to back pressure. The use of float-type traps is definitely advisable, where the fitting dimensions allow this.

Tracer heater condensate with different steam pressures

If different condensate flows are routed to one collection station from tracer heaters with different steam pressures, water hammer will not occur, if the condensate is at saturation temperature. The condensate flows have already flashed downstream of the traps.

10.0 Appendix

10.1 Saturated steam tables

Steam pressure	Saturated steam temperature	Specific volume	Specific mass	Sensible heat	Total enthalpy	Enthalpy of evaporation
P bar (abs.)	T_s °C	v_g m^3/kg	ρ kg/m^3	h_f kJ/kg	h_g kJ/kg	h_{fg} kJ/kg
0.010	6.98	129.200	0.00774	29.34	2514.40	2485.00
0.015	13.04	87.980	0.01137	54.71	2525.50	2470.70
0.020	17.51	67.010	0.01492	73.46	2533.60	2460.20
0.025	21.10	54.260	0.01843	88.45	2540.20	2451.70
0.030	24.10	45.670	0.02190	101.00	2545.60	2444.60
0.035	26.69	39.480	0.02533	111.85	2550.40	2438.50
0.040	28.98	34.800	0.02873	121.41	2554.50	2433.10
0.045	31.04	31.140	0.03211	129.99	2558.20	2428.20
0.050	32.90	28.190	0.03547	137.77	2561.60	2423.80
0.055	34.61	25.770	0.03880	144.91	2564.70	2419.80
0.060	36.18	23.740	0.04212	151.50	2567.50	2416.00
0.065	37.65	22.020	0.04542	157.64	2570.20	2412.50
0.070	39.03	20.530	0.04871	163.38	2572.60	2409.20
0.075	40.32	19.240	0.05198	168.77	2574.90	2406.20
0.080	41.53	18.100	0.05523	173.86	2577.10	2403.20
0.085	42.69	17.100	0.05848	178.69	2579.20	2400.50
0.090	43.79	16.200	0.06171	183.28	2581.10	2397.90
0.095	44.83	15.400	0.06493	187.65	2583.00	2395.30
0.10	45.83	14.670	0.06814	191.83	2584.80	2392.90
0.15	54.00	10.020	0.09977	225.97	2599.20	2373.20
0.20	60.09	7.650	0.1307	251.45	2609.90	2358.40
0.25	64.99	6.204	0.1612	271.99	2618.30	2346.40
0.30	69.12	5.229	0.1912	289.30	2625.40	2336.10
0.40	75.89	3.993	0.2504	317.65	2636.90	2319.20
0.45	78.74	3.576	0.2796	329.64	2641.70	2312.00
0.50	81.35	3.240	0.3086	340.56	2646.00	2305.40
0.55	83.74	2.964	0.3374	350.61	2649.90	2299.30
0.60	85.95	2.732	0.3661	359.93	2653.60	2293.60

Steam pressure	Saturated steam temperature	Specific volume	Specific mass	Sensible heat	Total enthalpy	Enthalpy of evaporation
P bar (abs.)	T_s °C	v_g m^3/kg	ρ kg/m^3	h_f kJ/kg	h_g kJ/kg	h_{fg} kJ/kg
0.65	88.02	2.535	0.3945	368.62	2656.90	2288.30
0.70	89.96	2.365	0.4229	376.77	2660.10	2283.30
0.75	91.79	2.217	0.4511	384.45	2663.00	2278.60
0.80	93.51	2.087	0.4792	391.72	2665.80	2274.00
0.85	95.15	1.972	0.5071	398.63	2668.40	2269.80
0.90	96.71	1.869	0.5350	405.21	2670.90	2265.60
0.95	98.20	1.777	0.5627	411.49	2673.20	2261.70
1.00	99.63	1.694	0.5904	417.51	2675.40	2257.90
1.50	111.37	1.159	0.8628	467.13	2693.40	2226.20
2.00	120.23	0.885	1.1290	504.70	2706.30	2201.60
2.50	127.43	0.718	1.3920	535.34	2716.40	2181.00
3.00	133.54	0.606	1.6510	561.43	2724.70	2163.20
3.50	138.87	0.524	1.9080	584.27	2731.60	2147.40
4.00	143.62	0.462	2.1630	604.67	2737.60	2133.00
4.50	147.92	0.414	2.4170	623.16	2742.90	2119.70
5.00	151.84	0.375	2.6690	640.12	2747.50	2107.40
5.50	155.46	0.343	2.9200	655.78	2751.70	2095.90
6.00	158.84	0.316	3.1700	670.42	2755.50	2085.00
6.50	161.99	0.293	3.4190	684.12	2758.80	2074.00
7.00	164.96	0.273	3.6670	697.06	2762.00	2064.90
7.50	167.75	0.255	3.9150	709.29	2764.80	2055.50
8.00	170.41	0.240	4.1620	720.94	2767.50	2046.50
8.50	172.94	0.227	4.4090	732.02	2769.90	2037.90
9.00	175.36	0.215	4.6550	742.64	2772.10	2029.50
9.50	177.66	0.204	4.9010	752.81	2774.20	2021.40
10.00	179.88	0.194	5.1470	762.61	2776.20	2013.60
11.00	184.07	0.175	5.6370	781.13	2779.70	1998.50
12.00	187.96	0.163	6.1270	798.43	2782.70	1984.30
13.00	191.61	0.151	6.6170	814.70	2785.40	1970.70
14.00	195.04	0.141	7.1060	830.08	2787.80	1957.70
15.00	198.29	0.132	7.5960	844.67	2789.90	1945.20
16.00	201.37	0.124	8.0850	858.56	2791.70	1933.20
17.00	204.31	0.117	8.5750	871.84	2793.40	1921.50
18.00	207.11	0.110	9.0650	884.58	2794.80	1910.30
19.00	209.80	0.105	9.5550	896.81	2796.10	1899.30
20.00	212.37	0.100	10.0500	908.59	2797.20	1886.60
21.00	214.85	0.095	10.5400	919.96	2798.20	1878.20
22.00	217.24	0.091	11.0300	930.95	2799.10	1868.10
23.00	219.55	0.087	11.5200	941.60	2799.80	1858.20
24.00	221.78	0.083	12.0200	951.93	2800.40	1848.50
25.00	223.94	0.080	12.5100	961.96	2800.90	1839.00
26.00	226.04	0.077	13.0100	971.72	2801.40	1829.60
27.00	228.07	0.074	13.5100	981.22	2801.70	1820.50
28.00	230.05	0.071	14.0100	990.48	2802.00	1811.50
29.00	231.97	0.069	14.5100	999.53	2802.20	1802.60

10.0 Appendix

Steam pressure	Saturated steam temperature	Specific volume	Specific mass	Sensible heat	Total enthalpy	Enthalpy of evaporation
P bar (abs.)	T_s °C	v_g m^3/kg	ρ kg/m^3	h_f kJ/kg	h_g kJ/kg	h_{fg} kJ/kg
30.00	233.84	0.067	15.0100	1008.40	2802.30	1793.90
32.00	237.45	0.062	16.0200	1025.40	2802.30	1776.90
34.00	240.88	0.059	17.0300	1041.80	2802.10	1760.30
36.00	244.16	0.055	18.0500	1057.60	2801.70	1744.20
38.00	247.31	0.052	19.0700	1072.70	2801.10	1728.40
40.00	250.33	0.050	20.1000	1087.40	2800.30	1712.90
42.00	253.24	0.047	21.1400	1101.60	2799.40	1697.80
44.00	256.05	0.045	22.1800	1115.40	2798.30	1682.90
46.00	258.75	0.043	23.2400	1128.80	2797.00	1668.30
48.00	261.37	0.041	24.2900	1141.80	2795.70	1653.90
50.00	263.91	0.039	25.3600	1154.50	2794.20	1639.70
55.00	269.93	0.036	28.0700	1184.90	2789.90	1605.00
60.00	275.55	0.032	30.8300	1213.70	2785.00	1571.30
65.00	280.82	0.030	33.6500	1241.10	2779.50	1538.40
70.00	285.79	0.027	36.5300	1267.40	2773.50	1506.00
75.00	290.50	0.025	39.4800	1292.70	2766.90	1474.20
80.00	294.97	0.024	42.510	1317.10	2759.90	1442.80
85.00	299.23	0.022	56.610	1340.70	2752.50	1411.70
90.00	303.31	0.021	48.790	1363.70	2744.60	1380.90
95.00	307.21	0.019	52.060	1386.10	2736.40	1350.20
100.00	310.96	0.018	55.430	1408.00	2727.70	1319.70
110.00	318.05	0.016	62.480	1450.60	2709.30	1258.70
120.00	324.65	0.014	70.010	1491.80	2689.20	1197.40
130.00	330.83	0.013	78.140	1532.00	2667.00	1135.00
140.00	336.64	0.012	86.990	1571.60	2642.40	1070.70
150.00	342.13	0.010	96.710	1611.00	2615.00	1004.00
160.00	347.33	0.009	107.400	1650.50	2584.90	934.30
170.00	352.26	0.008	119.500	1691.70	2551.60	859.90
180.00	356.96	0.007	133.400	1734.80	2513.90	779.10
190.00	361.43	0.007	149.800	1778.70	2470.60	692.00
200.00	365.70	0.006	170.200	1826.50	2418.40	591.90
220.00	373.69	0.004	268.300	2011.10	2195.60	184.50
221.20	374.15	0.003	315.500	2107.40	2107.40	

10.2 Superheated steam tables

Pressure	Specific enthalpy (h_g) [kJ/kg] at a temperature [°C]							
p bar	200	220	240	260	280	300	320	340
1	2875.40	2915.00	2954.60	2994.00	3034.40	3074.50	3114.80	3155.30
2	2870.50	2910.80	2951.10	2991.40	3031.70	3072.10	3112.60	3153.30
3	2865.50	2906.60	2947.50	2988.20	3028.90	3069.70	3110.50	3151.40
4	2860.40	2902.30	2943.90	2985.10	3026.20	3067.20	3108.30	3149.40
5	2855.10	2898.00	2940.10	2981.90	3023.40	3064.80	3106.10	3147.40
6	2849.70	2893.50	2936.40	2978.70	3020.60	3062.30	3103.90	3145.40
7	2844.20	2888.90	2932.50	2975.40	3017.70	3059.80	3101.60	3143.40
8	2838.60	2884.20	2928.60	2972.10	3014.90	3057.30	3099.40	3141.40
9	2832.70	2879.50	2924.60	2968.70	3012.00	3054.70	3097.10	3139.40
10	2826.80	2874.60	2920.60	2965.20	3009.00	3052.10	3094.90	3137.40
11	2820.70	2869.60	2916.40	2961.80	3006.00	3049.60	3092.60	3135.30
12	2814.40	2864.50	2912.20	2958.20	3003.00	3046.90	3090.30	3133.20
13	2808.00	2859.30	2908.00	2954.70	3000.00	3044.30	3088.00	3131.20
14	2801.40	2854.00	2903.60	2951.00	2996.90	3041.60	3085.60	3129.10
15	2794.70	2848.60	2899.20	2947.30	2993.70	3038.90	3083.30	3127.00
16	-	2843.10	2894.70	2943.60	2990.60	3036.20	3080.90	3124.90
18	-	2831.70	2885.40	2935.90	2984.10	3030.70	3076.10	3120.60
20	-	2819.90	2875.90	2928.10	2977.50	3025.00	3071.20	3116.30
22	-	2807.50	2866.00	2920.00	2970.80	3019.30	3066.20	3112.00
24	-	-	2855.70	2911.60	2963.80	3013.40	3061.10	3107.50
26	-	-	2845.20	2903.00	2956.70	3007.40	3056.00	3103.00
28	-	-	2834.20	2894.20	2949.50	3001.30	3050.80	3098.50
30	-	-	2822.90	2885.10	2942.00	2995.10	3045.40	3093.90
32	-	-	2811.20	2875.80	2934.40	2988.70	3040.00	3089.20
34	-	-	-	2866.20	2926.60	2982.20	3034.50	3084.40
36	-	-	-	2856.30	2918.60	2975.60	3028.90	3079.60
38	-	-	-	2846.10	2910.40	2968.90	3023.30	3074.80
40	-	-	-	2835.60	2902.00	2962.00	3017.50	3069.80
42	-	-	-	2824.80	2893.50	2955.00	3011.60	3064.80
44	-	-	-	2813.60	2884.70	2947.80	3005.70	3059.70
46	-	-	-	2802.00	2875.60	2940.50	2999.60	3054.60
48	-	-	-	-	2866.40	2933.10	2993.40	3049.40
50	-	-	-	-	2856.90	2925.50	2987.20	3044.10
55	-	-	-	-	2831.80	2905.70	2971.00	3030.50
60	-	-	-	-	2804.90	2885.00	2954.20	3016.50
70	-	-	-	-	-	2839.40	2918.30	2987.00
80	-	-	-	-	-	2786.80	2878.70	2955.30
90	-	-	-	-	-	-	2834.30	2920.90
100	-	-	-	-	-	-	2783.50	2883.40
110	-	-	-	-	-	-	2723.50	2841.70
120	-	-	-	-	-	-	-	2794.70
130	-	-	-	-	-	-	-	2740.60
140	-	-	-	-	-	-	-	2675.70
150	-	-	-	-	-	-	-	-
160	-	-	-	-	-	-	-	-
180	-	-	-	-	-	-	-	-
200	-	-	-	-	-	-	-	-
250	-	-	-	-	-	-	-	-

Pressure	Specific enthalpy (h_g) [kJ/kg] at a temperature [°C]							
p bar	**360**	**380**	**400**	**420**	**440**	**460**	**480**	**500**
1	3196.00	3237.00	3278.20	3319.70	3361.40	3403.40	3445.60	3488.10
2	3194.20	3235.40	3276.70	3318.30	3360.10	3402.10	3444.50	3487.00
3	3192.40	3233.70	3275.20	3316.80	3358.80	3400.90	3443.30	3486.00
4	3190.60	3232.10	3273.60	3315.40	3357.40	3399.70	3442.10	3484.90
5	3188.80	3230.40	3272.10	3314.00	3356.10	3398.40	3441.00	3483.80
6	3187.00	3228.70	3270.60	3312.60	3354.80	3397.20	3439.80	3482.70
7	3185.20	3227.10	3269.00	3311.20	3353.40	3395.90	3439.60	3481.60
8	3183.40	3225.40	3267.50	3309.70	3352.10	3394.70	3437.50	3480.50
9	3181.60	3223.70	3266.00	3308.30	3350.80	3393.50	3436.30	3479.40
10	3179.70	3222.00	3264.40	3306.90	3349.50	3392.20	3435.10	3478.30
11	3177.90	3220.30	3262.90	3305.40	3348.10	3391.00	3434.00	3477.20
12	3176.00	3218.70	3261.30	3304.00	3346.80	3389.70	3432.80	3476.10
13	3174.10	3217.00	3259.20	3302.50	3345.40	3388.50	3431.60	3475.00
14	3172.30	3215.30	3258.20	3301.10	3344.10	3387.20	3430.50	3473.90
15	3170.40	3213.50	3256.60	3299.70	3342.80	3386.00	3429.30	3472.80
16	3168.50	3211.80	3255.00	3298.20	3341.40	3384.70	3428.10	3471.70
18	3164.70	3208.40	3251.90	3295.30	3338.70	3382.20	3425.80	3469.50
20	3160.80	3204.90	3248.70	3292.40	3336.00	3379.70	3423.40	3467.30
22	3156.90	3201.40	3245.50	3289.40	3333.30	3377.10	3421.10	3465.10
24	3153.00	3197.80	3242.30	3286.50	3330.60	3374.60	3418.70	3462.90
26	3149.00	3416.33	3239.00	3283.50	3327.80	3372.10	3416.30	3460.60
28	3145.00	3190.70	3235.80	3280.50	3325.10	3369.50	3413.90	3458.40
30	3140.90	3187.00	3232.50	3277.50	3322.30	3367.00	3411.60	3456.20
32	3136.80	3183.40	3229.20	3274.50	3319.50	3364.40	3409.20	3454.00
34	3132.70	3179.70	3225.90	3271.50	3316.80	3361.80	3406.80	3451.70
36	3128.40	3028.90	3222.50	3268.40	3314.00	3359.20	3404.40	3449.50
38	3124.20	3172.20	3219.10	3265.40	3311.20	3356.60	3402.00	3447.20
40	3119.90	3168.40	3215.70	3262.30	3308.30	3354.00	3399.60	3445.00
42	3115.50	3115.00	3212.30	3259.20	3305.50	3351.40	3397.70	3442.70
44	3111.10	3160.60	3208.80	3256.00	3302.60	3348.80	3394.70	3440.50
46	3106.70	3156.70	3205.30	3252.90	3299.80	3346.20	3392.30	3438.20
48	3102.20	3152.80	3201.80	3249.70	3296.90	3343.50	3389.80	3435.90
50	3097.60	3148.80	3198.30	3246.60	3294.00	3340.90	3387.40	3433.70
55	3085.90	3138.60	3189.30	3238.50	3286.70	3334.20	3381.20	3427.90
60	3074.00	3128.30	3180.10	3230.30	3279.30	3327.40	3375.00	3422.20
70	3049.10	3106.70	3161.20	3213.50	3264.20	3313.70	3362.40	3410.60
80	3022.70	3084.20	3141.60	3196.20	3248.70	3299.70	3349.60	3398.80
90	2994.80	3060.50	3121.20	3178.20	3232.70	3285.30	3336.50	3386.80
100	2964.80	3035.70	3099.90	3159.70	3216.20	3270.50	3323.20	3374.60
110	2932.80	3009.60	3077.80	3140.50	3199.40	3255.50	3309.60	3362.20
120	2898.10	2982.00	3054.80	3120.70	3182.00	3240.00	3295.70	3349.60
130	2860.20	2952.70	3030.70	3100.20	3164.10	3224.20	3281.60	3336.80
140	2818.10	2921.40	3005.60	3079.00	3145.80	3208.10	3267.10	3323.80
150	2770.80	2887.70	2979.10	3057.00	3126.90	3191.50	3252.40	3310.60
160	2716.50	2851.10	2951.30	3034.20	3107.50	3174.50	3237.40	3297.10
180	2569.10	2766.60	2890.30	2985.80	3066.90	3139.40	3206.50	3269.60
200	-	2660.20	2820.50	2932.90	3023.70	3102.70	3174.40	3241.10
250	-	-	2582.00	2774.10	2901.70	3002.30	3088.50	3165.90

10.3 Symbols used, units

Symbol	Definition of quantity	SI unit	(Old unit)
A	Area, cross-section	m^2	
a	Thermal diffusivity	m/m K, 1/K	
α	Coefficient of heat transfer	W/m^2 K	($kcal/m^2$ grd)
c	Specific heat capacity	J/kg K	(J/kg grd)
c_p , c_v	Specific heat, p = constant, v = constant	J/kg K	
ς	Specific weight (density), medium	kg/m^3, kg/dm^3	
h, (i)	Specific enthalpy (internal energy)	kJ/kg	(kcal/kg)
h_g	Enthalpy of superheated steam, total energy of saturated steam	kJ/kg	
h_f	Enthalpy of water (sensible heat)	kJ/kg	
k_l	Coefficient of heat transmission per meter	W/m K	
L, D, d, s	Length, diameter, thickness	m	
λ	Coefficient of thermal conductivity	W/m K	(kcal/m h grd)
ν	Kinematic viscosity	m^2/s	
M	Mass (weight)	kg	
m_K	Mass flow rate, condensate	kg/h	
m_D	Mass flow rate, steam	kg/h	
p	Pressure	bar, Pa	(kp/cm^2, ata)
p_1	Service pressure, upstream pressure	bar	
p_2	Back pressure, condensate pipe	bar	
p_S	Pressure at boiling point of water	bar	
Δp	Pressure difference (p1 - p2), pressure loss	bar	
$\dot{Q}$	Heat flow	W	
h_{fg}	Enthalpy of evaporation for 1 kg of water	kJ/kg	(kcal/kg)
s	Specific entropy	kJ/kg K	(kcal/kg grd)
s', s''	Specific entropy of water, steam	kJ/kg K	
t, ϑ,T [1)]	Temperature, medium	°C, K	(grd)
Δt, $\Delta\vartheta$,ΔT	Temperature difference (t1 - t2), medium	°C	(grd)
t_D, t_K	Steam, condensate temperature	°C	
t_S	Boiling point of condensate, water	°C	
V_D, V_K	Steam, condensate volume	m^3	
v_g	Specific volume	m^3/kg, dm^3/kg	
w	Flow velocity	m/s	
x	Relative steam moisture, percentage of	kg/kg	
ξ	Resistance coefficient / zeta value		
ν		m^2/s	

[1)] T in Kelvin
The triple point of water defined by international accepted standards by Guildner, Johnson and Jones (1976) is at (611,657 ± 0,010) Pa (ca. 6 mbar) and 273,16 K (0,01 °C).

Multiples and fractions of units:

d	Deci	= 10^{-1}	h	Hecto	= 10^2
c	Centi	= 10^{-2}	k	Kilo	= 10^3
m	Milli	= 10^{-3}	M	Mega	= 10^6
m	Micro	= 10^{-6}	G	Giga	= 10^9

10.4 Conversion factors for units

Pressure units	N/m^2 (Pa)	bar	kp/cm^2	Torr	lbf/ft^2
1 Pascal	1	10^{-5}	1.02×10^{-5}	7.505×10^{-3}	2.088×10^{-2}
1 Bar	105	1	1.02	7.505×10^{2}	2.088×10^{3}
1 kp/cm^2	9.806×10^{4}	0.981	1	7.355×10^{2}	2.048×10^{3}
1 pound-force/square foot	47.88	4.788×10^{-4}	4.883×10^{-4}	0.3591	1

Temperatures	°K	°C	°F
T_K (K - Kelvin)	T_K	$T_K - 273$	$1.8 (T_K - 273) + 32$
t_C (°C - Celsius)	$t_C + 273$	t_C	$1.80\, t_C + 32$
t_F (°F - Fahrenheit)	$0.5556\, t_F + 255.2$	$0.5556 (t_F - 32)$	t_F
(Conversion factor for temperatures specified in °T_R (Rankine): $T_R = 1.80\, T_K$)			

Energy units	J = N m = W s	KW h	kcal	BtU
1 joule	1	2.778×10^{-7}	2.388×10^{-4}	9.478×10^{-4}
1 kilowatt hour	3.600×10^{6}	1	8.598×10^{2}	3.412×10^{3}
1 kilocalorie	4.187×10^{3}	1.162×10^{-3}	1	3.968
1 British thermal unit	1.055×10^{3}	2.933×10^{-4}	0.252	1
Power units	W = J/s = N m/s	kcal/h	kp m/s	BtU/s
1 watt	1	0.8597	0.102	9.478×10^{-4}
1 kilocalorie / hour	1.163	1	0.119	1.102×10^{-3}
1 kilopond metre / s	9.807	8.431	1	9.259×10^{-3}
1 British thermal unit / s	1.055×10^{3}	9.073×10^{2}	1.076×10^{2}	1

Unit	$W/m^2\, K$	$kcal/m^2\, h\, K$	W/m K	kcal/m h K
1 W / m^2 grd (α, k)	1	0.860	-	-
1 kcal / m^2 h grd	1.163	1	-	-
1 W / m grd (λ)	-	-	1	0.860
1 kcal / m h grd (λ)	-	-	1.163	1
Coefficients of heat transfer (α), heat transmission (k) and thermal conductivity (λ)				

Selected units: Conversion to Anglo-American units:

1 Fuß (foot. ft) = 12 in = 0.3048 m	1 m = 3.281 ft	1 km = 0.6214 mile
1 Quadrat-Inch (in^2) = 6.452 cm^2	1 cm^2 = 0.155 in^2	1 m^2 = 10.764 ft^2
1 Quadratyard (yd^2) =9 ft2	9 ft^2 = 0.8361 m^2	1 m^2 = 1.196 yd^2
1 Kubikfuß (ft^3) = 0.0283 m^3	1 m^3 = 35.31 ft^3	1 m^3 = 1.308 yd^3
1 barrel = 36 Imp. gal = 163.6 l	159 l = 1U.S. barrel	1 m/s = 3.281 ft/s
1 m^3 /h = 0.00981 ft^3 /s	1 t/h = 0.6124 lb/s	1 pound-force (lbf) = 4.4482 N
1 bar = 14.5 lbf/ in^2	1 kPa = 20.89 lbf/ft^2	1 J = 0.7376 ft-lbf
1 KWh = 3413 BTU	1 W = 3.412 BTU/h	1 KW = 1.342 HP (horsepower)

10.5 Flow velocities in pipes

For calculation equations, information about volume flow rates and flow velocities in pipes (condensate, steam), refer to *Chapter 4.0 Pipes*.

Equations:

$V = w\ d^2/354$ $w = (V/d^2) \times 354$	$V_K = m_K\ v_g$ $V_D = m_D\ v''$	m^3/h m/s
w	Flow velocity	m/s
d	Inside pipe diameter	m
v_g	Specific volume	m^3/kg
V_K, V_D	Volume flow of condensate, steam	m^3/h
m_K, m_D	Mass flow of condensate, steam	kg/h

10.6 Recommended values in technical literature

Steam	**w (m/s)**	
- Steam vent or flash steam	15 to 25 m/s	
- Saturated steam pipes	20 to 40 m/s	
- Low-pressure superheated steam	Less than 10 bar	20 to 35 m/s
- Medium-pressure superheated steam	10 to 40 bar	20 to 40 m/s
- High-pressure superheated steam	40 to 125 bar	30 to 60 m/s
- High-pressure superheated steam	High capacities	45 to 65 m/s
Water		
- Feedwater inlet pipes (suction pipes)	0.5 to 1.0 m/s	
- Feedwater discharge pipes	1.5 to 3.5 m/s	
- Feedwater preheaters	0.01 to 0.15 m/s	
- Cooling water suction pipes	0.7 to 1.5 m/s	
- Cooling water discharge pipes	1.0 to 4.5 m/s	
- Condensate pipes, slightly sub-cooled condensate		0.5 to 1.0 m/s
- Condensate pipes, increased sub-cooling of condensate		1.0 to 2.0 m/s
- Drinking, service water main pipes	1.0 to 3.0 m/s	
- Local drinking, service water systems	0.6 to 1.0 m/s	
- Turbine pipes, small diameter	2.0 to 4.0 m/s	
- Turbine pipes, large diameter	3.0 to 7.0 m/s	
Gases		
- Gas pipes	Max. 2 bar	4.0 to 20 m/s
- Gas pipes	Max. 5 bar	11 to 35 m/s
- Gas, household pipes	Max. 1.0 m/s	
- Compressed air pipes		15 to 25 m/s
Oils, naphthas		
- Oil pipelines	1.5 to 2.0 m/s	
- Heavy oil pipes	0.5 to 2.0 m/s	
- Naphtha, benzene, gas oil pipes	1.0 to 2.0 m/s	

Coefficients of thermal conductivity (λ), heat transfer (α) and heat transmission (k)

Coefficients of thermal conductivity λ in W/m K at approx. 20 °C

Substances			
Earth	1 to 1.9	Synthetic resin foam	0.03 to 0.05
Glass	0.7 to 1.4	Air (still)	0.031
Graphite	Max. 160	Methyl alcohol	0.12 to 0.20
Glass wool	0.05 to 0.15	Porcelain	0.8 to 1.9
Glycerin	0.28 to 0.30	Soot	0.10
Naphtha, benzene	0.1 to 0.14	Fireclay bricks	0.6 to 1.20
Hard rubber	0.13 to 0.20	Steel (0.2 % C)	30 to 50
Wood	0.42	Slag wool	0.04 to 0.13
Gravel concrete, mortar	0.9 to 1.5	Paraffin oil	0.13
Diatomaceous earth	0.05 to 0.18	Toluene	0.13
Scale	0.5 to 2.0	Transformer oil	0.13
Corkboard	0.03 to 0.06	Water	0.65

Coefficient of heat transfer α in W/m^2 K

Heat transfer is carried out by the convection of liquid or gaseous media (fluidium) on a solid wall with a given temperature drop. The heat transfer coefficient α = f(w, Δt, Re, Nu, Pr, etc.) is determined by means of experiments, but can also be calculated using the similarity theory and dimensionless variables.

Variables		
Nusselt number	$Nu = \alpha\, l / \lambda$	$a = Nu\, l / l$
Reynolds number	$Re = w\, l / \nu$	
Prandtl number	$Pr = Pe / Re = \nu / \alpha$	
Péclet number	$Pe = w\, l / \alpha$	
Grashof number	$Gr = \beta\, g\, \Delta t\, l^3 / \nu^2$	
Non-circular cross-section	$d \equiv d_{G\,LI} = 4\, A / U$ (hydraulic diameter)	

Calculation equations for forced turbulent flow (selected)	
Flow in the pipe	
Liquids	$Nu \approx 0.032\, Re^{0.8}\, Pr^{0.34}\, (d/l)^{0.054}$ for Pr = 0.7 to 370, $Re \geq 4500$, $t_m = 0.5\,(t_W + t_F)$ °C
Water	$\alpha \approx 3370\, w^{0.85}\, (1+ 0.014\, t_m)$; for $d \leq 0.1$ m
Gases, vapours	$Nu \approx 0.024\, Re^{0.786}\, Pr^{0.45}\, (1+ (d/l)^{0.67})$ (non-condensing)
Air	$\alpha \approx (4.13 + 0.19\, t / 100)\, w^{0.75}\, d^{-0.25}$
Liquids, gases: Flat wall	$Nu \approx 0.036\, Re^{0.8}\, Pr^{0.33}$; l = length of wall, t_m
Liquids, gases: Around the tube bundle – perpendicular	$Nu \approx C\, Re^n\, Pr^{0.31}$; $Re \geq 4000$ --> C = 0.19; n = 0.62; $Re \approx 40000$ --> C = 0.027; n = 0.81

Calculation equations for forced turbulent flow (selected)	
Flow in the pipe	
Air around the tube bundle	$\alpha \approx 1.63\ T^{0.25}\ w^{0.61} / d^{0.39}$
Air parallel "	$\alpha \approx 7.5\ w^{0.78}$; (w > 5 m/s)
Other gases " "	$\alpha \approx 7.5\ w^{0.78}$ Z, (e.g. Z for CO_2, O_2, SO_2 = 1; NH_3 = 1.25) (He = 1.1; H_2 = 1.5; carbonic acid = 1.12, etc.)

Condensation of saturated steam (properties for t_m):	
Film condensation	$\alpha \approx 0.943\ (\lambda^3\ \varpi^2\ g\ r / \nu\ h\ (t_S - t_W))^{0.25}$; g = 9.81 m /$s^2$
Condensation of superheated steam	$\alpha \approx$ like saturated steam, satisfied for r --> (h - h_f) and $t_W < T_S$;
Evaporation of boiling water – nucleate boiling:	
Vertical tube	$\alpha \approx 2.5\ p^{0.18}\ q^{0.7}$ or $\alpha \approx 59\ (\Delta t)^{2.33}$, ($\Delta t = t_W - t_{fl}$); P = 1 to 100 bar; $12000 \le q \le 46000$ (W/m^2), q = heat flux in W/m^2
Horizontal plate	$\alpha \approx 5.6\ (\Delta t)^3$

A few approximated α values in W/m^2 K (to facilitate rough estimates – ϕ values)		
Evaporation of boiling water, vertical surface	$\alpha \approx 4650$	W/m^2 K
Evaporation of boiling water, horizontal surface	$\alpha \approx 2250$	W/m^2 K
Condensing water vapour	$\alpha \approx 11600$	W/m^2 K
Flowing water	$\alpha \approx 3500$	W/m^2 K
Still water	$\alpha \approx 580$	W/m^2 K
Air around walls, tubes	$\alpha \approx 100$	W/m^2 K
Superheated steam (dry, for 400 °C, ϕ d = 0.1 m)	$\alpha \approx 880$	W/m^2 K
Flue gases or gases, general	$\alpha \approx 110$	W/m^2 K
Heat transmission coefficients k in W/m^2 K		
Heat transmission is the transfer of heat from a liquid or gaseous substance with t_1 through a membrane to a colder liquid or gaseous substance with t_2. Its subprocesses are heat transfer a from t_1 --> t_{W1} (wall surface 1), heat conduction λ to t_{W2} (wall surface 2), heat transfer α from t_{W2} --> t_2 of the colder substance. The heat transmission resistance 1/k is dependent on a large number of factors and influences, such as flow velocity, baffle plates or internals in the heat exchangers, insulation, dirt layers and the temperature curve along the heating surface. The k value must be calculated using the function equations in the technical literature. Example for a multi-layer wall: $k = (1/\alpha_1 + å\ (s/\lambda) + 1/\alpha_2)^{-1}$		

A few approximated k values for different heat exchanger types to facilitate rough estimates			
Heat exchanger / wall material	**Shell side / medium**	**Tube bundle, coil / medium**	**k W/m^2 K**
Tube bundle heat exchanger	Gas (1 bar)	Gas (1 bar)	6 to 25
	Gas (1 bar)	Liquid	20 to 60
	Liquid	Gas (high-pressure)	450
	Liquid	Liquid	670
	Water vapour	Liquid	770
Evaporator, natural circulation	Water vapour	Low-viscosity liquids	1000
Evaporator, forced circulation	Water vapour	Liquids	2500
Condenser	Organic vapours	Cooling water	650
Waste heat boiler	Boiling water	Hot gases	15 to 40
Double-pipe heat exchanger	Gas (1 bar)	Gas (1 bar)	12 to 35
	Liquid	Gas (high-pressure)	350
	Liquid	Liquid	750
Spray cooler	Cooling water	Gas (1 bar)	30 to 50
Channel air preheater	Gas (1 bar)	Gas (1 bar)	12 to 30
Spiral heat exchanger	Water vapour	Liquid	2200
Cooler	Water	Oil distillate	100
Tubular heat exchanger	Water	Water	350
Tubular air preheater	Water vapour	Water	1100
Preheater	Flue gases	Crude oil, tar	25
	Flue gases	Water	20
Submerged-type evaporator	Flash steam	Water or products	3300
	Heating medium	Medium to be heated	
Steel, cast iron	Air	Air	12
Cast, wrought iron	Water	Air	15
Copper, brass	Water	Flue gas	20
Cast, wrought iron	Water	Water	330
Copper	Water	Water	410
Cast iron, steel	Water vapour	Water	1000
Copper	Water vapour	Water	1150
Cast iron, steel	Water vapour	Air	20
Copper	Water vapour	Air	25

Properties – specific heat c, temperatures t_S, density ς			
Substances	**Specific heat c** **kJ/(kg K)**	**Density ς** **kg/ dm^3**	**Boiling point** **t_S °C**
Selected liquids:			
Acetone	2.156	0.791	56
Alcohol	2.428	0.790	78
Ammonia solution	4.180	0.91	-33.4
Benzene	1.842	0.88	80
Naphtha	2.278	0.680	95
Crude oil, oil	1.883 to 1.675	0.91	390
Glycerin	2.412	1.26	290
Heating oil	1.975	0.92	180 to 350
Naphthalene	1.298	1.280	218
Phenol	1.394	1.350	180.5
Paraffin oil	3.270	0.900	300
Coal tar	1.670	1.200	300
Toluene	1.675	0.868	110.8
Water (20 °C)	4.187	0.998	100
Ethyl alcohol	2.290	0.806	78.3
Solids:			
Ash	0.837	0.700	-
Aluminium (99.5 %)	0.921	2.700	2270
Concrete (gravel)	0.879	2.00	-
Glass	0.837	2.500	2600
Graphite	0.879	1.8 to 2.3	-
Grey cast iron	0.535	7.1 to 7.3	2500
Wood (spruce)	2.386	0.500	-
Hard rubber	1.420	1.2 to 1.8	-
Diatomaceous earth	0.879	2.0 to 2.6	-
Cork	2.051	0.2 to 0.3	-
Copper	0.093	8.900	2310
Porcelain, stoneware	0.880; 0.775	2.40	-
Fireclay	0.795	1.8 to 2.2	-
Slag	0.837	2.6	-
Steel (0.2 % C)	0.461	7.85	2500
Sulphur (native)	0.754	1.98	444.6

Properties – specific heat c, temperatures t_S, density ς			
Substances	**Specific heat c** **kJ/(kg K)**	**Density ς** **kg/ m^3**	**Boiling point** **t_S °C**
Gases, vapours (at 0 °C, p = 760 Torr (conversion: kJ/kg K x molecular mass = kJ/kmol K)):			
Ammonia NH_3	2.21	0.771	-33
Acetylene C_2H_2	1.683	1.171	-84
Carbon dioxide CO_2	0.837	1.977	-78.5
Air	1.005	1.293	-192
Methane CH_4	2.219	0.717	-161.4
Nitrogen N_2	1.046	1.250	-195.7
Oxygen O_2	0.921	1.429	-183
Sulphur dioxide SO_2	0.628	2.927	-10
Hydrogen H_2	14.28	0.090	-253
Ethane C_2H_6	1.728	1.356	-89
(Sources: Handbook of Mechanical Engineering 1, Dubbel; Technical Thermodynamics, Faltin; etc.)			

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