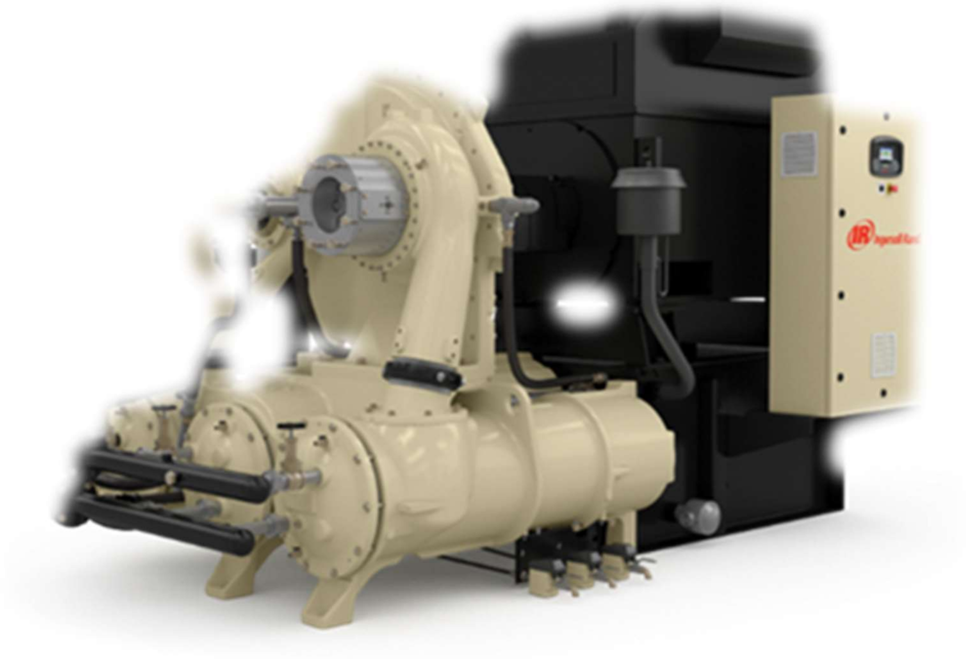
Reference Manual for

Steam and Compressed Air Systems



Professional Level Elective Module of Singapore Certified Energy Manager (SCEM) Programme

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By
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PREFACE

The Singapore Certified Energy Manager (SCEM) programme offers a formal training and certification system for energy managers in Singapore and is co-administered by the National Environment Agency (NEA) and The Institution of Engineers, Singapore (IES) since 2008. The programme equips facility managers, engineers, technicians and others who intend to build their careers as energy professionals with the technical skills and competencies needed to manage energy services within their organisations.

The Steam and Compressed Air Systems module is one of the elective modules in the SCEM programme. This reference manual aims to help SCEM candidates with their course work and serve as reference material for practising energy managers.

The reference manual contains fifteen chapters on different aspects of how to design and operate typical steam, compressed air and heat recovery systems to be energy efficient. Each chapter includes a brief introduction to assist readers who may not be familiar with some of the basic concepts associated with each topic, and the expected learning objectives.

Chapters 1 to 4 cover steam properties, boilers, steam applications and how to optimise steam systems. Thereafter, Chapter 5 describes the main aspects of how to conduct an energy audit of a steam system.

Chapters 6 to 9 provide an introduction to compressed air systems and include auxiliary systems, theory of compression and compressor controls. This is followed by Chapter 10 which describes common energy saving measures for compressed air systems and Chapter 11 which outlines the main aspects of how to conduct an energy audit of a compressed air system.

Chapter 12 provides an introduction to waste heat recovery while, Chapter 13 covers heat exchangers and various heat transfer concepts associated with them. Chapter 14 describes some of the commercially available heat recovery systems followed by Chapter 15, which provides an insight into the important aspects of an energy audit of a heat recovery system.

I would like to take this opportunity to thank all those who have assisted in the preparation of this reference manual by providing technical information, images and case studies.

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1.0 STEAM PROPERTIES

Water can exist as liquid, solid or gas. Steam is the gaseous phase of water. It is produced by heating water in a boiler at constant pressure. The heat added to produce steam can later be extracted from it by condensing it back to water. This enables steam to be used for carrying large amounts of heat energy. In addition, steam is not toxic, is easily transportable, can be generated relatively efficiently and is not very costly to generate. Therefore, steam is the most common heating medium in industrial facilities. Steam is also used for power generation and in various chemical reactions and processes.

This chapter provides an introduction to some of the basic properties of steam, which are useful in understanding how boilers and steam systems operate.

Learning Outcomes:

The main learning outcomes from this chapter are to understand:

1. The relationship between saturation temperature and pressure
2. Various properties of steam
3. How to use steam tables

1.1 Introduction

Water (H_2O) is abundant on earth. Like many substances, it can exist in three physical states, which are solid, liquid and gas. For H_2O, the three states are called ice, water and steam respectively.

The molecular arrangement and the degree of excitation of the molecules determine the physical state of H_2O. When in the state of ice, the molecules are locked together and can only vibrate about a mean bonded position. If heat is added, the vibration increases and when the heat added reaches a certain point, some of the molecules break away from their bonds. At this stage, the solid starts to melt to a liquid state. At atmospheric pressure, melting occurs at 0°C.

In the liquid phase, the molecules are free to move, but are still close to each other due to mutual attraction. When heat is added, molecular agitation and collisions increase. The temperature of the liquid also rises until it reaches the boiling temperature when some molecules attain sufficient kinetic energy to allow them to escape from the liquid. However, this is momentary and they fall back into the liquid.

Further addition of heat causes the excitation to increase so that some molecules will have sufficient energy to leave the liquid. When this happens, bubbles of steam will rise and break through the surface. When the number of molecules leaving the liquid surface is more than those re-entering, the water is freely vapourising into steam.

1.2 Saturation temperature

When water is heated to its boiling point or its saturation temperature, it is saturated with heat energy. If more heat is added at this stage while maintaining the pressure constant, the temperature of the water will not rise but it will result in the water forming saturated steam. The temperature of both the boiling water and saturated steam will be the same, but the heat energy (per unit mass) will be much greater in the steam than the boiling water.

At atmospheric pressure, the boiling point or the saturation temperature of water is 100°C. However, if the pressure is increased, the saturation temperature will increase (water temperature will need to be increased beyond 100°C for it to boil). Similarly, if the pressure is reduced below atmospheric pressure, water will boil at a temperature less than 100°C. The relationship between the saturation temperature and the pressure is known as the steam saturation curve and is shown in Figure 1.1. Water and steam can coexist at any pressure on this curve, and both will be at the saturation temperature.

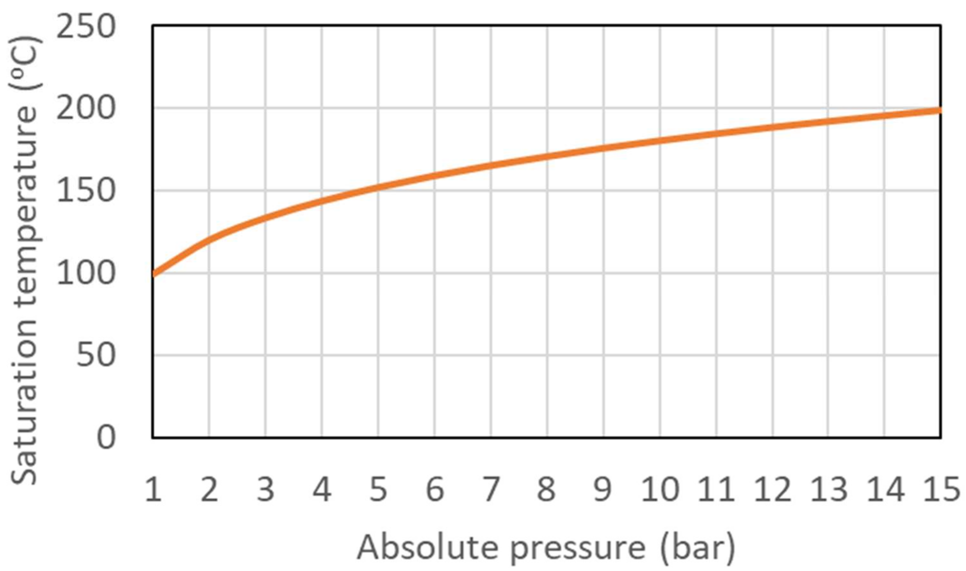

Figure 1.1 Steam saturation curve

The values of saturation temperature at different pressures are shown in Table 1.1.

Absolute pressure (bar)	**Saturation Temperature (°C)**
1	99.6
1.01 (atmospheric pressure)	100
2	120.2
3	133.5
4	143.6
5	151.8
6	158.8
7	165.0
8	170.4
9	175.4
10	179.9
11	184.1
12	188.0
13	191.6
14	195.0
15	198.3

Table 1.1 Saturation temperature of steam at different pressures

Example 1.1

What is the saturation temperature when the pressure is 2.5 bar?

Solution

From Figure 1.2, the saturation temperature at 2.5 bar pressure is approximately 125^{o}C.

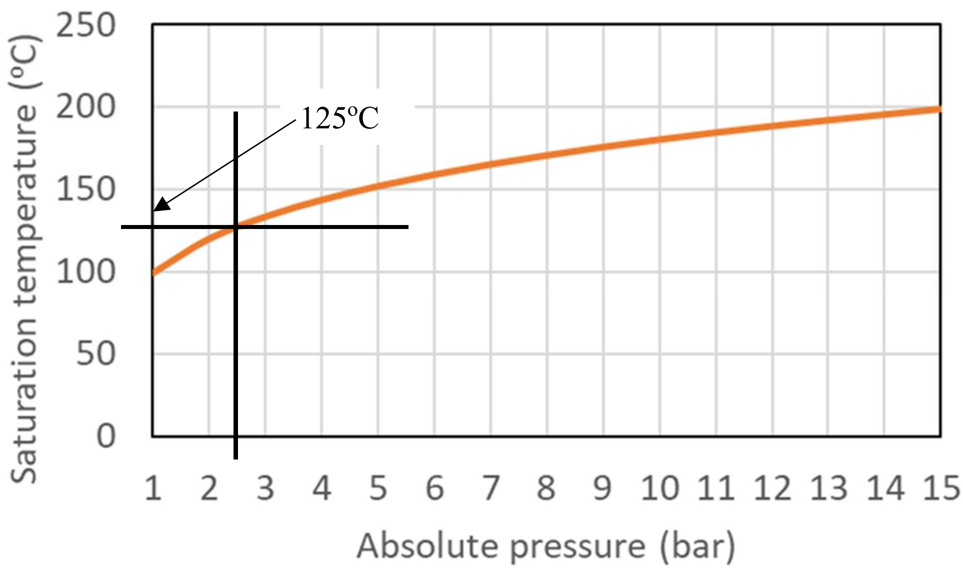

Figure 1.2 Steam temperature for Example 1.1

Example 1.2

A heating process requires steam at 200ºC. What is the minimum required steam pressure for this application?

Solution

From Figure 1.3, a steam pressure of about 15 bar is required to achieve a saturation temperature of 200ºC.

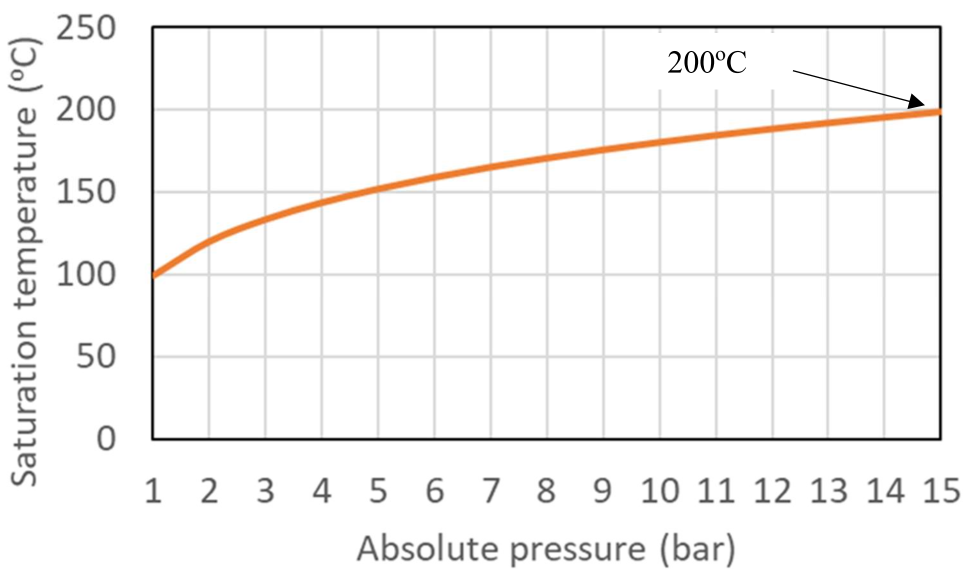

Figure 1.3 Steam temperature for Example 1.2

1.3 Sensible and latent heat

When water is heated by adding heat energy, the temperature of the water rises. Such a process where adding heat leads to a corresponding increase in temperature is called sensible heating and the heat added is called sensible heat.

Specific heat capacity is the amount of heat energy required to raise the temperature per unit of mass of substance. In SI units, specific heat capacity is the amount of heat in joules required to raise 1 kilogramme of a substance by 1 Kelvin. For water, the specific heat capacity at atmospheric pressure is 4.19 kJ/kg.K.

For water, the amount of sensible heat required to increase the temperature of 1 kg of water from 0^{o}C to the boiling temperature is 419 kJ/kg (4.19 kJ/kg.K x 100 K). It is also called the "liquid enthalpy" or enthalpy of water (explained later in this chapter).

When water is heated to its boiling point, further adding of heat does not increase the temperature of water but only results in boiling of the water to form steam. Such heating which does not result in an increase in the temperature of the substance that is heated, but only results in a change in phase, is called latent heating. The heat added during such a process is called latent heat.

Latent heat is energy absorbed during evaporation of a liquid (or released during condensing of a vapour) that occurs without changing its temperature. The latent heat in SI units is expressed in joules per unit mass in grams of the substance undergoing a change of state. For water, the amount of heat required to evaporate 1 kg of water at its boiling point is termed the "enthalpy of evaporation". At atmospheric pressure, the enthalpy of evaporation of water is 2257 kJ/kg.

The liquid enthalpy and enthalpy of evaporation of water at different pressures is shown in Table 1.2.

Absolute pressure (bar)	**Enthalpy of water (kJ/kg)**	**Enthalpy of evaporation (kJ/kg)**
1	417	2258
1.01 (atmospheric pressure)	419	2257
2	505	2202
3	561	2164
4	605	2134
5	640	2109
6	670	2087
7	697	2067
8	721	2048
9	743	2031
10	763	2015
11	781	2000
12	798	1986
13	815	1972
14	830	1960
15	845	1947

Table 1.2 Liquid enthalpy and enthalpy of evaporation of water

Example 1.3

At atmospheric pressure, how much sensible heat is required to raise 1000 kg of water from 30^{o}C to 70^{o}C?

Solution

Specific heat capacity of water atmospheric pressure is 4.19 kJ/kg.K

Therefore, the sensible heat required to raise the temperature of 1000 kg is,

1000 kg x 4.19 kJ/kg.K x (70 – 30) K = 167,600 kJ

Example 1.4

At a pressure of 5 bar, how much latent heat is required to boil 1000 kg of water at its saturation temperature?

Solution

Enthalpy of evaporation at 5 bar pressure (from Table 1.2) is 2109 kJ/kg

Therefore, the total heat required to boil 1000 kg of water at 5 bar is,

1000 kg x 2109 kJ/kg = 2,109,000 kJ

1.4 Steam quality

In an industrial type boiler, where heat is supplied only to the water, it is not possible to produce dry steam. Due to turbulence and splashing in the boiler when bubbles of steam are released from the water surface, the steam contains some water droplets. Typically, steam produced by a shell-type boiler will contain about 5% water.

If the water content of the steam is say 4% by mass, then the steam is said to be 96% dry.

Steam quality or dryness fraction of steam is defined as the ratio of the mass of actual dry steam to the total mass of wet steam and can be expressed as:

$$x = m_g / m_g + m_f \qquad (1.1)$$

$$x = m_g / m$$

where,

x = steam quality (dryness fraction)

m_g = mass of dry steam

m_f = mass of water in mixture

m = mass of wet steam = $m_g + m_f$

Example 1.5

1000 kg of steam contains 50 kg of water. Compute the steam quality.

Solution

Steam quality, $x = m_g / m_g + m_f$

$$= (1000 – 50) / 1000$$

$$= 0.95$$

1.5 Superheated steam

When saturated steam is further heated so that the temperature of the steam exceeds the saturation temperature at a particular pressure, the steam is superheated. Superheating produces steam that has a higher temperature and lower density than saturated steam at the same pressure.

For instance, steam at 5 bar is at a saturation temperature of 159°C. Now if this steam is heated to 179°C, the steam will have a superheat of 20°C at 5 bar.

Superheating steam ensures that the steam is completely dry. Superheating is used in steam driven equipment such as turbines to avoid a drop in performance due to the presence of condensate and to prevent erosion and corrosion. However, superheated steam is normally not used in pure heating applications where the latent heat of condensation is the primary source of heating. When using superheated steam, first it needs to be cooled to its saturation temperature before it can condense. As a result, a larger heat transfer area will be required when using superheated steam compared to saturated steam for the same application.

Example 1.6

Steam at 10 bar is at a temperature of 210°C. What is the degree of superheat of the steam?

Solution

Saturation temperature of steam at 10 bar (from Table 1.1) is approximately 180°C.

Therefore, the degree of superheat is (210 – 180)°C = 30°C

1.6 Steam pressure vs volume

The specific volume is the total volume of steam divided by the total mass of steam (volume per unit mass). It has units of cubic metre per kilogramme (m^3/kg). The density of steam is the reciprocal of its specific volume.

$\rho = m/V = 1/v$

where,

ρ = density (kg/m^3)

m = mass of steam (kg)

V = volume of steam (m^3)

v = specific volume (m^3/kg)

As steam pressure increases, the density of steam also increases. Since the specific volume is inversely related to the density, the specific volume decreases with increased pressure.

Figure 1.4 shows the relationship between specific volume and pressure. As can be seen from the figure, the greatest change in specific volume occurs at lower pressures. At higher pressures, the change in specific volume is much less.

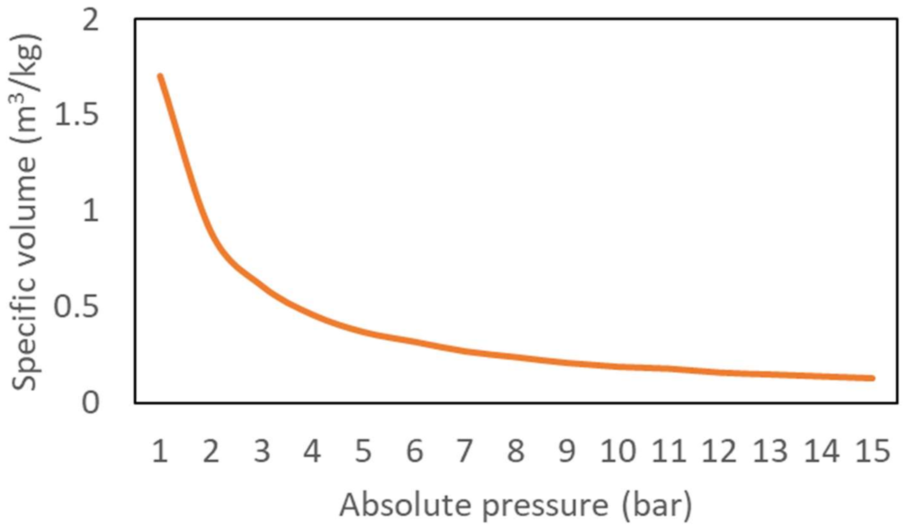

Figure 1.4 Steam pressure vs specific volume

Because the specific volume of water is several orders of magnitude lower than that of steam, the droplets of water in wet steam will occupy negligible space. Therefore, the specific volume of wet steam is less than that of dry steam.

Specific volume of wet steam = v_g x (1.2)

where,

v_g = specific volume of dry steam

x = steam quality

Table 1.3 shows values of specific volume of dry steam at different pressures. Therefore, if the specific volume of wet steam is known, the dryness fraction can be computed as shown in Example 1.7.

Absolute pressure (bar)	Specific volume of dry steam (m^3/kg)
1	1.7
2	0.89
3	0.61
4	0.46
5	0.37
6	0.32
7	0.27
8	0.24
9	0.21
10	0.19
11	0.18
12	0.16
13	0.15
14	0.14
15	0.13

Table 1.3 Specific volume of dry steam

Example 1.7

Steam is at 8 bar (absolute) pressure. The specific volume of the steam is 0.17 m^3/kg. Compute the "quality" of the steam.

Solution

From Table 1.3, the specific volume of dry steam at 8 bar is 0.24 m^3/kg.

From equation (1.2),

Specific volume of wet steam = v_g x

Therefore, steam quality, x = specific volume of wet steam / v_g

= 0.17 / 0.24

= 0.71

1.7 Enthalpy

Specific enthalpy is a measure of the energy content of a single unit of mass of a substance. The SI units of specific enthalpy are kJ/kg. For simplicity, specific enthalpy will be referred to as enthalpy in this reference manual.

Figure 1.5 shows the relationship between temperature and enthalpy for water when it is heated at a constant pressure to form steam.

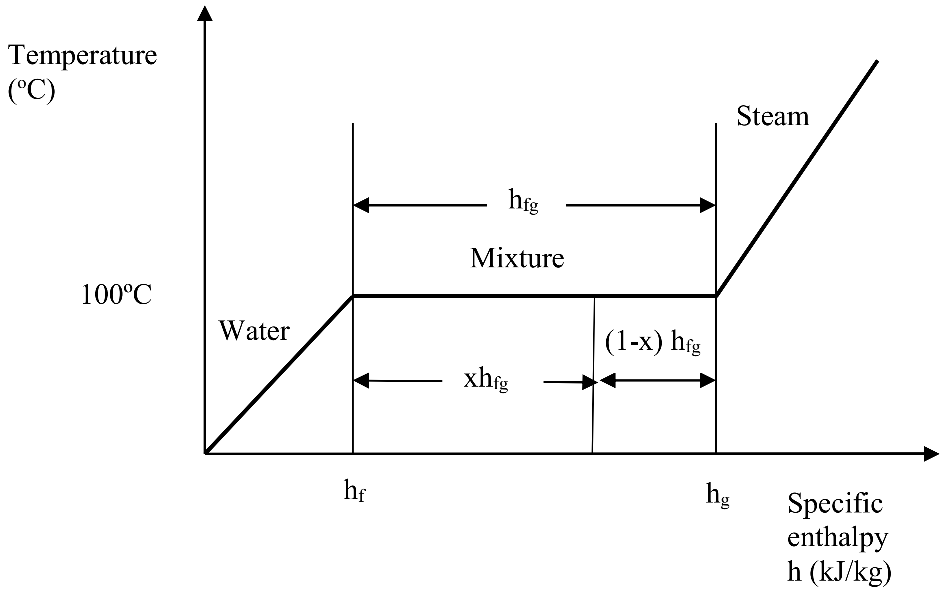

Figure 1.5 Relationship between temperature and enthalpy

As indicated in Figure 1.5, when water is heated, its temperature rises steadily until it reaches the saturation point where water cannot exist in the liquid form. At this point, addition of heat results in the boiling of water, producing steam. Further heating results in more steam being generated at the same temperature (saturation temperature). When all the water has been converted into steam, further addition of heat leads to an increase in steam temperature above the saturation temperature and the steam is then superheated.

Enthalpy value h_f refers to the enthalpy of water when it has reached the saturation temperature while h_g refers to the enthalpy of steam. The difference between h_g and h_f is the latent heat of vapourisation which is denoted by the symbol h_{fg}. Since saturated steam contains water, the value x refers to steam quality.

Therefore,

Enthalpy of wet steam $h_g = h_f + x\ h_{fg}$ (1.3)

Enthalpy of dry steam $h_g = h_f + h_{fg}$ (1.4)

Some values of h_f, h_g and h_{fg} at different pressures are shown in Table 1.4.

Absolute pressure (bar)	**Enthalpy of water, h_f (kJ/kg)**	**Enthalpy of evaporation, h_{fg} (kJ/kg)**	**Enthalpy of steam, h_g (kJ/kg)**
1	417	2258	2675
1.01 (atmospheric pressure)	419	2257	2676
2	505	2202	2707
3	561	2164	2725
4	605	2134	2739
5	640	2109	2749
6	670	2087	2757
7	697	2067	2764
8	721	2048	2769
9	743	2031	2774
10	763	2015	2778
11	781	2000	2781
12	798	1986	2784
13	815	1972	2787
14	830	1960	2790
15	845	1947	2792

Table 1.4 Enthalpy values for water and steam

For superheated steam, $h = h_f + h_{fg} + Cp\ (t_{sup} - t_{sat})$ kJ/kg

where,

t_{sup} = superheated temperature of steam (K)

t_{sat} = saturation temperature of steam (K)

$(t_{sup} - t_{sat})$ = degree of superheat (K)

Cp = Specific heat capacity of steam (kJ/kg.K)

Example 1.8

A boiler is supplied with feedwater at a temperature of 70ºC. The boiler produces steam at a pressure of 8 bar (abs.) and a temperature of 190ºC.

Determine the quantity of heat supplied per kg of steam generation (excluding losses). Take the specific heat capacity (Cp) of superheated steam to be 2.76 kJ/kg.K.

Solution

From Tables 1.1 and 1.4, the following enthalpy values at 8 bar pressure can be obtained

h_f = 721 kJ/kg
h_{fg} = 2048 kJ/kg
t_{sat} = 170.4ºC

Since steam is produced at 190ºC, the steam is superheated.

The enthalpy of superheated steam,
$h_{sup} = h_f + h_{fg} + Cp\ (t_{sup} – t_{sat})$

$= 721 + 2048 + 2.76\ (190 – 170.4)$
$= 2823.1$ kJ/kg

Enthalpy of feedwater at 70ºC is 293 kJ/kg (70ºC x 4.19 kJ/kg)

Therefore, heat added = (2823.1 – 293) kJ/kg = 2530.1 kJ/kg

1.8 Steam pressure vs enthalpy of evaporation

Figure 1.5 showed how the enthalpy changes when water is heated to produce steam. However, the enthalpy of evaporation changes with pressure. As shown in Figure 1.6, when the pressure increases, the enthalpy of evaporation (h_{fg}) reduces. Therefore, at higher operating pressures, more steam is required to produce the same amount of heating as the latent heat available becomes less.

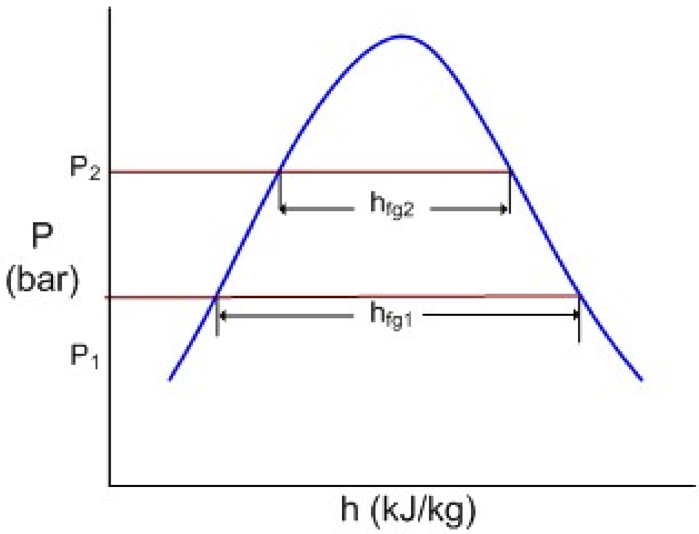

Figure 1.6 Enthalpy of evaporation at different pressures

1.9 Entropy

Entropy, is a thermodynamic measure of thermal energy per unit temperature of a system that is unavailable for doing useful mechanical work. Since work is obtained from orderly molecular motion, the amount of entropy is also a measure of the molecular disorder, or randomness, of a system.

Specific entropy usually denoted by symbol "s", is the entropy per unit mass of a system. The units of entropy are kJ/K, and for specific entropy are kJ/kg.K.

From Table 1.4, at atmospheric pressure, 2,257 kJ (2,676 – 419) is required to convert one kg of liquid water at 100°C to steam at 100°C. Since temperature remains constant, the change in specific entropy is 2,257 / 373 = 6.059 kJ/kg.K. Hence, by converting one kg of water to steam, the specific entropy has increased by 6.059 kJ/kg.K.

Values of specific entropy are usually provided in steam tables.

1.10 Condensate and flash steam

When high-pressure steam is condensed, the condensate will be at the saturation temperature of the steam. Later, when the condensate is returned to the feedwater tank, the pressure is reduced to atmospheric pressure. When this happens, the enthalpy of the condensate has to reduce from the value at the higher pressure to the value at atmospheric pressure.

For example, condensate at 8 bar pressure (absolute) will have an enthalpy of 721 kJ/kg (from Table 1.4). However, when this condensate pressure is reduced to atmospheric pressure, the enthalpy becomes 419 kJ/kg. The difference between the enthalpy values of 302 kJ/kg (721 – 419) is released by evaporating part of the condensate. The steam formed during such a pressure reduction is called flash steam.

1.11 Use of steam tables

Since water and steam are commonly used for transferring heat energy, their properties are required for various calculations. Therefore, their properties are tabulated in so called "Steam Tables". In these tables, important properties, such as pressure, temperature, enthalpy and specific volume are tabulated. An abridged steam table is provided in Table 1.5.

Absolute pressure (bar)	Saturation temperature (°C)	Specific volume of dry steam (m^3/kg)	Enthalpy of water, h_f (kJ/kg)	Enthalpy of evaporation, h_{fg} (kJ/kg)	Enthalpy of steam, h_g (kJ/kg)
1	99.6	1.7	417	2258	2675
1.01 (atmospheric pressure)	100	1.67	419	2257	2676
2	120.2	0.89	505	2202	2707
3	133.5	0.61	561	2164	2725
4	143.6	0.46	605	2134	2739
5	151.8	0.37	640	2109	2749
6	158.8	0.32	670	2087	2757
7	165.0	0.27	697	2067	2764
8	170.4	0.24	721	2048	2769
9	175.4	0.21	743	2031	2774
10	179.9	0.19	763	2015	2778
11	184.1	0.18	781	2000	2781
12	188.0	0.16	798	1986	2784
13	191.6	0.15	815	1972	2787
14	195.0	0.14	830	1960	2790
15	198.3	0.13	845	1947	2792

Table 1.5 Abridged steam table

Example 1.9

Determine the quantity of heat required to produce 1 kg of steam at pressure of 7 bar (absolute) using water at a temperature of 90ºC, under the following conditions:

a) When the steam is wet and having a dryness fraction of 0.8
b) When the steam is dry saturated
c) When the steam is superheated at a constant pressure to 240ºC (assume the specific heat capacity of superheated steam to be 3.68 kJ/kg K)

Solution

From Table 1.5, at 7 bar

h_f = 697 kJ/kg

h_{fg} = 2067 kJ/kg

t_{sat} = 165ºC

a) When the steam is wet

$h = h_f + x.\ h_{fg}$

$h = 697 + 0.8 \times 2067$

h = 2350 kJ/kg

Enthalpy of water at 90ºC = 377 kJ/kg

Therefore, actual heat required

$h = 2350 – 377$

h = 1973 kJ/kg

b) When the steam is dry saturated

$h = h_f + h_{fg}$

$h = 697 + 2067$

h = 2764 kJ/kg

h = 2764 – 377 = 2387 kJ/kg

c) When the steam is superheated

$h = hf + h_{fg} + Cp\ (t_{sup} – t_{sat})$

$h = 697 + 2067 + 3.68\ (240 - 165)$

h = 3040 kJ/kg

h = 3040 – 377 = 2663 kJ/kg

Often, the known parameter is between two values listed in the steam table. In such instances, the required steam property can be derived by linear interpolation as illustrated in Example 1.10.

Example 1.10

Determine the enthalpy of water at 6.3 bar (absolute pressure).

Solution

From Table 1.5, at 6 bar, h_f = 670 kJ/kg and at 7 bar, h_f = 697 kJ/kg

Therefore, using linear interpolation,

$$\frac{(h_{f\,@\,6.3\,bar} - h_{f\,@\,6\,bar})}{(h_{f\,@\,7\,bar} - h_{f\,@\,6\,bar})} = \frac{(6.3 - 6)}{(7 - 6)}$$

From the above relationship, $h_{f@6.3\ bar}$ = 678.1 kJ/kg

Summary

This chapter provided an introduction to some of the basic properties of steam, which are useful in understanding how boilers and steam systems operate. The relationship between saturation temperature and pressure, various properties of steam and the use of steam tables were illustrated using worked examples.

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2.0 BOILERS

Steam is vapour of water and is used in many industrial applications as the working fluid for heating and as the carrier of energy. Steam is widely used in industry because usually there is easy access to water required for steam generation, the ability to convert water to steam using boilers, the high heat carrying capacity of steam and the simplicity in transferring and distributing heat energy through a piping network.

In a typical boiler system (Figure 2.1), feedwater is provided at high pressure from a feedwater pump to the boiler. The feedwater is usually a mixture of condensed steam and make-up water. Fuel and air are provided to the boiler for combustion and the heat produced vapourises the water to generate steam. After combustion, the exhaust gases are released to the environment. The generated steam is supplied to the various end users while condensed steam is recovered and returned back to the boiler.

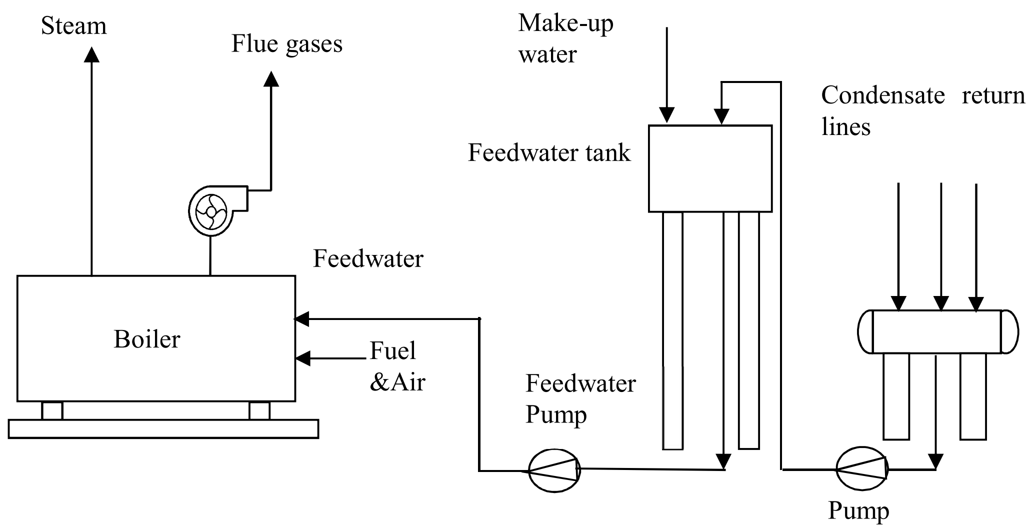

Figure 2.1 A typical boiler system

This chapter provides an introduction to steam boilers, how they operate, their main characteristics and the various methods of measuring boiler efficiency.

Learning Outcomes:

The main learning outcomes from this chapter are to understand:

1. The different types of boilers
2. Key characteristics of boilers

3. Parameters that affect boiler performance
4. How to assess boiler efficiency

2.1 Introduction to boilers

Boilers are used to produce steam by adding heat to water. A boiler generally consists of a combustion chamber that can burn fuel in the form of solid, liquid or gas to produce hot combustion gases, and a tubular heat exchanger, to transfer heat from the combustion gases to the water. They are pressure vessels into which liquid water is pumped at the operating pressure. After the heat energy from the fuel source has vapourised the liquid water, the resulting steam is directly provided for use or is passed through a superheater.

Fuel used by boilers can be in the form of solid, liquid or gas. Further, the fuels can be primary fuels that are naturally available or secondary fuels which are artificial. Some of the common types of fuels used are listed in Table 2.1.

Solid / liquid /gas	Primary fuel	Secondary fuels
Solid	Coal Peat Wood / Biomass	Coke Charcoal
Liquid	Petroleum	Diesel Fuel oil Coal tar Ethanol
Gas	Natural gas	Propane Methane Biogas

Table 2.1 Types of fuel used for boilers

In addition to the above, boilers can also be fired using waste heat from various processes. In such cases, waste heat can be channelled to heat recovery boilers to generate steam. Examples include exhaust from gas turbine power generators and petroleum refineries.

The calorific value of a fuel is the quantity of heat produced by its combustion, at "normal" conditions (temperature of 0°C and pressure of 1bar).

Gross Calorific Value assumes that the water produced during combustion is entirely condensed and the heat contained in the water vapour is recovered. Net Calorific Value assumes that the water vapour and the heat in the water vapour is not recovered. Therefore, the gross calorific value is higher than the net calorific value.

Gross calorific value of common fuels used in boilers is provided in Table 2.2.

Fuel	Calorific value (MJ/kg)
Natural gas	55
No. 2 oil (light oil)	46
No. 4 oil (heavy oil)	45
No. 6 oil (heavy oil)	44
Coal	32
Wood (dry)	16

Table 2.2 Calorific value of fuels

As shown in Figure 2.2, the main inflows to a typical boiler are fuel, air and feedwater while the outflows are steam or hot water, exhaust flue gases and blowdown.

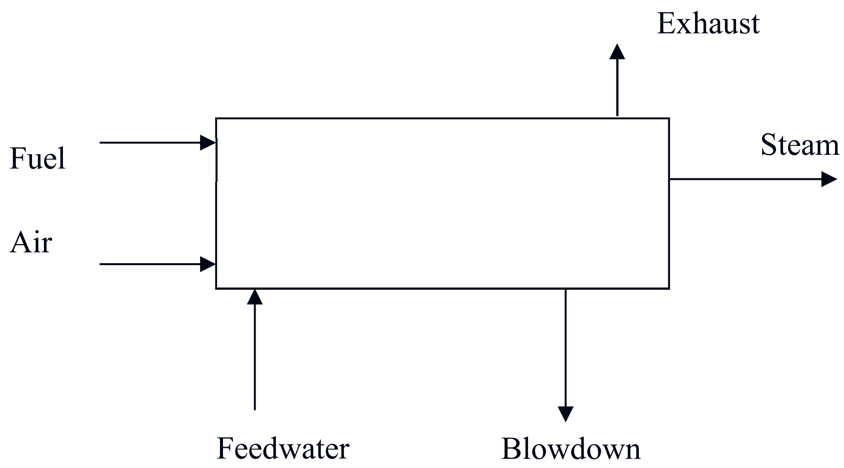

Figure 2.2 Main in-flows and out-flows for a typical boiler

Boilers are normally classified as fire tube or water tube boilers, depending on the flow arrangement of water and the hot gases inside the boiler. In fire tube boilers, the hot gases pass through the boiler tubes that are immersed in the water being heated, while in water tube boilers, water is contained in the tubes that are surrounded by the hot combustion gases.

Fire tube boilers are used for general applications. However, since water is contained in the circular boiler shell in fire tube boilers, increase in pressure requires increase in thickness of the boiler shell, which becomes impractical at high pressures. Therefore, water tube boilers, where water is circulated inside tubes (the relatively smaller diameter of the tubes compared to the shell diameter in fire tube boilers require less wall thickness), are used for high pressure applications. Fire tube boilers are usually used for producing steam for process heating and other low pressure applications, up to about 15 bar. For high pressure applications like in power plants where steam pressure could be in the range of 150 bar, water tube boilers are used.

To increase the surface area available for heat transfer between the combustion gases and water, the tubes in boilers are arranged to have a number of passes so that the hot flue gases can pass through a number of sets of tubes before being exhausted. A typical arrangement of a four-pass fire tube boiler is shown in Figure 2.3.

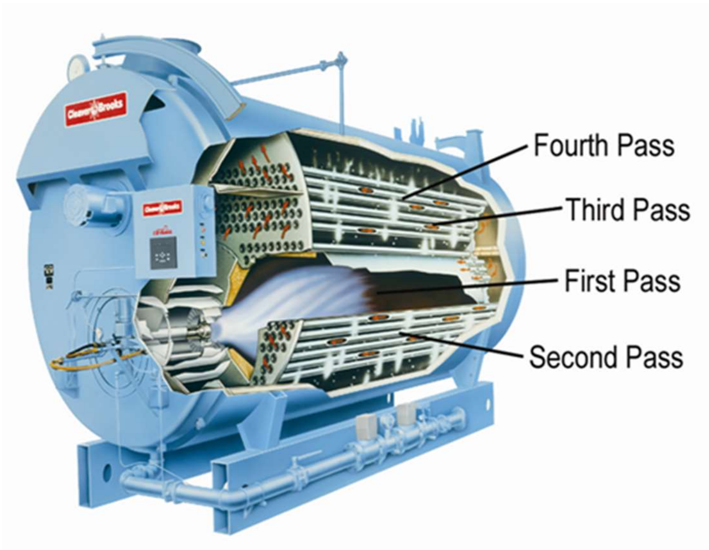

Figure 2.3 Arrangement of a four-pass fire tube boiler (courtesy of Cleaver-Brooks)

A cut-away of a commercial water tube boiler is shown in Figure 2.4 while the water circulation arrangement is shown in Figure 2.5. Feedwater enters the steam drum and moves down through the downcomer to the mud drum due to its higher density. This causes the warmer water to rise through the front tubes to the steam drum. The heat added causes some of the water to boil and the steam bubbles are separated and removed at the steam drum.

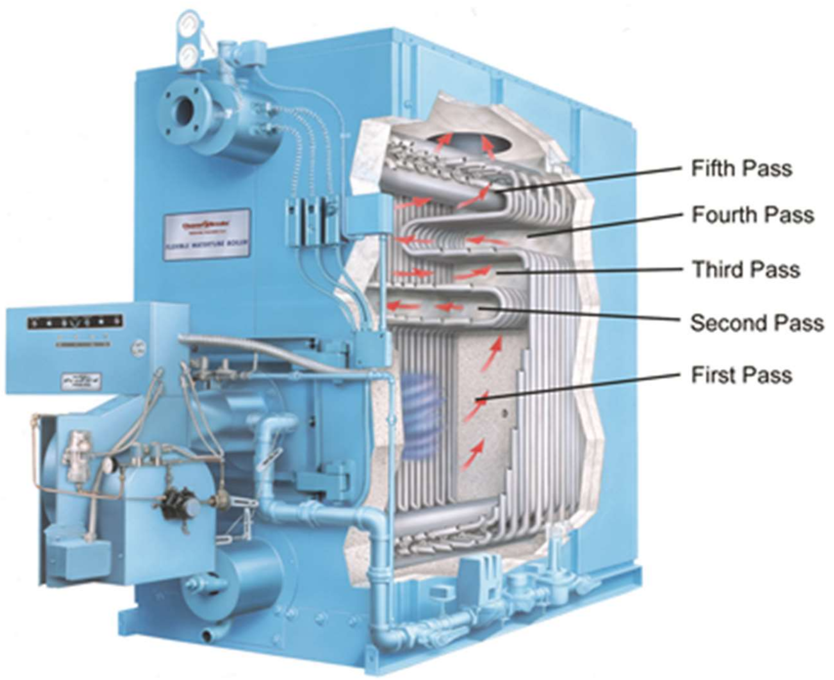

Figure 2.4 Cut-away of a commercial water tube boiler (courtesy of Cleaver-Brooks)

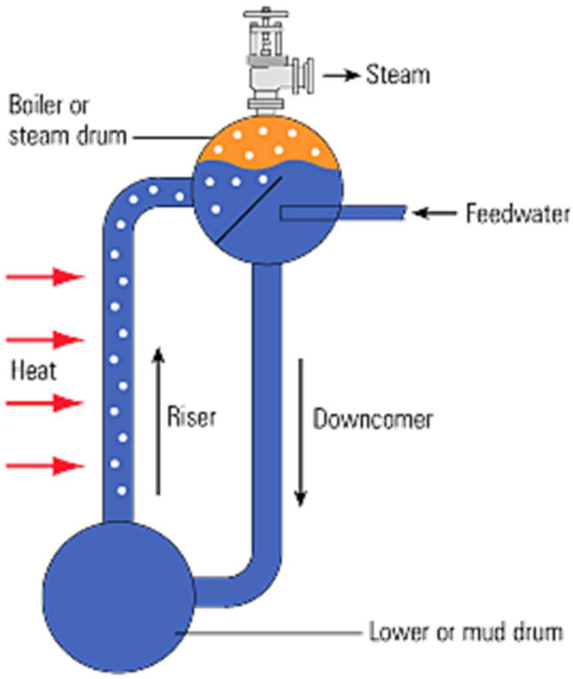

Figure 2.5 Water circulation in a water tube boiler (illustration taken from the Spirax Sarco website 'Steam Engineering Tutorials' at http://www.spiraxsarco.com)

Boiler capacity depends on the steam generation rate and steam pressure. The steam generation rate is usually rated in kg/hr, lb/hr or Tons/hr while the steam pressure is rated in bar or psi (pounds per square inch).

2.2 Main components of a boiler

In addition to the main combustion chamber and tubes used for converting liquid water to steam, a boiler comprises a number of other important components that are described below.

Combustion system

Various types of burners are used in liquid and gas fired boilers. The burners can be designed only for gaseous fuel, liquid fuel or dual fuel use. In dual fuel burners, the fuel type can be changed from gas, which is usually the primary source, to liquid, by closing the gas lines and opening the oil valves and re-firing the boiler. Such a switch-over can usually be carried out quickly so that there is no disruption to the steam supply.

In solid fuel boilers, various fuel feed arrangements with conveyors and hoppers are used. Some boilers use fixed grates while larger boilers use moving, vibrating and reciprocating grates. Newer technologies available for solid fuel combustion include fluidised beds.

Superheater

A superheater is often used in a boiler to convert wet steam or saturated steam into dry steam. Superheating is achieved by further heating the wet steam to ensure the steam is dry and to increase the temperature of the steam. Superheated steam is required by equipment such as steam turbines or when steam has to be transported over long distances.

There are three main types of superheaters. They are:

1. Radiant type
2. Convection type
3. Separately fired

Radiant type superheaters are placed directly in the combustion chamber of boiler while convection type superheaters are placed in the path of the combustion gases as illustrated in Figure 2.6. Separately fired superheaters, when used, are placed external to the boiler and consists of a separate combustion chamber with the steam coil used for superheating passing through it.

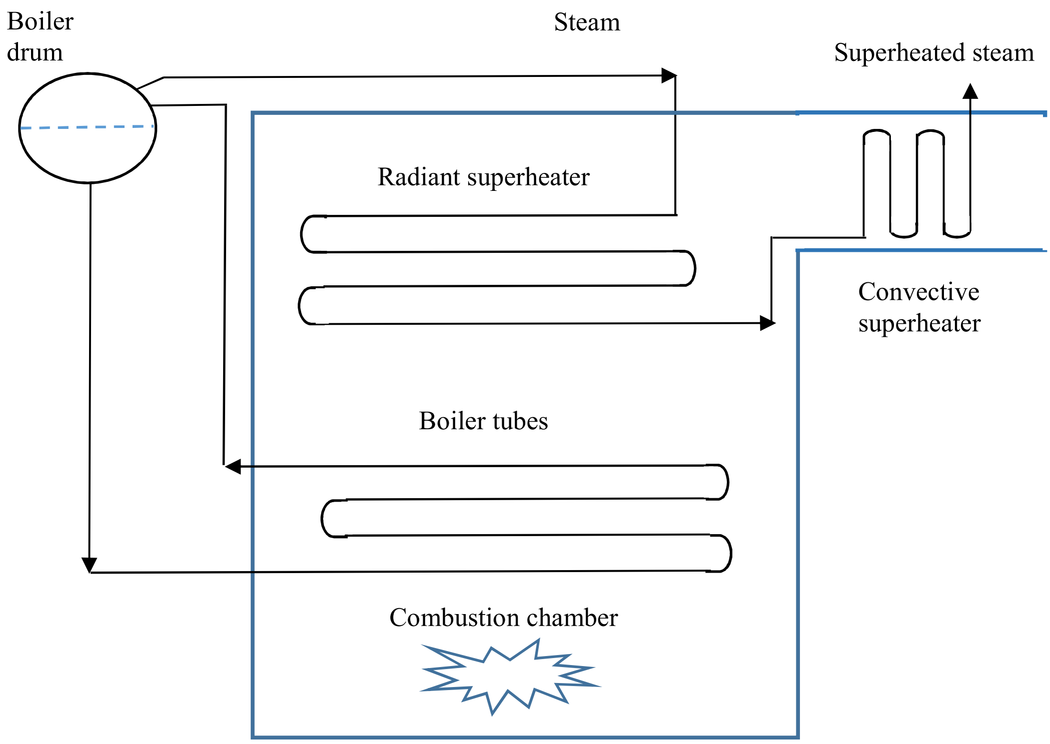

Figure 2.6 Arrangement of radiant and convective superheaters

Pumps

Pumps are used to provide water to boilers. They are usually called feedwater pumps. Level controllers are used in boilers to maintain the water level within a set range (maximum and minimum values). Feedwater pumps take water from the feedwater tank, deaerator or condensate water tank and raise the water pressure to the pressure inside the boiler. Feedwater pumps usually operate at constant speed, and the water level in the boiler is controlled by switching the feedwater pump on and off.

Feedwater is usually a mixture of condensate recovered by condensing back steam after various process heating applications (e.g. hot water generation, pasteurisation, jacket heating), and fresh water added to make up for losses.

Fans

In a boiler, air (oxygen) needs to be provided to enable combustion of the fuel. Air can be supplied by natural convection or mechanically through the use of fans.

In boilers relying on natural convection, when the hot flue gases rise through the chimney due to their lower density and are exhausted, surrounding air that has a higher density enters the combustion chamber. This creates a natural draught and provides the air required for combustion.

Since the natural draft created depends on factors such as the temperature of the outside air, temperature of the flue gases and the height of the chimney, it is difficult to control combustion in natural draught boilers.

Modern boilers rely on mechanical draught to provide air for combustion. In such boilers, air for combustion is provided using a fan. There are three main arrangements of mechanical draught, which are:

1. Induced draught where a fan is installed at the exit of the boiler to pull out the exhaust gases;
2. Forced draught where a fan is installed at the air intake to push in the air for combustion; and
3. Balanced draught where both induced draught and forced draught fans are used.

Balanced draught arrangements are common in large boilers where the combustion gases have to travel a long distance and over a number of heat exchanger passes within the boiler, resulting in high pressure losses.

The airflow in mechanical draught systems is controlled using damper arrangements and by adjusting the fan speed.

Deaerator

A deaerator is a device used external to the boiler to remove oxygen and other dissolved gases from the feedwater. If a deaerator is not used, oxygen and other dissolved gases can cause corrosion or form oxides on the boiler surfaces in contact with water, leading to damage and component failure.

The two common types of deaerators are spray type and tray type. In a spray type deaerator shown in Figure 2.7, feedwater is sprayed into the chamber with a steam sparge coil submerged in feedwater. The feedwater is heated to the saturation temperature by the steam. The gases removed from the water rise to the top and are released using the air vent.

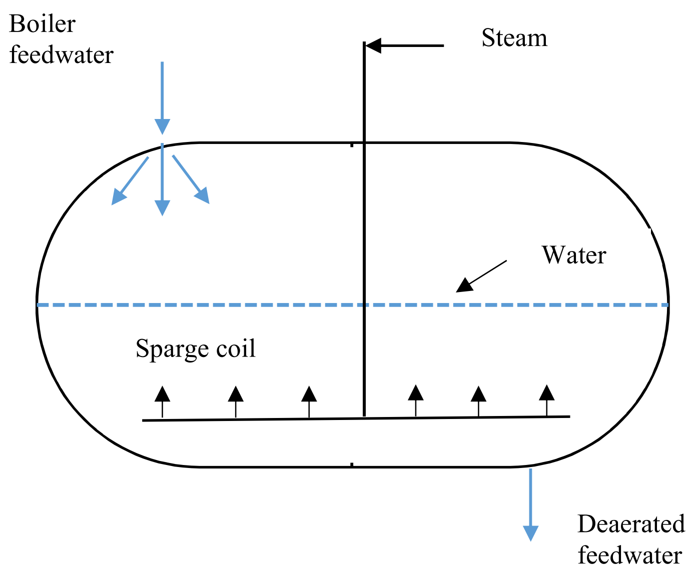

Figure 2.7 Arrangement of a spray type deaerator

The other common type of deaerator is the tray type shown in Figure 2.8. the operating principle is similar to the spray type except that an additional tray arrangement is used to provide a longer contact time between the feedwater and steam.

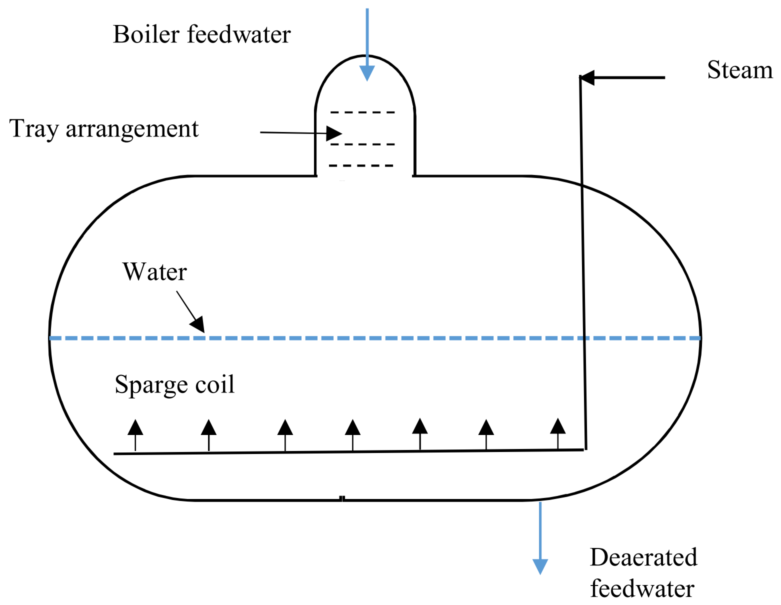

Figure 2.8 Arrangement of a tray type deaerator

2.3 Boiler operation

Some of the key operating requirements of boilers are listed below.

Air to fuel ratio

Air is required for combustion of the fuel. If the quantity of air that is supplied is greater than what is required for complete combustion of fuel, then the excess air provided will remove part of the heat from the combustion chamber which otherwise would have contributed to producing steam.

If insufficient air is provided, then some of the fuel will be unused in the combustion process. This results in energy wastage as part of the fuel is discharged without being used. In addition, it can lead to unsafe operation as the unburnt fuel can ignite while passing through the boiler. Unburnt fuel also leads to soot formation on heat transfer surfaces leading to poor heat transfer.

Therefore, the air to fuel ratio has to be optimum for the efficient and safe operation of a boiler. A suitable control system has to be used to ensure that the air to fuel ratio is maintained to provide sufficient air for combustion while not being significantly excessive, to prevent unnecessary heat loss from the boiler. Recommended excess air values for common fuels are provided in Table 2.4.

Blowdown

Makeup water used for boilers contains various impurities. As water is converted to steam, the concentration of the impurities that remain in the boiler increases. If this concentration is allowed to increase, it will lead to accelerated corrosion, scaling, and fouling of the heat transfer surfaces of the boiler. Therefore, it is necessary to remove part of the concentrated water from the boiler and replace it with fresh water.

Boiler blowdown is carried out as part of the water treatment process and involves removal of sludge and solids from the boiler to maintain the concentration of solids within an acceptable range.

The blowdown rate is normally determined based on the total dissolved solids (TDS) in the boiler water. The acceptable TDS value to be maintained depends on many factors like boiler design, boiler capacity, water level and load characteristics. For typical two-pass and three-pass fire tube boilers, the allowable TDS level is 3000 to

3500 ppm (parts per million) while for water tube boilers, the allowable level is about 1500 ppm.

Blowdown can be manual where a fixed volume of water is drained from the boiler periodically, or continuous where a small amount of water is removed continuously in order to maintain the quality of water within acceptable limits.

Water softening

Make-up water used for boilers usually contains minerals such as calcium and magnesium which can cause scaling and damage to the boiler. Scale deposits form a layer on the water side of the boiler heat transfer surfaces and act as a resistance to heat transfer which reduces boiler efficiency.

A water softening plant removes positively charged ions like magnesium, calcium and iron using negatively charged resin beads. As the water goes through the resin tank, the minerals are chemically attracted to the negative charge of the resin beads. This process removes the majority of “hard” minerals from the water, creating so-called “soft” water.

2.4 Boiler efficiency

A typical heat balance for a boiler is shown in Figure 2.9. As shown in the figure, only part of the heat content of the fuel is converted into useful heat while the rest is lost through exhaust gases, blowdown and radiation losses. The efficiency of boilers is usually rated based on combustion efficiency, thermal efficiency and overall efficiency.

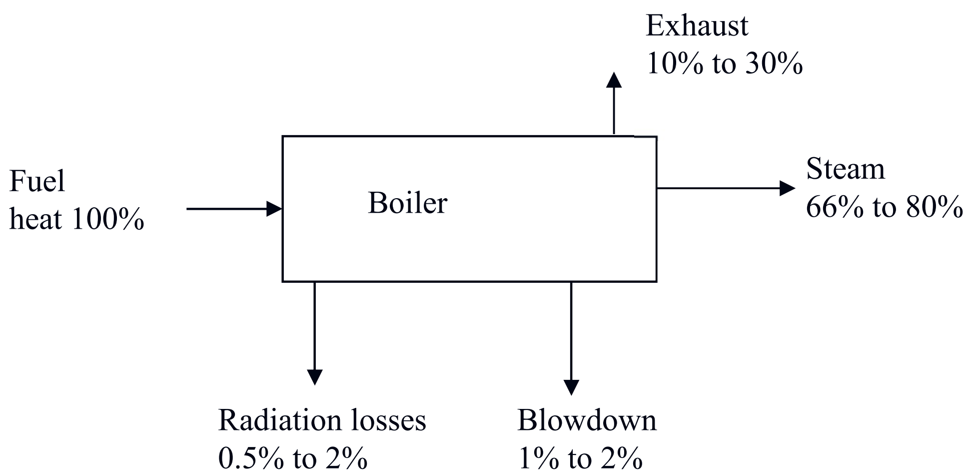

Figure 2.9 Typical heat balance for a boiler

Combustion efficiency

One of the most common measures of boiler efficiency is combustion efficiency, which indicates the ability of the combustion process to burn the fuel completely.

A typical combustion process in boilers involves the burning of fuels containing carbon (oil, gas, coal) with oxygen to generate heat. Oxygen required for combustion is normally taken from air supplied to the burner of the boiler. The amount of air needed for combustion depends on the type of fuel used. To ensure complete combustion of fuel, more air than required (excess air) for combustion is provided to ensure that the fuel is completely burnt. Since excess air leads to lower boiler efficiency (due to removal of heat by the excess air as it passes through the boiler), the objective is to ensure that the minimum amount of excess air is provided.

As shown in Figure 2.10, when excess air is increased, the combustion efficiency increases until a point when the heat carried away by the excess air is more than the extra heat generated by the efficient combustion. Stoichiometric combustion is when the oxygen provided is exactly sufficient for complete combustion of fuel. The best combustion efficiency is usually achieved when the excess air is a few percent greater than that required for stoichiometric combustion.

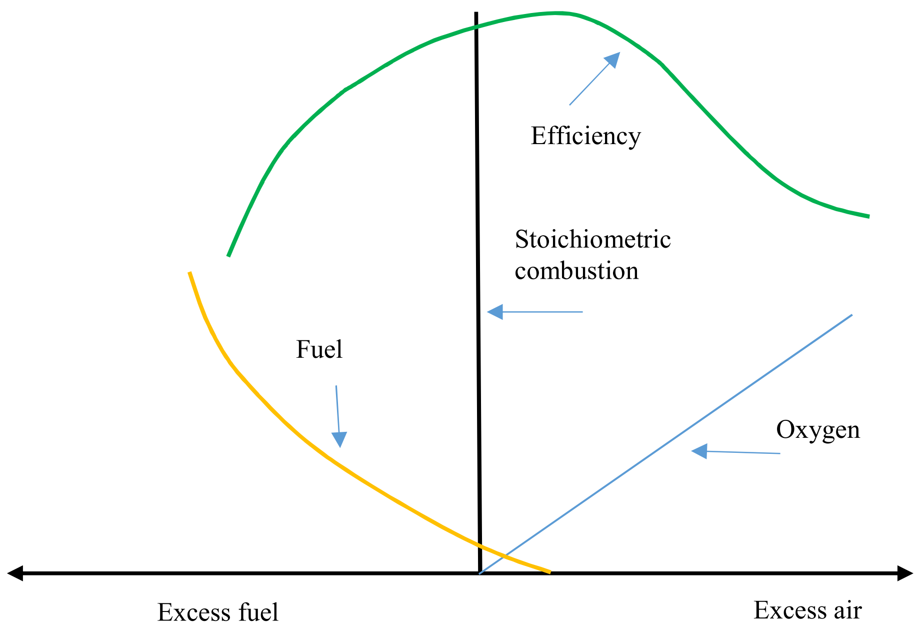

Figure 2.10 Excess air vs combustion efficiency

It is normally measured by sampling the exhaust flue gas to find the composition and temperature using a combustion analyser. Figure 2.11 shows an image of a combustion analyser.

Figure 2.11 A combustion analyser

The combustion efficiency values at different excess air levels and net stack temperature (difference between flue gas temperatures and boiler room air temperature) for natural gas are shown in Table 2.3.

Excess air (%)	Excess oxygen (%)	Net stack temperature (°C)				
		90	150	200	260	315
9.5	2	85.4	83.1	80.8	78.4	76.0
15	3	85.2	82.8	80.4	77.9	75.4
28	5	84.7	82.1	79.5	76.7	74.0
45	7	84.1	81.2	78.2	75.2	72.1
82	10	82.8	79.3	75.6	71.9	68.2

Table 2.3 Combustion efficiency for natural gas

Recommended excess air values for achieving best combustion efficiency are listed in Table 2.4 for some common fuels.

Fuel	Excess air
Natural gas	5 – 10%
Fuel oil	5 – 20%
Coal	15 – 60%

Table 2.4 Recommended excess air values

The combustion efficiency can be expressed as net combustion efficiency or gross combustion efficiency.

Net combustion efficiency assumes that the energy contained in the water vapour that is formed as a product of combustion is recovered and is not exhausted from the flue or stack.

Gross combustion efficiency assumes that the energy contained in the water vapour is not recovered. This value is about 8% lower than the net value.

Thermal efficiency

Thermal efficiency is a measure of the efficiency of the heat exchange in the boiler. It provides an indication of how well the heat exchanger can transfer heat from the combustion process to water or steam in the boiler. It does not take into consideration the conduction and convection losses from the boiler. Therefore, it is not very useful in evaluating the performance of a boiler.

Overall efficiency

Another measure of boiler efficiency is the overall boiler efficiency, which is a measure of how well the boiler can convert the heat input from the combustion process to the steam or hot water. It is also called fuel-to-steam efficiency.

$$\text{Overall boiler efficiency} = \left(\frac{\text{Heat output}}{\text{Heat input}}\right) \times 100 \qquad (2.1)$$

The heat input depends on the amount of fuel burnt and its calorific value (heating value). The calorific value, normally expressed in kJ/kg, multiplied by the amount of fuel burnt in kg/s gives the heat input in kJ/s (kW).

The heat output is the difference in heat content of feedwater and steam (or hot water) produced, multiplied by the flow rate of water or steam. The heat content of water and steam is expressed in kJ/kg, and the flow rate of water or steam is expressed in kg/s, which yields the heat output also in kJ/s (kW).

Example 2.1

Find out the efficiency of a boiler operating on coal by the direct method using the following operating data.

Quantity of coal consumed = 1.6 T/hour

Quantity of steam (dry) generated = 7.5 T/hour

Steam pressure = 12 bar (absolute)

Steam temperature = 180°C

Feedwater temperature = 80°C

GCV of coal = 13,200 kJ/kg

Solution

Enthalpy of steam at 12 bar (absolute) pressure = 2784 kJ/kg (h_g value from Table 1.4)

Enthalpy of feedwater at 80°C = 335.2 kJ/kg (80°C x 4.19 kJ/kg)

Boiler efficiency = $\left(\frac{\text{Heat output}}{\text{Heat input}}\right)$ x 100

$$= \left(\frac{7.5 \text{ x } 1000\ (2784 - 335.2)}{1.6 \text{ x } 1000 \text{ x } 13{,}200}\right) \text{ x } 100$$

= 87%

The overall efficiency of a boiler is lower than the thermal efficiency as it takes into account radiative and convective losses from the boiler and other losses, such as cycle losses, due to passing of air through the boiler during the “off” cycle.

Measurement of boiler efficiency

There are two basic methods of measuring boiler efficiency called the “Direct method” and the “Indirect method”.

1) Direct method

This method is used where the energy gain of the working fluid (water and steam) is compared to the energy content of the fuel used. The energy input is computed based on the calorific value of the fuel and the amount of fuel used, while the energy output is computed based on the amount of steam generated and the heat content of the feedwater and steam.

For this method, the parameters to be measured are the quantity of steam generated, the quantity of fuel used, the pressure and temperature of steam generated (to find the specific enthalpy of the steam), the temperature of the feedwater (to find the specific enthalpy of the feedwater) and the gross calorific value of the fuel used.

$$\text{Boiler efficiency } (\eta) = \frac{\text{Enthalpy of steam - Enthalpy of feedwater}}{\text{Heat released in boiler}} \quad (2.2)$$

$$= \frac{Q \text{ x } (h_g - h_f)}{q \text{ x GCV}} \quad (2.3)$$

where,

Q = quantity of steam generated in kg

q = quantity of fuel used in kg

h_g = specific enthalpy of saturated steam in kJ/kg of steam

h_f = specific enthalpy of feedwater in kJ/kg of water

GCV = gross calorific value in kJ/kg

Example 2.2

The following data was recorded during a boiler test:

Mass of solid fuel used = 280 kg

Water evaporated = 2450 kg

Steam pressure = 12 bar (absolute)

Dryness fraction of steam = 0.97

Feedwater temperature = 52ºC

The calorific value of the fuel used is 28,500 kJ/kg and the enthalpy of the feedwater (from steam tables) is 217.6 kJ/kg. Compute the boiler efficiency using the direct method.

Solution

From the steam tables, enthalpy of steam generated at 12 bar

h_f = 798 kJ/kg

h_{fg} = 1986 kJ/kg

Enthalpy equation for wet steam, $h_g = h_f + x\ h_{fg}$

$= 798 + (0.97 \times 1986)$

$= 2724$ kJ/kg

Boiler efficiency $= Q \times (h_g - h_f) / q \times GCV$ (from equation 2.2)

$= [2450 \times (2724 - 217.6)] / [280 \times 28{,}500]$

$= 0.77$

$= 77\ \%$

2) Indirect method

In this method, the boiler efficiency is computed by subtracting the percentage values of the various losses from 100.

Boiler efficiency by indirect method = 100 – (A + B + C + D + E + F + G) (2.4)

Where the major losses that occur in boilers are subtracted to estimate the efficiency (based on per kg of fuel) are:

A = Percentage of heat loss due to dry flue gas

B = Percentage of heat loss due to evaporation of water in hydrogen in fuel

C = Percentage of heat loss due to moisture in fuel

D = Percentage of heat loss due to moisture in combustion air

E = Percentage of heat loss due to incomplete combustion

F = Percentage of heat loss due to radiation and convection

G = percentage of heat loss due to unburnt carbon in fly ash and bottom ash

Percentage of heat loss due to dry flue gas

This is the greatest loss, which is due to the heat carried away by the boiler flue gases.

$$A = \left(\frac{\mathrm{m\ x\ Cp\ x\ (T_f - T_a)}}{\mathrm{GCV}}\right) \text{ x } 100 \qquad (2.5)$$

where,

m = mass of dry flue gas in kg/kg of fuel

Cp = specific heat capacity of fuel gas in kJ/kg K

T_f = flue gas temperature in °C

T_a = boiler room air temperature in °C

GCV = gross calorific value of fuel in kJ/kg

Percentage of heat loss due to evaporation of water in hydrogen in fuel

The combustion of hydrogen causes a heat loss because the product of combustion is water. The evaporation of water absorbs the heat in the form of latent heat.

$$B = \left(\frac{H_2 \times [2676 + Cp \times (T_f - T_a)]}{GCV}\right) \times 100 \qquad (2.6)$$

where,

H_2 = kg of hydrogen present in fuel (per kg of fuel)

Cp = specific heat of superheated steam in kJ / kg °K

T_f = flue gas temperature in °C

T_a = boiler room air temperature in °C

2676 = latent heat of water vapour (corresponding to pressure) in kJ/kg

GCV = gross calorific value of fuel in kJ/kg

Percentage heat loss due to moisture present in fuel

Moisture enters the boiler with the fuel. This moisture is first heated to bring it to the boiling point (sensible heat) and evaporated (latent heat) and then superheated to the temperature of the exhaust gas.

$$C = \left(\frac{M \times [2676 + Cp \times (T_f - T_a)]}{GCV}\right) \times 100 \qquad (2.7)$$

where,

M = kg moisture in fuel on 1 kg basis

Cp = specific heat of superheated steam in kJ/kg °C

T_f = flue gas temperature in °C

T_a = boiler room air temperature in °C

2676 = latent heat of water vapour (corresponding to pressure) in kJ/kg

GCV = gross calorific value of fuel in kJ/kg

Percentage heat loss due to moisture in air

Water vapour in the air provided for combustion is superheated as it passes through the boiler and exits with the flue gases.

$$D = \left(\frac{m \times \omega \times Cp \times (T_f - T_a)}{GCV}\right) \times 100 \qquad (2.8)$$

where,

m = actual mass of air supplied per kg of fuel

ω = kg of water / kg of dry air (see Table 2.5)

Cp = specific heat of superheated steam in kJ/kg K

T_f = flue gas temperature in (°C)

T_a = boiler room air temperature in °C

GCV = gross calorific value of fuel in kJ/kg

Dry bulb temperature of air (°C)	Relative humidity of air (%)	ω (g of water / kg of dry air)
20	50%	7.2
20	70%	10.1
25	50%	9.78
25	70%	13.72
30	50%	13.13
30	70%	18.43
35	50%	17.44
35	70%	24.52
40	50%	22.96
40	70%	32.32

Table 2.5 Typical humidity ratio values for air

Percentage heat loss due to incomplete combustion

Products formed by incomplete combustion could be mixed with oxygen and burned again with a further release of energy. Such products include CO, H2, and various hydrocarbons and are generally found in the flue gas of the boilers. Carbon monoxide is the only gas whose concentration can be determined conveniently in a boiler plant test.

$$E = \left(\frac{\%CO \text{ x } C}{\%CO + \%CO_2}\right) \text{ x } \left(\frac{5744}{GCV}\right) \text{ x } 100 \qquad (2.9)$$

where,

%CO = % of CO in flue gas

$\%CO_2$ = % of CO_2 in flue gas
C = carbon content kg/kg of fuel
5744 = heat loss due to partial combustion of carbon kJ/kg
GCV = gross calorific value of fuel in kJ/kg

Percentage heat loss due to radiation and convection

This accounts for heat losses from the boiler shell in the form of radiation and convection (explained later). There are many expressions for calculating these two losses. One such equation is expressed in equation (2.10).

$$L_{CR} = 0.548 \text{ x } \left[\left(\frac{T_s}{55.55}\right)^4 - \left(\frac{T_a}{55.55}\right)^4\right] + 15.97 \text{ x } (T_s - T_a) \text{ x } 1.25 \text{ x } \sqrt{\frac{(196.85\ V_m + 68.9)}{68.9}} \quad (2.10)$$

where,
L_{CR} = convective and radiative losses in W/m^2
V_m = wind velocity in m/s
T_s = surface temperature of boiler in °C
T_a = boiler room air temperature in °C

Since such expressions are tedious to use, normally estimated values are used. Based on the American Boiler Manufacturers Association, the convective and radiative losses can be taken as 1 to 1.5% at 100% boiler loading for a typical package boiler of 5 to 10 T/hour (lower value is at the higher capacity range). These losses increase to between 2 to 3% at 50% load and 4 to 8% at 25% boiler loading.

Percentage heat loss due to unburnt carbon in fly ash and unburnt carbon in bottom ash

Small amounts of carbon will be left in the ash and this constitutes a loss of potential heat in the fuel. Samples of ash can be analysed to quantify the carbon content.

G = heat loss due to unburnt fly ash + heat loss due to unburnt bottom ash

$$\text{Loss due to unburnt fly ash} = \left(\frac{\left(\frac{\text{kg of ash collected}}{\text{kg of fuel burnt}}\right) \text{ x GCV of fly ash}}{\text{GCV of fuel}}\right) \text{ x } 100 \quad (2.11)$$

$$\text{Loss due to unburnt bottom ash} = \left(\frac{\left(\frac{\text{kg of ash collected}}{\text{kg of fuel burnt}}\right) \text{ x GCV of bottom ash}}{\text{GCV of fuel}}\right) \text{ x } 100 \quad (2.12)$$

Comparison of direct and indirect efficiency

Both the methods described above are able to compute boiler efficiency. The main benefit of the indirect method is that it also identifies each of the sources of losses. This helps to identify which losses are the greatest so action can be taken to reduce them. On the other hand, although direct efficiency calculation does not provide the magnitude of individual losses, it is easier to compute. The values computed by the two methods may not be exactly the same because, indirect efficiency is measured at a particular time whereas direct efficiency is measured over a period of time (hence losses due to fluctuating loads, boiler switching on-off are also taken into consideration).

Summary

This chapter provided an introduction to steam boilers, how they operate, their main characteristics and the various methods of measuring boiler efficiency. The different types of boilers, their key characteristics and how to assess boiler efficiency were explained in this chapter.

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3.0 APPLICATIONS OF STEAM

Steam is used for various heating applications in industry and for power generation. Steam is generated in boilers, which are usually centrally located. Thereafter, the generated steam is distributed to the various users.

A typical steam distribution system where steam generated in a boiler is used for jacket heating, process heating, sparge coils and ejector systems is shown in Figure 3.1. The steam supplied for jacket heating and process heating in heat exchangers is condensed and normally returned to the boiler unlike in applications like sparge coils and ejectors where the steam is directly injected into the system.

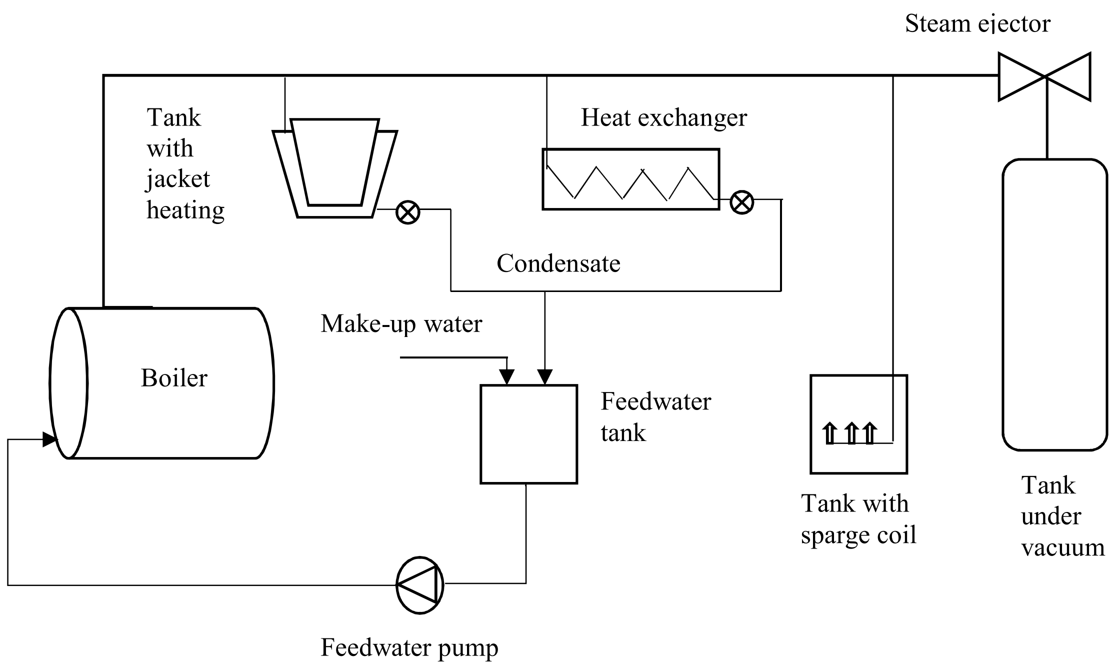


Figure 3.1 A typical steam distribution system

This chapter will describe the main components found in steam distribution systems and their role in supplying good quality steam. Thereafter, some of the main applications of steam will be illustrated.

Learning Outcomes:

The main learning outcomes from this chapter are to understand:

1. The main components in steam distribution systems
2. Key design considerations for steam distribution systems
3. Common applications of steam

3.1 Steam distribution systems

System pressure

When steam flows through the distribution system from the boiler to the users, the pressure reduces due to losses caused by pipe and fitting resistance to the steam flow. In addition, pressure losses occur in distribution systems due to condensing of steam as a result of heat loss to the environment.

Since most steam users require a minimum pressure to be maintained, the system pressure has to be set after accounting for the above losses. Often, the pressure is set higher than the minimum requirements because at higher pressure, the specific volume of steam is less (explained in section 1.6). Since steam occupies less volume per unit mass at higher pressure, smaller diameter pipes and fittings can be used for the distribution system.

However, as will be explained in section 4.2, the latent heat available reduces when the steam pressure is increased, and this results in an increase in steam demand for heating applications.

Pressure reducers

Since the system pressure is set based on the minimum pressure requirements of the user needing the highest pressure and other considerations like sizing of distribution piping and minimum operating pressure for the boiler, the steam supply pressure may exceed the maximum allowable pressure for some users. In such cases, a pressure-reducing valve is used to automatically maintain the downstream pressure.

The image of a typical pressure-reducing valve is shown in Figure 3.2. The device contains a mechanism to regulate the steam flow through it to maintain the downstream pressure at the desired value.

Usually a separator is installed prior to the reducing valve to remove water from the wet steam. Also, a safety valve is installed downstream of the pressure reducer to protect the steam-using equipment in the event of a failure of the pressure reducer.

Figure 3.2 Image of a pressure-reducing valve (image taken from the Spirax Sarco website 'Steam Engineering Tutorials' at http://www.spiraxsarco.com)

Separators

Steam separators are used in steam distribution systems to remove water from wet steam. When steam flows through the distribution system, some steam condenses due to heat losses through the pipe wall. The condensed steam, if not removed, will form droplets and will travel with the steam. Once water enters a heat exchanger, it reduces the heat transfer efficiency. Water droplets can also damage equipment like control valves. The arrangement of a steam separator is shown in Figure 3.3.

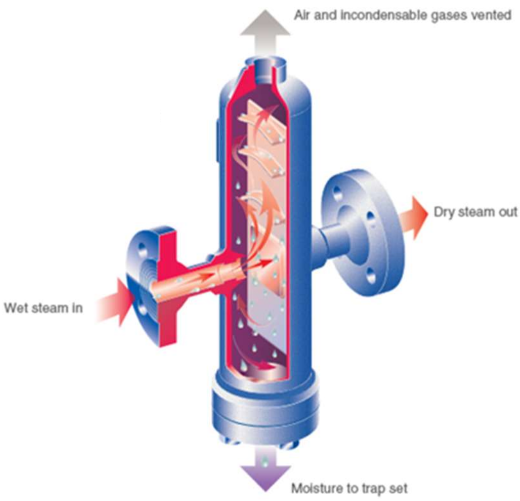


Figure 3.3 Cut-away of a steam separator (illustration taken from the Spirax Sarco website 'Steam Engineering Tutorials' at http://www.spiraxsarco.com)

Air venting

Boiler feedwater can contain dissolved air. When the water is heated in the boiler, the air is released and it will mix with steam. If air is not removed, it can form pockets in heat transfer equipment and reduce the surface area available for heat exchange. Also, when air is mixed in steam, the heat content of steam will be lower.

Air can also be present in equipment and distribution piping at start-up. When a steam system is shut down, the steam inside it will condense and form a vacuum which then can draw air inwards.

Some common signs of air in the systems are reduction in heating output of equipment, air bubbles in the condensate and corrosion.

Deaerators (explained in section 2.2) are used in high pressure systems to remove air from feedwater. However, air can still be present in steam systems and require vents to remove it. Air vents are devices that can automatically remove air. They are installed in locations where air tends to accumulate like the upper part of steam jackets, the end of steam mains and the top level of closed vessels.

Drainage

Since some steam will condense in steam pipes due to heat losses, the distribution system has to be designed to effectively remove condensate. Steam mains should be designed to have a gradient downwards in the direction of steam flow as shown in Figure 3.4. The gradient should be at least 100 mm for every 10 m in pipe length.

The gradient is designed to be downwards so that the condensate and steam will flow in the same direction and condensate can be removed by creating suitable drain points. If the gradient is set upwards, the condensate will tend to flow backwards while the steam will drag it upwards.

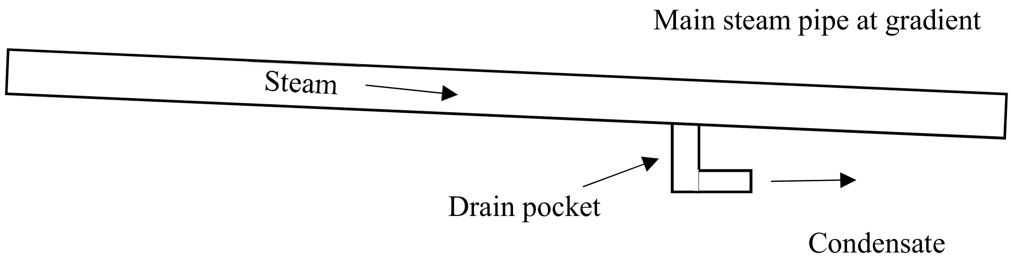


Figure 3.4 Recommended arrangement for steam mains

Drain pockets should be installed along the steam mains at intervals sufficient to remove condensate. Normally, drain points are to be located every 30 to 50 m and at low points in the system. The arrangement of a steam drain pocket is shown in Figure 3.5.

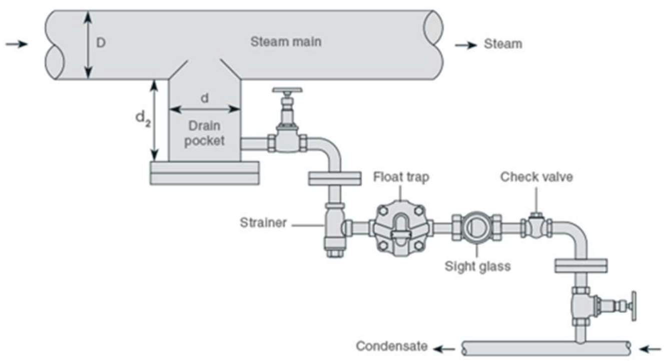


Figure 3.5 Arrangement of a steam drain pocket (illustration taken from the Spirax Sarco website 'Steam Engineering Tutorials' at http://www.spiraxsarco.com)

Branch connections should be taken from the top of the main pipes rather than from the side or bottom to ensure dryness of steam as shown in Figure 3.6.

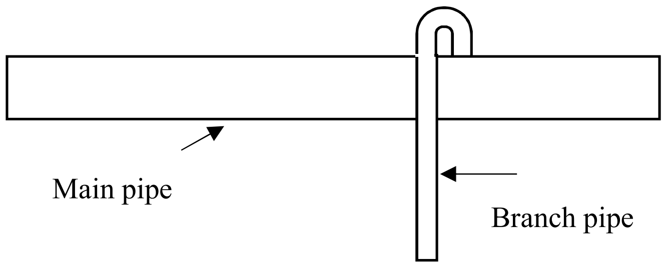


Figure 3.6 Recommended arrangement of a branch connection

Traps

Steam traps are used in steam systems to remove condensate and non-condensable gases. They are mainly used for steam heating coils and for condensate removal from steam headers.

Steam traps are generally classified as thermostatic, mechanical, or thermodynamic. Thermostatic steam traps are designed to work based on the difference in temperature of steam and condensate. They usually contain a bimetallic strip to allow subcooled condensate to be removed while preventing live steam, which is at a higher temperature, from passing through (Figure 3.7). These traps can be used for jacketed piping, steam tracers, and heat transfer equipment, which can accommodate back up of condensate.

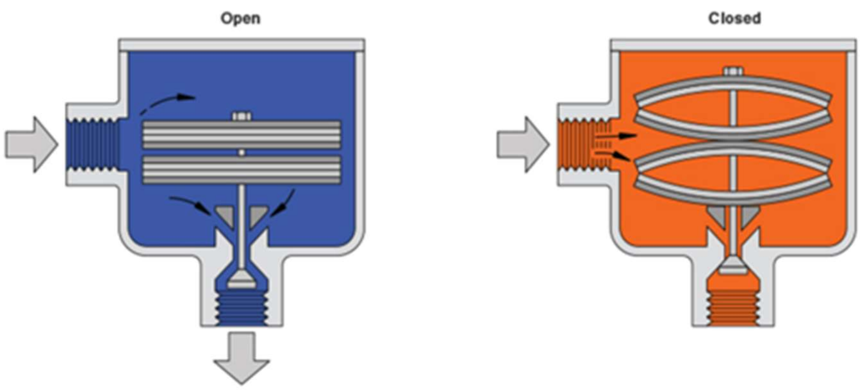
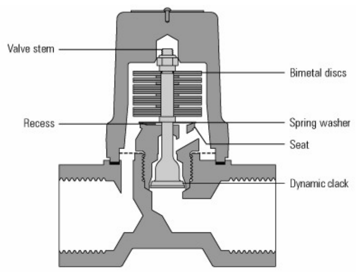


Figure 3.7 Operation of a thermostatic (bimetal) steam trap (images taken from the Spirax Sarco website 'Steam Engineering Tutorials' at http://www.spiraxsarco.com)

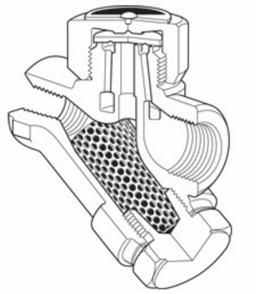
Bucket and float type traps are common types of mechanical steam traps. As the names imply, they have floating balls or buckets that operate on the buoyancy of condensate to mechanically open and close ports in the traps to discharge only condensate. They usually have built-in air-venting features and are used for continuous and intermittent loads. They are commonly used on steam heat exchanger coils.

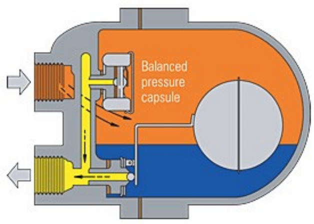
Thermodynamic traps operate based on the difference in flow characteristics of steam and condensate. When air or condensate enters the trap, a disc lifts up to allow it to be discharged. When steam enters the trap, due to its increased velocity (higher velocity pressure), the static pressure below the disc is reduced, which lowers the disc, closing the trap. They are often used for condensate removal from main steam distribution pipes.

Commonly used steam traps are illustrated in Figure 3.8.

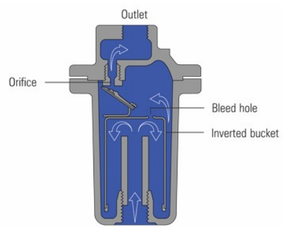


Bimetal steam trap


Thermodynamic steam trap



Float type steam trap



Inverted bucket steam trap

Figure 3.8 Common types of steam traps (images taken from the Spirax Sarco website 'Steam Engineering Tutorials' at http://www.spiraxsarco.com)

Pipe sizing

Steam pipes can be sized based on steam velocity and pressure drop. When sizing based on velocity, the pipe inner diameter is selected to maintain the steam pressure within an acceptable range. The common design velocity is in the range of 25 to 35 m/s.

The cross-sectional area of the pipe is calculated using the design flow velocity, specific volume of steam and the mass flow rate of steam.

$$A = \frac{Q}{v} \tag{3.1}$$

where,
A = cross-sectional area of pipe (m^2)
Q = volumetric flow rate (m^3/s)
v = velocity of steam (m/s)

Q is the product of mass flow rate (kg/s) and specific volume (m^3/kg).

Example 3.1

Compute the pipe size required to carry 3000 kg/hour of dry saturated steam at 8 bar (gauge pressure) ensuring that the steam velocity does not exceed 25m/s.

Solution

From Table 1.3, specific volume is 0.21 m^3/kg at (9 bar absolute).
Mass flow rate of steam = 3000 kg/hour
Volumetric flow rate, Q = 3000/3600 (kg/s) x 0.21 (m^3/kg) = 0.175 m^3/s

From equation (3.1),
A = 0.175 (m^3/s) / 25 (m/s)
= 0.007 m^2

Since, $(\pi \times D^2)/4 = A = 0.007$

The required pipe diameter, D is 47 mm (50 mm).

In some designs, it may be necessary to size piping based on pressure drop to ensure that the pressure does not drop below a certain minimum value. In such cases, reference charts that indicate the recommended pipe diameter for a set of steam pressure, steam flow rate and pressure loss per unit length can be used.

3.2 Common applications of steam

Non-flow heating

Non-flow type heating is when a substance in a vessel is heated. The heating can be through an internal steam coil or an external jacket. Heating in non-flow applications serve two purposes, namely heating the material to the required temperature, and maintaining the temperature while heat is lost to the surroundings.

The heat required at start-up to raise the temperature of the material being heated can be computed using the following equation.

$$Q = m \times Cp \times \Delta T \qquad (3.2)$$

where,

Q = heat added (kW)

m = mass of the material being heated (kg)

Cp = specific heat capacity of material (kJ/kg.K)

ΔT = rise in temperature of the material (K)

Example 3.2

Estimate the heat required to raise the temperature of 10,000 kg of oil having a specific heat capacity of 2.2 kJ/kg.K from 30^{o}C to 70^{o}C.

Solution

Using equation (3.2),

$$\begin{aligned} Q &= 10{,}000\ (kg) \times 2.2\ (kJ/kg.K) \times [70-30]\ (K) \\ &= 880{,}000\ kJ \end{aligned}$$

The mean heating rate required is the amount of heat added divided by the time taken to raise the temperature from the initial to the final temperature.

Example 3.3

Compute the mean heating rate for the case in Example 3.2 if it takes 2 hours to raise the temperature to 70°C.

Solution

Total heating required is 880,000 kJ

Mean heating rate = 880,000 / (2 hours x 3600 seconds)
= 122.2 kW (kJ/s)

The heat required to maintain the material temperature in a vessel while heat is lost to the surroundings can be estimated using the heat transfer surface area and the overall heat transfer coefficient as follows.

$$Q_{loss} = U \times A \times \Delta T \qquad (3.3)$$

Where,

Q_{loss} = heat loss from tank surface (W)

U = overall heat transfer coefficient (W/m^2.K)

A = heat transfer surface area (m^2)

ΔT = temperature difference between material in the tank and surrounding air (K)

The overall heat transfer value depends on many factors like the thickness of the insulation, surrounding air velocity and orientation of the heat transfer surface. Some typical values are provided in Table 3.1.

Temperature difference (°C)	**25 mm insulation**			**50 mm insulation**		
	Base	**Side**	**Top**	**Base**	**Side**	**Top**
30	1.5	2.2	2.8	0.75	1.1	1.4
50	1.6	2.4	3.0	0.8	1.2	1.5
70	1.7	2.5	3.2	0.85	1.25	1.6
90	1.8	2.6	3.3	0.9	1.3	1.7
100	1.9	2.8	3.4	0.95	1.4	1.8

Table 3.1 Typical U values in W/m^2.K

The values in Table 3.1 should be corrected for air movement by multiplying with the factors in Table 3.2.

Air velocity (m/s)	0	1	2	4	6	10
Multiplication factor	1	1.4	1.7	2.4	3.0	4.1

Table 3.2 Correction factors for U value

Example 3.4

Estimate the heat loss from side walls of a tank having a surface area of 100 m^2. The material in the tank is maintained at 80^{o}C. The tank has a 50 mm insulation. The surrounding air is at 30^{o}C and the average air velocity is 1 m/s.

Solution

A = 100 m^2

ΔT = (80 – 30) = 50^{o}C

U = 1.2 W/m^2.K (From Table 3.1, side wall with 50mm insulation and ΔT of 50^{o}C)

Correction factor for air movement of 1 m/s is 1.4 (Table 3.2)

From equation (3.3),

Q_{loss} = (1.2 x 1.4) x 100 x 50 (W)

= 8.4 kW

Flow-type heating

Flow-type heating takes place when the material being heated flows over a heating surface. Typical flow-type heating devices are shell and tube heat exchangers, plate type heat exchangers and air heaters.

The relationship between the steam temperature and the increase in temperature of the material being heated is illustrated in Figure 3.9. If the flow rate of the material being heated is constant, the heat transfer rate is proportional to the temperature difference between the two streams. This temperature difference is greatest at the inlet and reduces at the outlet. The mean of these two values can be used to estimate the instantaneous heating done. For most steam heating applications, since the steam temperature remains constant throughout the entire heat exchanger, the arithmetic mean can be used (the concept of Log Mean Temperature Difference or LMTD will be described in Chapter 13 when describing heat recovery systems).

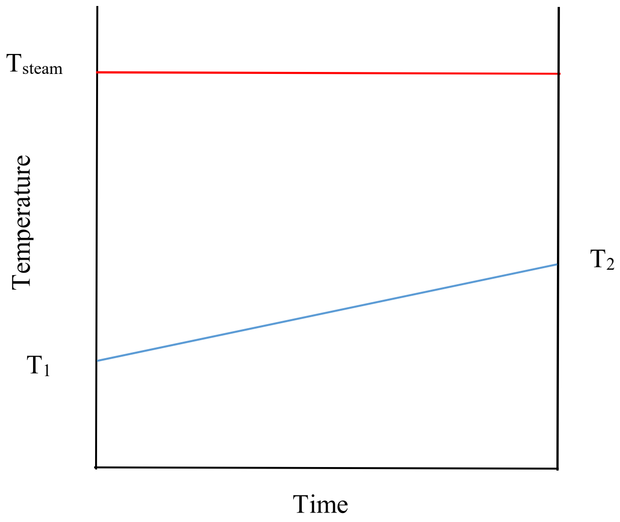


Figure 3.9 Temperature profile for flow-type heating

The heat transfer rate can be estimated using equation (3.4).

$$Q = U \times A \times \Delta Tm \qquad (3.4)$$

where,

Q = rate of heat transfer (W)

A = heat transfer surface area

U = overall heat transfer coefficient for heat exchanger ((W/m^2.K)

ΔTm = mean temperature difference (arithmetic or LMTD) (K)

$$\Delta Tm = T_S - \left(\frac{T_2 + T_1}{2}\right) \qquad (3.5)$$

The U value depends on many factors such as the type of heat exchanger and flow velocity of the fluid and will be explained in detail in Chapter 13.

Example 3.5

Steam having a saturation temperature of 140^{o}C is used to heat a liquid from 30^{o}C to 70^{o}C. The heat transfer surface area is 10m^2. The U value for the heat exchanger is 500 W/m^2.C. Compute the heat transfer rate.

Solution

$\Delta Tm = 140 - [(70 + 30) / 2] = 90\ ^{o}C$

$Q = U \times A \times \Delta Tm = 500\ (W/m^2.K) \times 10\ (m^2) \times 90\ (K)$

$= 450\ kW$

Sparge coils

Sparge coils are steam coils with holes that are submerged in the liquid being heated. They inject steam directly into the liquid being heated and the steam bubbles condense to release heat. This method of heating is used when dilution and increase in mass of the liquid being heated is acceptable. Sparge coils are therefore usually used for heating water.

The mean heat transfer rate can be computed using equation (3.6).

$$Q = m_s \times (h_g - T \times Cp) \qquad (3.6)$$

where,

Q = mean heat transfer rate (kW)

m_s = mass flow rate of steam (kg/s)

h_g = enthalpy of steam (kJ/kg)

T = final temperature of water (°C)

Cp = specific heat capacity of water (kJ/kg.C)

The above equation shows that the latent heat and liquid enthalpy of the steam (up to the temperature at the end of the heating process) are used to heat the water.

Example 3.6

3000 kg of water is to be heated from 30°C to 70°C using a sparge coil with steam at 4 bar (absolute). If the water is to be heated in two hours, estimate the required mean steam flow rate. The specific heat capacity of water is 4.19 kJ/kg.K.

Solution

Total heat required from equation (3.2),

$Q = m \times Cp \times \Delta T$

$= 3000\ (kg) \times 4.19\ (kJ/kg.K) \times (70 - 30)\ (K)$

= 502,800 kJ

The mean heat transfer rate = 502,800 / 2 (hours) x 3600 (seconds)

= 69.8 kW

From equation (3.6),

The mean heat transfer rate, 69.8 kW = m_s x (h_g – T x Cp)

h_g = 2739 kJ/kg (from Table 1.5 for steam at 4 bar absolute pressure)

Therefore, the required steam mass flow,

m_s = 69.8 / (2739 – 70 x 4.19) = 0.029 kg/s (102.8 kg/hour)

Tracers

When transporting some substances through piping systems, it is necessary to maintain a minimum temperature to prevent it from solidifying inside the pipes or to maintain the viscosity. Usually pipes used to convey such materials are insulated to maintain the temperature inside the pipe above the minimum required value. However, in some cases, the insulation itself is not sufficient to maintain the temperature. In such situations, steam tracing is used.

Steam tracing involves having a small diameter steam pipe along the outside surface of the pipe within the insulation or having a steam jacket over the entire outer surface of the pipe as shown in Figure 3.10.

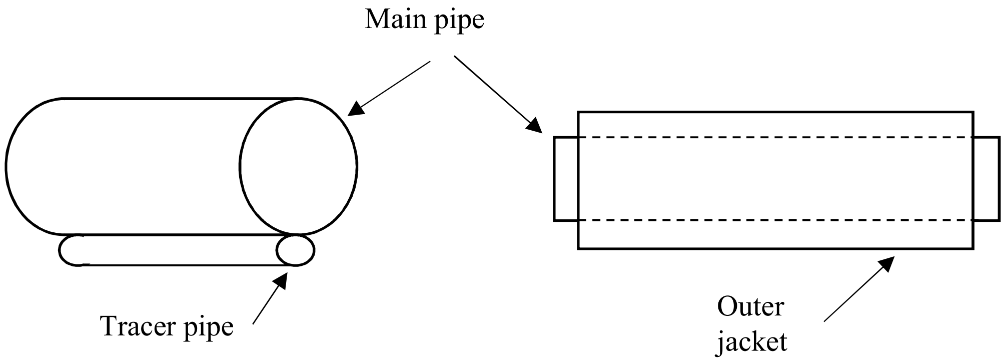


Figure 3.10 Arrangement of steam tracing

The size and quantity of steam tracing lines required for a particular application depends on the heat loss from the pipe, which in turn depends on the pipe insulation, temperature difference between material in the pipe and ambient air and wind speed.

Some approximate values of heat loss in W/m length of pipe at various temperature differences between the material in the pipe and surrounding air are summarised in Table 3.3, for piping with 50 mm insulation and average wind speed of 10m/s.

Temperature difference (°C)	**Pipe diameter**			
	100 mm	**200 mm**	**300 mm**	**500 mm**
25	15	25	35	50
75	45	70	100	150
100	60	100	135	200
150	85	145	200	300
200	115	190	270	400

Table 3.3 Approximate heat loss from pipes in W/m

Example 3.7

Estimate the heat loss from a 300 mm diameter pipe with 50 mm thick insulation, which is 150 m in length. The material inside the pipe is at 130°C and the surrounding air is at 30°C. Also compute the steam demand for the tracer if it is supplied with steam at 5 bar pressure (absolute).

Solution

Heat loss per metre length of the pipe from Table 3.3 is 135 W/m (300 mm pipe with temperature difference of 100°C).

Total heat loss for 150 m pipe length = 135 x 150 = 20,250 W = 20.25 kW

Latent heat of steam at 5 bar (absolute) from Table 1.5 is 2109 kJ/kg

Therefore, steam demand = 20.25 / 2109 = 0.01 kg/s = 34.6 kg/hour

Ejectors

Steam ejectors operate based on the venturi principal to create a vacuum. As shown in Figure 3.11, high pressure steam is supplied to an expanding nozzle. The nozzle provides expansion of the motive steam to convert pressure into velocity, which creates a vacuum. The motive steam and the gas removed in creating the vacuum are passed through a diffuser so that the velocity can be converted to pressure to achieve the required discharge pressure.

Steam ejectors can be arranged in a number of stages to achieve higher vacuum levels. They can handle condensable and non-condensable gases and are commonly used in petrochemical, food and process industries.

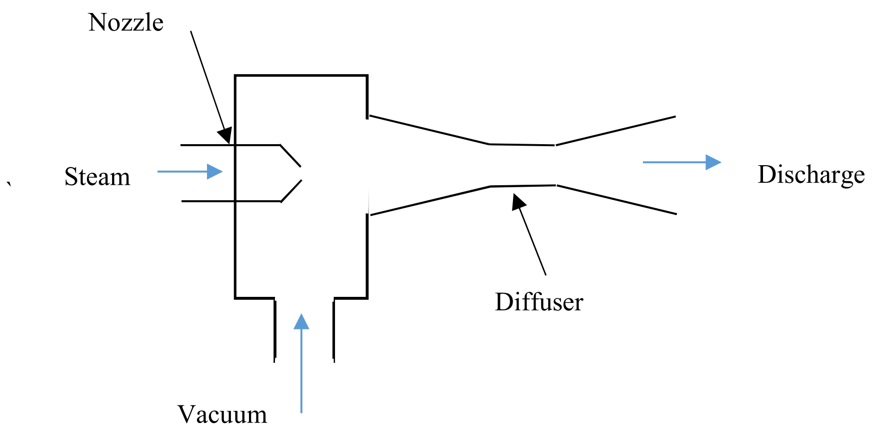


Figure 3.11 Arrangement of a steam ejector

Power generation

Steam is also commonly used for power generation using the thermodynamic Rankine cycle. As shown in Figure 3.12, high pressure steam is generated in a boiler using a variety of different fuels such as oil, gas, coal or biomass and expanded in a steam turbine, which is coupled to an alternator. The low-pressure steam exiting the turbine is condensed and fed back to the boiler. The typical operating efficiency of this cycle is about 45%.

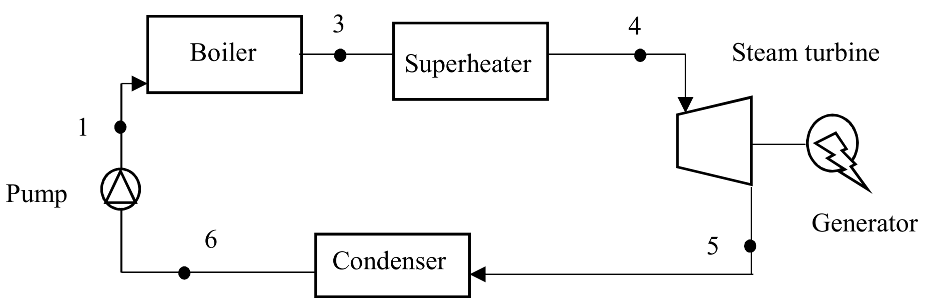


Figure 3.12 Arrangement of a steam turbine power generation plant

The temperature vs entropy (T-S) diagram for a steam turbine system is shown in Figure 3.13 (details of this cycle are explained in detail in the SCEM reference manual for the module on Combined Heat and Power (CHP) systems.

The cycle consists of four processes, which are:

- 6 to 1: Isentropic compression (pump)
- 1 to 3: Isobaric heating (boiler)
- 3 to 4: Superheating of steam
- 4 to 5: Isentropic expansion (steam turbine)
- 5 to 6: Isobaric cooling (condenser)

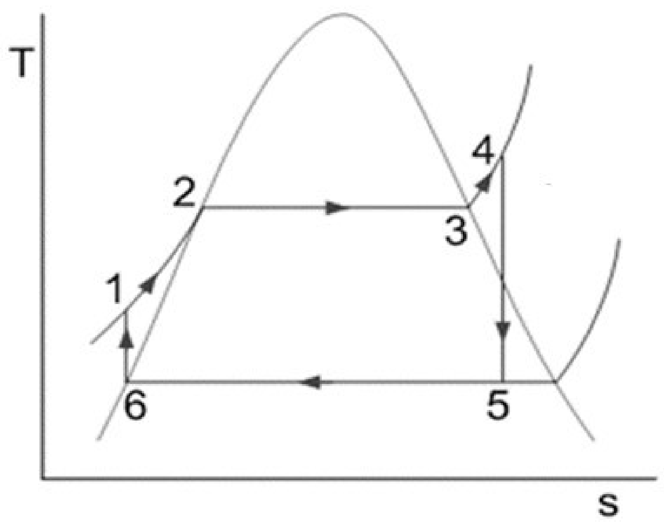


Figure 3.13 T-S diagram for steam turbine

The work and heat inputs and work output can be expressed in terms of specific enthalpies at the different points and the mass flow rate.

Output from the turbine = m x (h_4 – h_5)
Work input to pump = m x (h_1 – h_6)
Heat input = m x (h_4 – h_1)

where m is the mass and h is the specific enthalpy.

The thermal efficiency of the system, η = net work output / heat input

$\eta = [m (h_4 - h_5) - m (h_1 - h_6)] / m (h_4 - h_1)$

$$\eta = [(h_4 - h_5) - (h_1 - h_6)] / (h_4 - h_1) \tag{3.7}$$

Summary

This chapter covered the main components found in steam distribution systems and their role in supplying good quality steam. Thereafter, some of the main applications of steam were described and examples were used to illustrate important design aspects for these systems.

References

1. Improving Steam System Performance: A Source Book for Industry. US Department of Energy, Industrial Technologies Program, Washington DC, 2012
2. Jayamaha, Lal, Energy-Efficient Industrial Systems, Evaluation and Implementation, McGraw-Hill, New York, 2016.
3. The Steam and Condensate Loop, Spirax Sarco Limited, England 2008.

4.0 OPTIMISING STEAM SYSTEMS

Boilers and steam systems are widely used in industrial facilities for various process heating and power generation applications. Boilers use solid, liquid and gaseous fuel and produce steam, hot water, hot oil or other thermic fluids. They account for a significant portion of the energy consumption in many industrial plants.

This chapter describes in detail the most common energy-saving measures that can be implemented to optimise the operation and energy efficiency of such boilers and heating systems. Various examples will be used to illustrate the energy savings achievable for each energy-saving measure.

Learning Outcomes:

The main learning outcomes from this chapter are to understand:

1. How to improve the operating efficiency of boilers
2. How to reduce losses and energy wastage in steam systems
3. How to optimise the operation of auxiliary systems used with boilers

4.1 Improving combustion efficiency

The major loss in any boiler is due to the hot gases discharged into the chimney. If there is a lot of excess air, the increased quantity of exhaust gas will lead to extra flue gas losses. Similarly, insufficient air for combustion results in wastage of fuel due to incomplete combustion and reduces the heat transfer efficiency due to soot build-up on heat transfer surfaces.

The amount of excess air required depends on the type of fuel and, in general, a minimum of about 10 to 15% excess air is required for complete combustion. This translates to about 2 to 3% excess oxygen.

As explained earlier, boiler combustion efficiency, which indicates the ability of the combustion process to burn the fuel completely (with minimum excess air) can be measured by sampling the exhaust flue gas to find its composition and temperature using a combustion analyser. Most good combustion analysers are able to give a direct reading of the combustion efficiency based on the fuel used. If this facility is not available on the instrument used, a combustion efficiency versus oxygen (O_2) concentration chart (Figure 4.1) can be used to estimate the combustion efficiency.

The drop in combustion efficiency due to excess air is dependent on the type of boiler and the amount of excess air. Based on the chart (Figure 4.1), if the excess air is increased from 15% to 30%, the O_2 concentration will increase from 3% to 5% (as normal air contains 21% oxygen) and the resulting drop in efficiency will be about 1%.

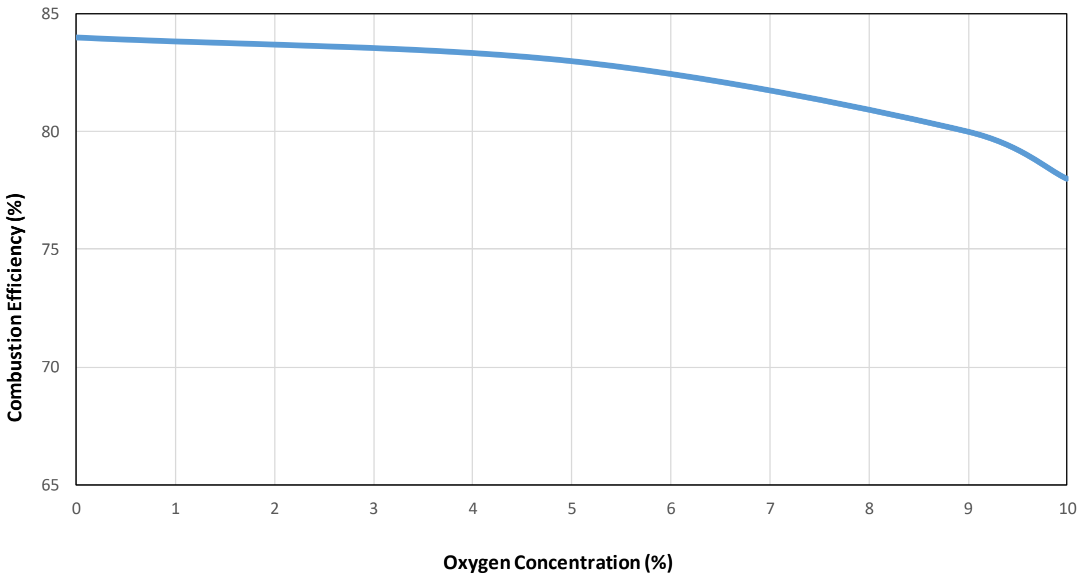


Figure 4.1 Combustion efficiency versus O_2 concentration (courtesy of Cleaver-Brooks)

Example 4.1

Flue gas contains 10% oxygen. Estimate the improvement in efficiency achievable if the oxygen level in the flue gas can be reduced to 3%.

Solution

From Figure 4.2, the combustion efficiency is 78% when the O_2 concentration in the flue gas is 10%. The same chart also shows that the combustion efficiency improves to 83.5% when the O_2 concentration in the flue gas is 3%.

Therefore, the improvement in combustion efficiency = (83.5% – 78%)

= 5.5%

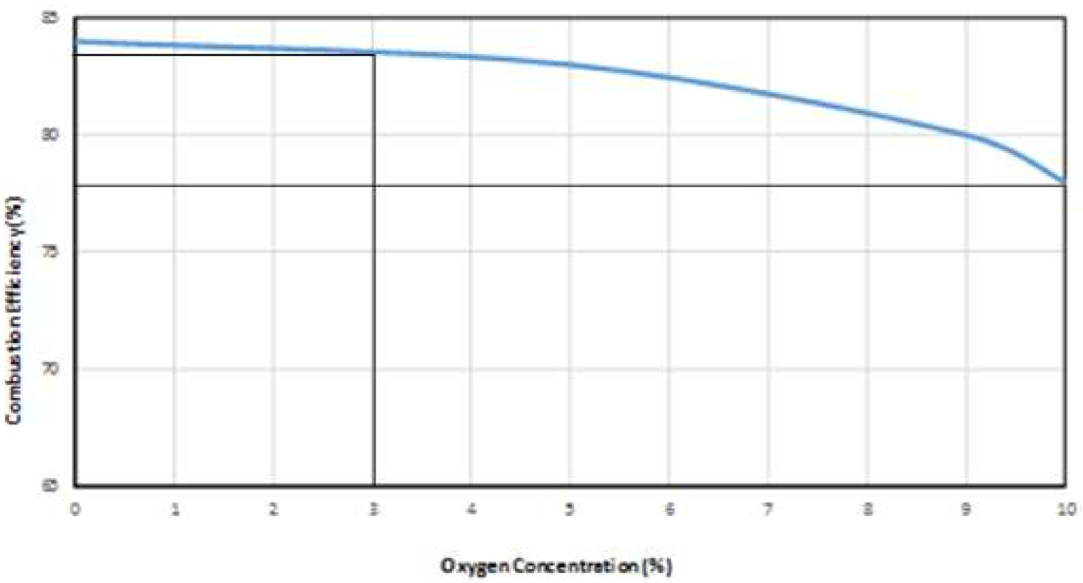


Figure 4.2 Chart for Example 4.1

For boilers operating at high-excess air levels, the combustion burner operation needs to be tuned to adjust the air to fuel ratio. This can normally be achieved by adjusting the mechanical linkages that control fuel and air flow to the burner to provide the correct ratio between the two at different operating loads for the boiler. Ideally, an oxygen (O_2) trim system should be installed which can continuously monitor the oxygen level in the flue gas and automatically adjust the air to fuel ratio to maximise combustion efficiency.

The amount of excess air also increases sometimes due to excessive draft created by the stack. If the stack is high, the natural draft created by the buoyancy of the combustion gases can be significant. This effect can be overcome by having a draft-control system, which consists of an opening with a damper installed on the exhaust duct between the boilers and the stack, to automatically control the draft by opening or closing the damper.

The amount of excess air required for combustion also depends on the type of burner. Some old burners require much more excess air for complete combustion than others. Such burners can also be replaced with low excess-air burners to improve combustion efficiency.

4.2 Reducing steam pressure

Boilers have a maximum operating pressure rating, based on their construction, as well as a minimum value to prevent carryover of water. The actual operating pressure is normally set based on the requirements of the end-users, while ensuring it is within the specified maximum and minimum values.

Since boiler efficiency depends on the operating pressure, if the operating pressure is set much higher than required, energy savings can be achieved by reducing it to match the actual requirements. Typically, reducing boiler pressure can help to improve boiler efficiency by 1 to 2%.

In addition to improving boiler efficiency, reducing steam pressure helps to reduce steam leaks and wastage due to overheating in some applications. Reducing pressure also lowers the temperature of the distribution piping which helps to cut down on losses. Another benefit of reducing pressure is the reduction of flash steam from vents of condensate recovery systems.

When the boiler pressure is reduced, more latent heat is available for heating applications (the lower the pressure, the higher the latent heat) as shown in Figure 4.3. As a result, less steam is required for a particular heating load. In addition, the heat input required to raise feedwater to the saturation temperature also reduces with lowering of steam pressure. Hence, for the same heating load, the total heat input to the boiler will reduced as illustrated in Example 4.2.

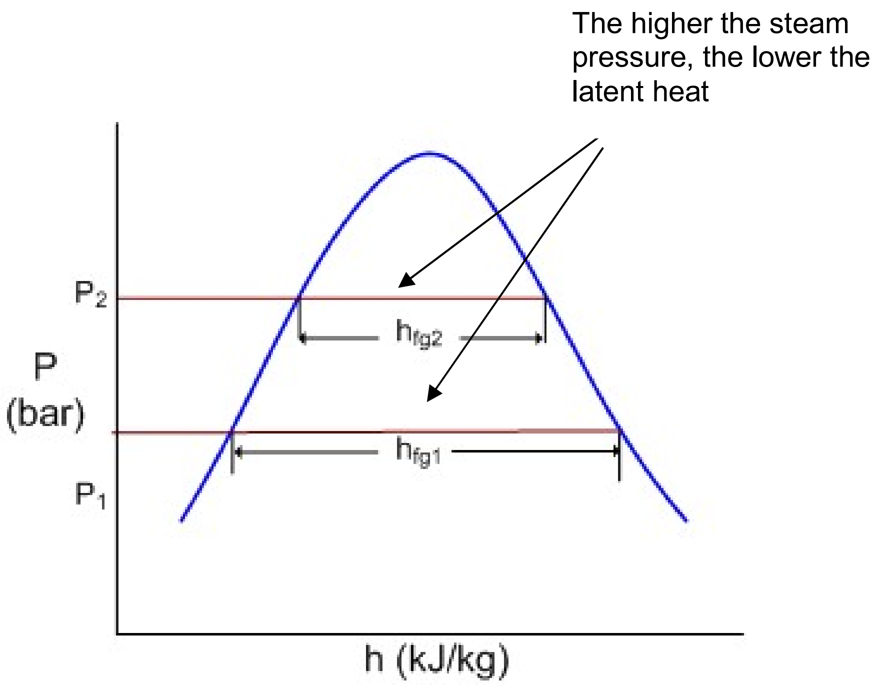


Figure 4.3 Change in latent heat with change in steam pressure

Example 4.2

Steam is generated by a boiler at 9.0 bar (gauge pressure) and the steam temperature of 180°C. The highest steam temperature required for heating applications is 140°C. Compute the savings that can be achieved if the boiler operating pressure is reduced to 6.5 bar to produce steam at 167°C.

The boiler feedwater temperature is 64°C and the average steam demand is 1.61 T/hour. The boiler operating efficiency is 83% and the boiler operates 24 hours a day and 7 days a week.

Solution

Current feedwater temperature	= 64°C
Specific enthalpy of feedwater at 64°C (steam tables)	= 272 kJ/kg
Average steam demand	= 1.61 T/hr
	= 0.447 kg/s
Present boiler pressure (gauge)	= 9.0 bar
Specific enthalpy of water (h_f) at 9.0 bar (steam tables)	= 763 kJ/kg
Latent heat (h_{fg}) at 9.0 bar (steam tables)	= 2015 kJ/kg
Estimated current heating load	= 0.447 x 2015 = 900 kW
Proposed new boiler pressure	= 6.5 bar (gauge)
Specific enthalpy of water (h_f) at 6.5 bar (steam tables)	= 710 kJ/kg
Latent heat (h_{fg}) at 6.5 bar (steam tables)	= 2055 kJ/kg

Steam flow rate required at 6.5 bar = 900 / 2055 = 0.438 kg/s

Reduction in heat required to raise the temperature of feedwater

= [0.447 x (763 - 272)] – [0.438 x (710 - 272)]

= 27.64 kW

Current boiler efficiency	= 83%
Average boiler operating hours	= 24 hours x 6 days/week
Annual fuel energy saving	= (27.64 / 0.83) x 24 x 7 x 52
	= 290,919 kWh/year

However, it should be noted that when steam pressure is reduced, the distribution pipe sizing needs to be sufficient to transport the higher volume of steam.

If it is not possible to reduce the pressure of the entire system, parts of the distribution system can be operated at lower pressures by installing pressure reducing valves at appropriate points in the distribution network.

In some systems, one steam user may require steam at a much higher pressure than the others. In such a system, if the steam usage of the high pressure user is relatively low, it may be better to have a separate steam generator (located near the user) operating at the required higher pressure while the rest of the system can be operated at a lower pressure.

4.3 Optimising operation of auxiliary equipment

Auxiliary equipment such as feedwater pumps, boiler draft fans, hot water circulating pumps and condensate pumps also consume an appreciable amount of energy. Therefore, significant energy savings can be achieved by ensuring that they are operated only when required and at the capacity required to maintain system requirements.

In some installations that have additional equipment to provide extra reliability (standby equipment) or to match certain boiler load conditions, plant operators may run more equipment than what is required to meet the operating load. In such situations, some auxiliary equipment can be switched off either manually or by using automatic controls.

Generally, each boiler has its own feedwater pump, which is automatically switched on and off to maintain the level of water in the boiler. Their operation is interlocked with the boiler so that the feedwater pump is switched off when the boiler is not in operation.

In larger systems, multiple boilers can be served by a common set of feedwater pumps as shown in Figure 4.4. In such an arrangement, the individual boilers take the required water flow by opening and closing the feedwater valves to maintain the water level in the boilers. The excess water is returned to the feedwater tank, which results in wastage of pumping energy.

This system can be improved to reduce the energy consumption of the pumps by varying the capacity (speed) of the feedwater pumps, which helps to maintain a set pressure in the feedwater header pipe, as shown in Figure 4.5. A pressure-activated valve (normally closed) can be installed on the return pipe as a safety measure to open if the pressure exceeds a set value (which may occur due to failure of the pump control system).

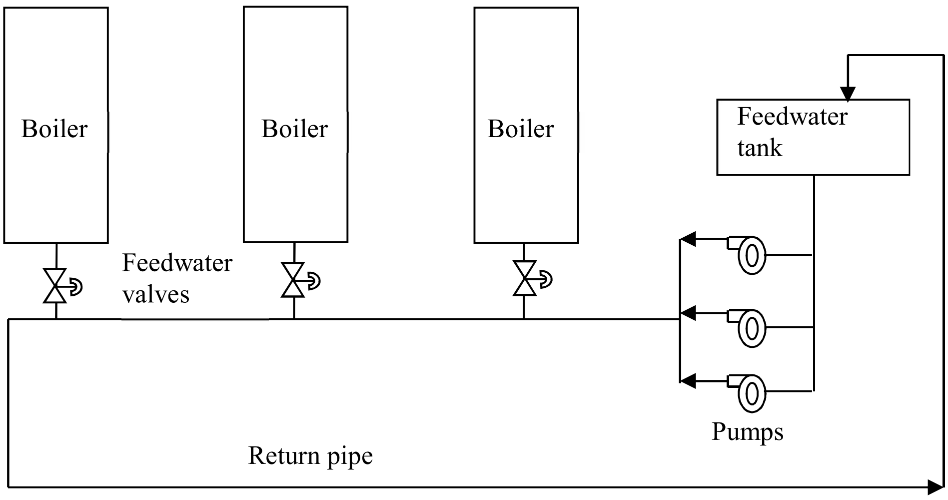


Figure 4.4 Feedwater pump arrangement for a multiple boiler operation

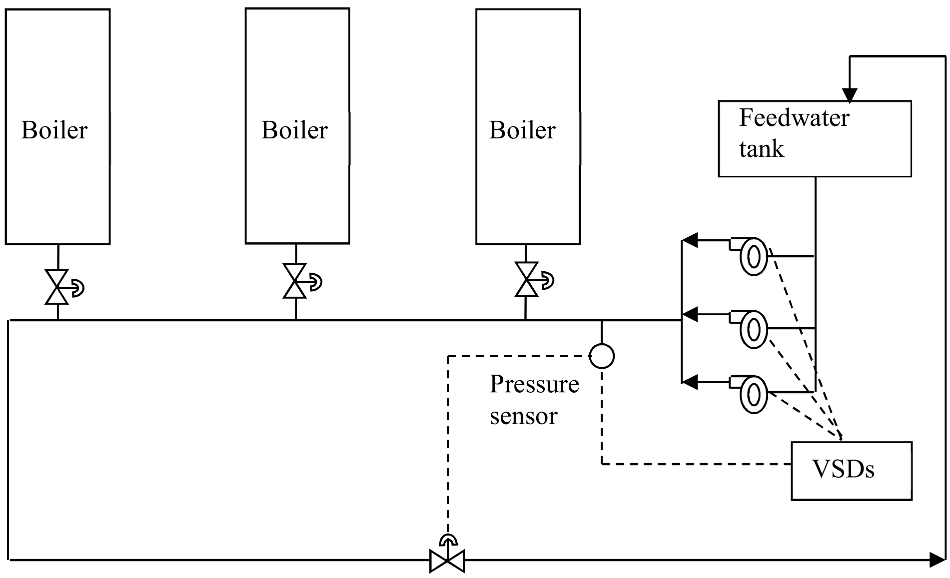


Figure 4.5 Suggested feedwater pump arrangement for a multiple boiler operation

Boiler fans used to create the draft necessary for combustion and carry the flue gases through the boiler normally operate at constant speed and dampers are used to control the air flow to match boiler load conditions. In such systems, when the boiler operates at part load, a damper throttles the air flow by inducing a resistance across the path of the air flow. As a result, the energy consumption of the fan does not reduce proportionately to the air flow. However, if a variable speed fan is used for this application (Figure 4.6), due to the cube law (fan power $\propto$ (air flow rate)^3), the reduction in fan energy consumption would be proportional to the third power of the load. Therefore, theoretically, if the load on the boiler reduces by 20%, the energy consumption of the fan will be reduced by about 50% (0.8^3).

The application of this energy saving measure depends on the load profile of the boiler. If the load is highly variable and results in the boiler operating at low loads for long periods of time, this is a good opportunity to incorporate a variable speed drive for the forced-draft or induced-draft fan of the boiler. Generally, such a retrofit is most economical for large boilers with modulating burners.

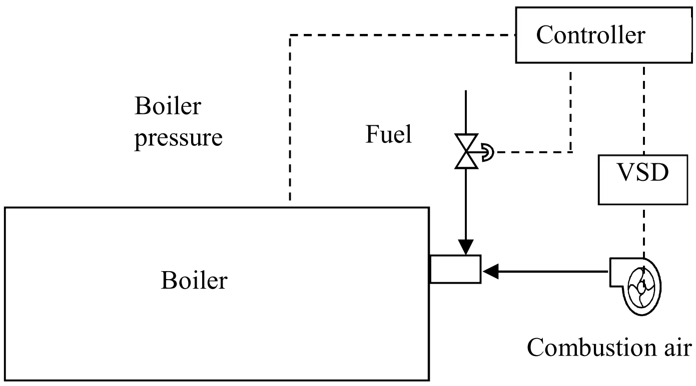


Figure 4.6 Application of VSD for boiler fan

Installation of variable speed drives for boiler fans may require consultation with the boiler manufacturer to ensure that the necessary control modifications (to keep the damper fully open while controlling the fan speed based on load) can achieve the proper air-to-fuel ratio at different load conditions.

Example 4.3

The operating loading and associated forced-draft fan power consumption of a boiler is given in Table 4.1.

Boiler loading	Operating hours a	Fan motor power
100%	2	22
80%	4	21
60%	10	19
40%	8	16

Table 4.1 Boiler Operating Data for Example 4.3

The boiler users a damper system to control the airflow rate. Estimate the savings achievable by installing a VSD to control the fan capacity based on boiler load.

Solution

The energy savings that can be achieved by installing a VSD can be estimated as shown in Table 4.2.

Boiler loading A	Operating hours a day B	Fan motor power with damper (kW)	Fan motor power with VSD (kW) $D = A^3 \times 22$	Power saving (kW)	Energy savings (kWh)
100%	2	22	22	0	0
80%	4	21	11	10	40
60%	10	19	5	14	140
40%	8	16	1.4	14.6	116.8
				Total	296.8

Table 4.2 Estimate of Savings for Example 4.3

Based on Table 4.2, the total savings is 296.8 kWh a day. This value can be multiplied by the number of operating days a year and the electricity tariff to calculate the annual cost savings.

4.4 Standby losses

Standby losses take place when a boiler is not firing and the hot surfaces inside the boiler lose heat to colder air circulating inside it. Such air circulation can take place due to natural convection and purging.

Losses due to natural convection occur when the air in the boiler gets heated (by the hot surfaces) making it lighter causing it to move up the stack, circulating cold air through the boiler. This can be avoided if dampers are installed to prevent the circulation of air when the boiler is not being fired.

Purging losses take place when the boiler combustion space is purged by the fan before firing the burners to ensure that there is only air present (to prevent possible explosions). Some burner systems also follow a purging cycle when firing stops. Losses due to purging can be reduced by minimising the on-off cycle of the burner system. This can be achieved by using burners that have a high turndown ratio (ratio of maximum heat output to the minimum heat output of a burner) to enable the burner to function at low loads without switching off the flame (compared to normal burners which have to be switched off at low loads and later switched on back again).

4.5 Minimising conduction and radiation losses

When a steam system is in operation, the surface temperature of boilers, auxiliary equipment and distribution piping become much hotter than the surrounding areas. Therefore, they lose heat by radiation and conduction. The amount of heat lost depends on the surface temperature of the hot surface, which in turn depends on the insulation (thickness, thermal conductivity and condition). To minimise heat loss, all hot surfaces should be insulated with material having sufficient resistance to heat transfer. Further, the insulation should be of adequate thickness and should be in good condition.

As stated earlier, a 10 T/hour boiler will have a loss of about 1% due to radiation and convection when operating at full load. Since the radiation and convective losses remain the same irrespective of boiler loading, 1% losses at full load can increase to 2% at 50% loading and 4.5% at 25% loading. Some typical values for radiation and convection losses are shown in Table 4.3.

Boiler capacity (T/hr)	Boiler loading			
	100%	**75%**	**50%**	**25%**
5	1.5	2	3	8
10	1	1.5	2	4.5
15	0.9	1.25	1.75	4
25	0.8	0.9	1.5	3

Table 4.3 Estimated radiation and convection losses for boilers

4.6 Heat recovery from flue gas

A significant amount of heat energy is lost through flue gases as all the heat produced by the burning fuel cannot be transferred to the water or steam in the boiler. As the temperature of the flue gas leaving a boiler typically ranges from 150 to 250°C, about 10 to 20% of the heat energy is lost through it.

Therefore, recovering part of the heat from flue gas can help to improve the efficiency of the boiler. Heat can be recovered from the flue gas by passing it through a heat exchanger (commonly called an economiser) installed after the boiler, as shown in Figure 4.7. The recovered heat can be used to preheat boiler feedwater, combustion air, or for other applications. The amount of heat recovered depends on the flue gas temperature and the temperature of the fluid to be heated.

One of the major problems associated with flue gas heat recovery is corrosion due to acid condensation. Acid condensation takes place when the flue gas is cooled below its acid dew point. The sulphur in the fuel combines with water to form sulphuric acid, which is corrosive. Therefore, the temperature of the flue gas needs to be maintained well above the acid dew point to prevent corrosion unless a heat recovery system specially designed to withstand acid corrosion is used.

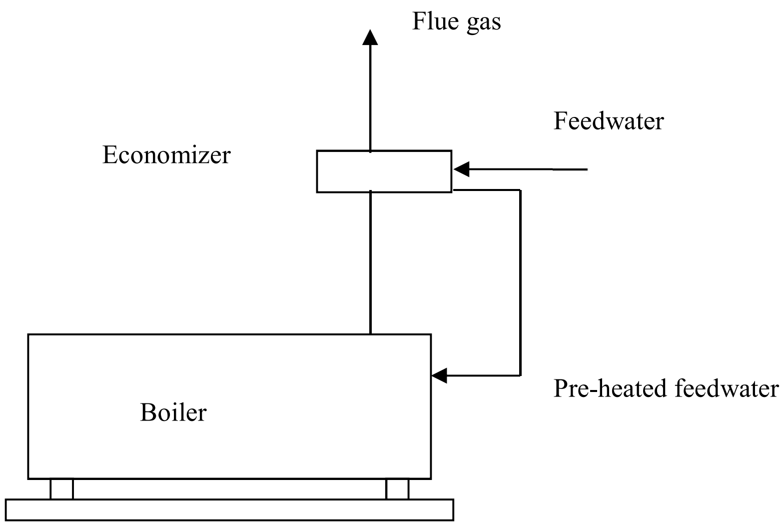


Figure 4.7 Arrangement of a typical economizer

The acid dew point depends on the sulphur content of the fuel. Some typical values are given in the Table below.

Fuel	Acid dew point temperature (°C)	Allowable exit stack temperature (°C)
Natural gas	66	120
Light oil	82	135
Low sulfur oil	93	150
High sulfur oil	110	160

Table 4.4 Acid dew point temperature for some common fuels

The feasibility of installing a heat recovery system for flue gas depends on factors such as by how much the stack temperature can be reduced, the inlet temperature of the fluid to be heated, and the operating hours of the boiler. Generally, the possible reduction in flue gas temperature should be at least 25°C to 30°C to make it economically viable to install a heat recovery system.

Since economisers induce extra pressure losses on the flue gas and the liquid being heated, care should be taken to ensure that the combustion fan and the pump for the liquid being heated have adequate capacity to overcome these losses.

Example 4.4

A diesel-fired boiler operates at a flue gas temperature of 220°C. Compute the energy savings achievable by installing an economiser to preheat feedwater. Assume that the air-to-fuel ratio is 15:1 and that the stack temperature has to be maintained above the allowable exit stack temperature of 150°C. The monthly diesel consumption is 330,000 lit/month. The density of diesel is 900 kg/m^3 and the specific heat capacity of flue gas is 1.1 kJ/kg.K

Solution

Diesel consumption rate = 333,000 lit/month = 333 m^3/month
= 333 x 900 kg/month
= (333 x 900)/(24 x 30) = 416.25 kg/hour

Air fuel ratio = 15:1

Air flow rate = 15 x 416.25 = 6243.75 kg/hour

Total mass flow rate of flue gas = 416.25 + 6243.75 = 6660 kg/hour = 6660/3600 = 1.85 kg/s

Heat recovered, Q = m x Cp x ΔT

where,

m = mass flow rate of flue gas (kg/s)

Cp = specific heat capacity of flue gas (kJ/kg.K)

ΔT = reduction in temperature of flue gas (K)

Heat recovered, Q = 1.85 x 1.1 x (220 – 150) = 142.45 kW

4.7 Flash steam recovery

The temperature of boiling water at high pressure will be at a temperature higher than the boiling point of water at atmospheric pressure. As can be seen from the sample steam table (Table 1.5), boiling water for instance at 6 bar will be at a temperature of 158.8°C which is much higher than the boiling point of 100°C at atmospheric pressure. Therefore, when this high-pressure water is released to atmospheric pressure at a steam trap or as blowdown (explained later), some of the water will flash back into steam. This flash steam, if produced in large quantities, can be used as low pressure steam or condensed back into feedwater.

Example 4.5

0.25 kg/s of condensate at 8 bar (absolute) is released through a steam trap. Compute the percentage and amount of flash steam produced.

Solution

Enthalpy of water at 8 bar = 721 kJ/kg (from Table 1.5)

Enthalpy of water at atmospheric pressure = 419 kJ/kg

Excess energy = 721 – 419 = 302 kJ/kg

Enthalpy of evaporation at atmospheric pressure = 2257 kJ/kg

Percentage of flash steam produced = (excess energy / enthalpy of evaporation) x 100%

= (302 / 2257) x 100%

= 13%

Amount of flash steam produced = 0.25 x 0.13 = 0.0325 kg/s

4.8 Automatic blowdown control

Boiler blowdown is part of the water treatment process and involves removal of sludge and solids from the boiler. Make-up water used for boilers contains various impurities. As water is converted to steam, the concentration of the impurities that remain in the boiler increases. If this concentration is allowed to increase, it will lead to accelerated corrosion, scaling, and fouling of the heat-transfer surfaces of the boiler. Therefore, it is necessary to remove part of the concentrated water from the boiler and replace it with fresh water.

The quality of water in the boiler is normally assessed using the TDS (total dissolved solids) level measured in ppm (parts per million). For typical two-pass and three-pass fire tube boilers, the allowable TDS level is 3000 to 3500 ppm while for water tube boilers the allowable level is about 1500 ppm.

The blowdown rate depends on the rate of steam production, TDS of the feedwater and the allowable maximum TDS and can be expressed as follows:

Blowdown rate (kg/hr) = $[S \times F_T] / [M_T - F_T]$ (4.1)

where,

S = steam production rate (kg/hr)

F_T = Feedwater TDS (ppm)

M_T = maximum allowable TDS (ppm)

Example 4.6

The TDS level of the boiler feedwater is 200 ppm. Estimate the blowdown rate for an 8000 kg/hr boiler if the allowable maximum TDS is 3000 ppm.

Solution

Blowdown rate (kg/hr) = $[S \times F_T] / [M_T - F_T]$

$= [8000 \times 200] / [3000 - 200]$

$= 571.4$ kg/hr

Boiler blowdown can be intermittent, where a fixed quantity of water is drained periodically, or continuous, where a small amount of the water is removed continuously to maintain the quality of water within acceptable limits.

Blowdown involves discharge of water at steam temperature, which has to be replaced by an equivalent amount of cold water. Energy losses resulting from blowdown can be minimised by installing automatic blowdown systems to reduce the amount of blowdown and recovering heat from blowdown (Figure 4.8).

Automatic blowdown control systems monitor the pH and conductivity of the boiler water and allow blowdown only when required to maintain an acceptable level of water quality. Automatic blowdown systems are preferred as they can maintain the TDS level close to the maximum allowable value. Manual blowdown (normally every eight hours) on the other hand, requires the TDS level to be reduced to a much lower value during the actual blowdown to ensure the TDS level does not exceed the maximum allowable value until the next blowdown cycle, resulting in a much lower average TDS level. A comparison of TDS variation for manual and automatic blowdown control is shown in Figure 4.9.

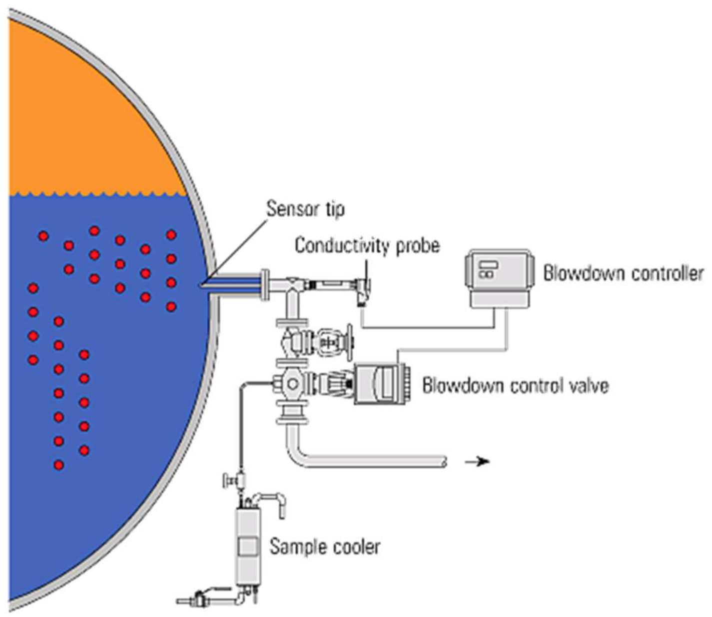


Figure 4.8 Typical arrangement of automatic blowdown control (illustration taken from the Spirax Sarco website 'Steam Engineering Tutorials' at http://www.spiraxsarco.com)

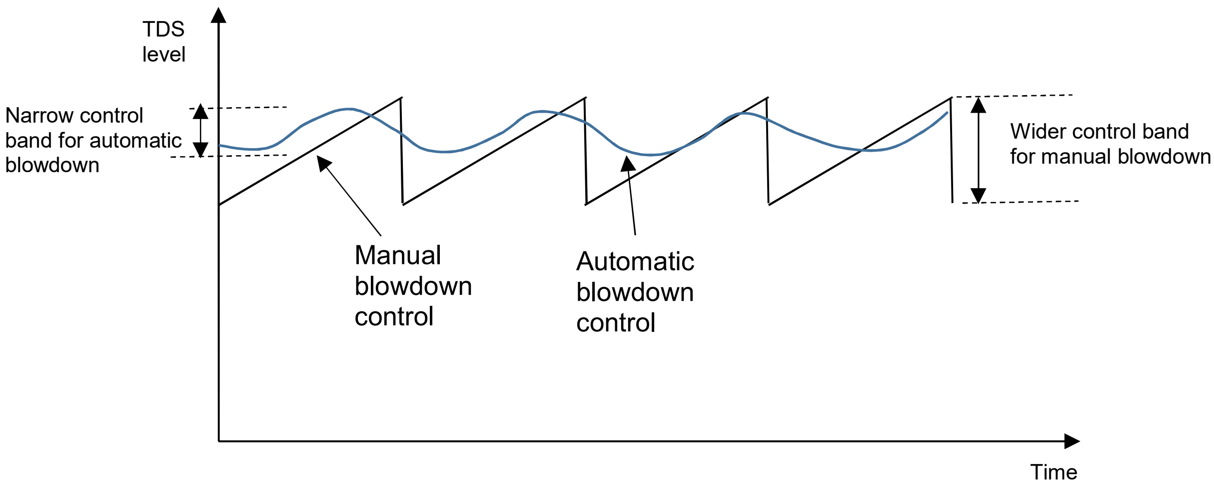


Figure 4.9 Comparison of TDS level for automatic and manual blowdown

Example 4.7

An 8000 kg/hr boiler uses manual blowdown. The average TDS level is maintained at 2200 ppm. Compute the reduction in blowdown rate that can be achieved by installing an automatic TDS control system to maintain the average TDS level at 3500 ppm. The TDS level of the feedwater is 250 ppm.

Solution

Blowdown rate (kg/hr) = [S x F_T] / [M_T – F_T]

Automatic blowdown rate = [8000 x 250] / [3500 – 250]
= 615.4 kg/hr

Manual blowdown rate = [8000 x 250] / [2200 – 250]
= 1025.6 kg/hr

Amount of blowdown that can be reduced = 1025.6 – 615.4 = 410.2 kg/hr

4.9 Heat recovery from blowdown

Heat recovery from blowdown involves using a heat exchanger to preheat cold make-up water using the blowdown. Such systems are feasible for boilers that operate most of the year using at least 5% of make-up water.

Flash steam (explained earlier) also can be recovered from blowdown and a typical arrangement is shown in Figure 4.10.

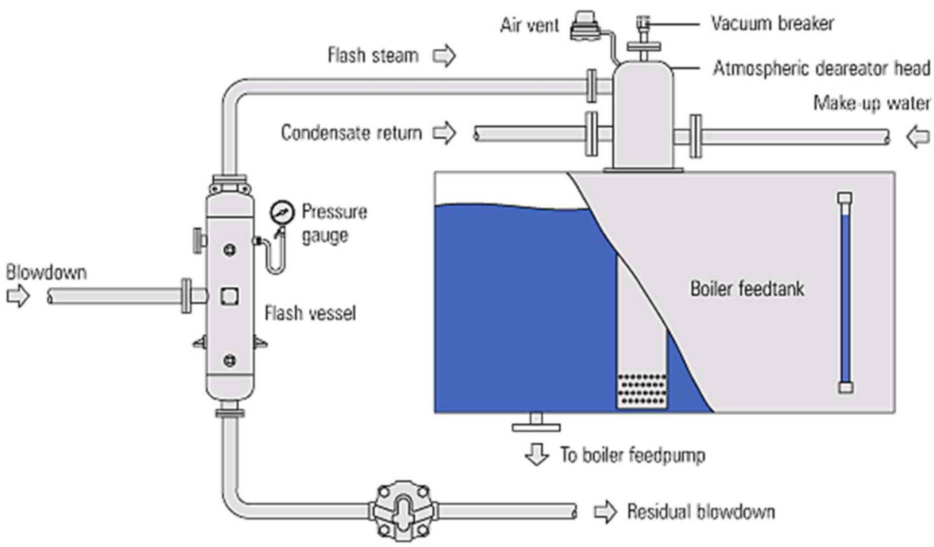


Figure 4.10 Typical blowdown flash steam recovery system (illustration taken from the Spirax Sarco website 'Steam Engineering Tutorials' at http://www.spiraxsarco.com)

Example 4.8

The TDS level of a 10000 kg/hr boiler is maintained at 2500 ppm using an automatic TDS control system. The TDS of the feedwater is 200 ppm and the boiler operating pressure is 10 bar (absolute). Compute the amount of heat energy in the blowdown and the amount of flash steam that can be recovered from the blowdown.

Solution

Blowdown rate (kg/hr) = $[S \times F_T] / [M_T - F_T]$

= [10000 x 200] / [2500 – 200]

= 869.6 kg/hr

= 0.24 kg/s

Enthalpy of blowdown at 10 bar, h_f = 763 kJ/kg (Table 1.5)

Heat energy in the blowdown = 0.24 (kg/s) x 763 (kJ/kg) = 183 kW

Enthalpy of water at atmospheric pressure = 419 kJ/kg

Excess energy = 763 – 419 = 344 kJ/kg

Enthalpy of evaporation at atmospheric pressure = 2257 kJ/kg

Percentage of flash steam = (excess energy / enthalpy of evaporation) x 100%

= (344 / 2257) x 100%

= 15%

Amount of flash steam = 0.24 x 0.15 = 0.036 kg/s

4.10 Condensate recovery

In most steam systems, steam is used mainly for heating by extracting its latent heat (h_{fg}). The resulting condensate is at steam temperature and still contains a considerable amount of heat energy (h_f). From Table 1.5, if steam is used at 8 bar (absolute), then the condensate enthalpy (h_f) will be 721 kJ/kg which is 26% of the total enthalpy of steam (2769 kJ/kg) and will be lost if condensate is not returned to the system. Therefore, returning condensate to the boiler feedwater tank will result in significant fuel energy savings.

Since condensate is distilled water, it is ideal for use as boiler feedwater. Therefore, condensate recovery helps to reduce water consumption (water cost), water treatment cost, and blowdown.

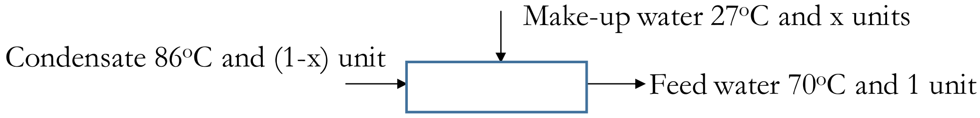
Usually, a low feedwater temperature or high make-up water flow indicates that less condensate is recovered. If the makeup water flow is metered, in applications that don't consume live steam (such as open sparge coils and direct steam injection systems), the difference between the amount of steam produced and make-up water flow will give an indication of the amount of condensate that is not recovered.

The amount of condensate recovered can be estimated by performing a heat balance for the feedwater, condensate water and make-up water as illustrated in Example 4.9.

Example 4.9

Feedwater is provided to a boiler at 70°C from the feedwater tank. The temperature of condensate water returning to the tank is 86°C, while the temperature of make-up water is 27°C. Estimate the amount of condensate that is recovered.

Solution



Performing a heat balance yields, $27x + (1-x)86 = 70$

Hence, $x = 0.27$ (27% is make-up water)

Therefore, 73% of the condensate is recovered.

The amount of heat that can be recovered from condensate can also be computed and is illustrated in Example 4.10.

Example 4.10

In a process heating application, 1 kg/s of a liquid that has a specific heat capacity of 3 kJ/kg.K is heated from 50°C to 90°C using steam at 6 bar. Estimate the amount of condensate produced at the heat exchanger and the amount of heat energy that can be saved if the condensate is returned to the boiler.

Solution

Amount of heat required for the application, $Q = m \times Cp \times \Delta T$

where,

m = mass flow rate of liquid
Cp = specific heat capacity of liquid
ΔT = increase in temperature of the liquid

Q = 1 (kg/s) x 3 (kJ/kg.K) x [90 – 50] = 120 kW

Latent heat of steam at 6 bar, h_{fg} = 2087 kJ/kg (Table 1.5)

Amount of steam required = 120 / 2087 = 0.057 kg/s

Enthalpy of condensate, h_f = 670 kJ/kg

Amount of heat energy in condensate = 670 (kJ/kg) x 0.057 (kg/s) = 38 kW

4.11 Faulty steam traps

The operation of steam traps is important because if they fail to operate properly and allow live steam to pass through them from the steam side to the condensate side, they result in obvious loss of energy. In addition, if the traps are unable to remove air at start-up times or if they are unable to remove condensate at a sufficient rate, the resulting reduced capacity and longer periods to heat up would also result in energy wastage.

Over time, the internal parts of steam traps wear out and result in failure to open and close properly. While an open trap would result in loss of live steam, a closed trap could result in loss of heat transfer area and water hammering, because part of the heat exchanger will be filed with water (since condensate is unable to be discharged through the steam trap). Water hammering can eventually result in damage to valves and other components in steam systems, which could result in steam leaks.

However, one of the main problems in maintaining steam traps is identifying defects in them. Often, the condensate released by traps is diverted to a condensate collecting tank, making it hard to spot leaking traps. Further, it is sometimes hard to distinguish between leaking steam and flash steam at the steam traps.

One way of identifying steam leaks from traps that are connected by piping to condensate tanks is to install sight glasses after the traps to provide a visual indication of leaks. Ultrasound leak detectors can also be used to detect leaking traps. Further, traps should be periodically inspected and repaired or replaced to ensure that they are in good working condition. In addition, the correct type of steam trap should be selected for each application.

4.12 Rectifying steam leaks

Steam leakage occurs from pipes, flanges, valves, connections, traps, and process equipment and can be substantial for some steam distribution systems. The amount of steam leaking from various openings depends on the size of the opening and the system pressure. Figure 4.11 shows the approximate steam leak rates at different operating pressures for various leaking hole sizes.

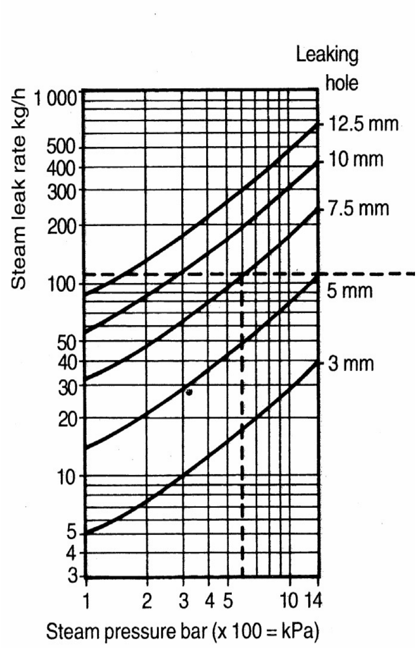


Figure 4.11 Steam losses through leaks (illustration taken from the Spirax Sarco website 'Steam Engineering Tutorials' at http://www.spiraxsarco.com)

Once the leakage rate is known, the amount of energy lost can be computed using the enthalpy data, while the amount of fuel wasted can be estimated using the boiler efficiency as illustrated in Example 4.11.

Example 4.11

The total steam leak rate for a plant is estimated to be 100 kg/hr. If the steam pressure is 7 bar (absolute), compute the amount of energy wasted due to the steam leaks. As the fuel, the boiler uses natural gas which has a calorific value of 55,000 kJ/kg and the operating boiler efficiency is 88%. Estimate the reduction in fuel used by the boiler that can be achieved if all the steam leaks in the plant are rectified.

Solution

Enthalpy of steam at 7 bar, h_g = 2764 kJ/kg (Table 1.5)

Total steam leak rate = 100 kg/hr = 0.028 kg/s

Amount of heat energy wasted = 0.028 (kg/s) x 2764 (kJ/kg) = 77.4 kW

Amount of input heat energy wasted = 77.4 / 0.88 = 87.95 kW

Amount of fuel wasted = 87.95 (kW) / 55,000 (kJ/kg) = 0.0016 kg/s = 5.76 kg/hr

4.13 Feedwater tank

The feedwater tank is a very important part of any steam system. It provides a reservoir of returned condensate and fresh make-up water for the boilers. The feedwater tank gives a good indication of the system performance. Excessive feedwater temperature may indicate that some traps may be passing live steam, while a high make-up water flow may indicate that some condensate is not being returned to the tank.

Feedwater is normally hot due to returned condensate and recovery of heat from other sources. Therefore, the tank should be elevated to prevent hot water being flashed off as steam at the feedwater pump inlet, in order to prevent cavitation.

Since the feedwater tank is hot, steps should be taken to minimise heat losses from the tank. Other than insulating the tank, the top of the tank should be covered since a great amount of losses usually take place at the water surface.

4.14 Fouling and scaling in boilers

Fouling, scaling, and soot build-up on heat transfer surfaces of boilers acts as an insulator and leads to reduced heat transfer. This results in lower heat transfer to water in the boiler and higher flue gas temperature. If at the same load conditions and same excess air setting the flue gas temperature increases with time, this is a good indication of increased resistance to heat transfer in the boiler. Data for energy wastage due to scaling is shown in Table 4.5.

Scale thickness (mm)	Energy wastage (%)
0.4	1
0.8	2
1.2	3
1.6	4

Table 4.5 Estimated energy wastage due to scaling

When this occurs, the boiler heat-transfer surface should be cleaned. On the fire-side surfaces should be cleaned of soot, while on the water-side scaling and fouling should be removed. For boilers using gas and light oil, it is generally sufficient to clean the fire-side surfaces once a year. However, for boilers using heavy oil, cleaning may need to be done several times a year.

In addition, preventive steps should also be taken. For scaling, as it is caused by inadequate water treatment, steps should be taken to improve water softening and maintaining a lower TDS level. For soot build up, which is normally due to defective burner or insufficient air for combustion, steps should be taken to repair or retune the combustion system.

4.15 Fuel switching

Most modern boilers have capability to switch between fuels (e.g. oil and gas). Since cost of fuel can change due to seasonal and other reasons, fuel switching can help to minimise operating cost.

When switching between fuels. It is necessary to consider not only cost of fuel but also factors such as heat content (calorific value) and combustion efficiency.

Example 4.12

A boiler requires 0.12 l/s of No. 2 oil when operating at 80% efficiency. The same boiler requires 0.13 m^3/s of natural gas when operating at the same efficiency. If the cost of No.2 oil is \$0.30 /litre and the cost of natural gas is \$0.18 /m^3, calculate the hourly operating cost for using oil and gas.

Solution

Cost of using No. 2 oil is:

0.12 l/s x 3600 s/hour x \$0.30 /litre = \$129.60 / hour

Cost of using natural gas is:

0.13 m^3/s x 3600 s/hour x \$0.18 /m^3 = \$84.24 / hour

Therefore, it is cheaper to run the boiler on natural gas than on No. 2 oil.

4.16 Using heat pumps

Often steam is used to produce hot water for industrial applications. Depending on the required temperature of hot water, it may be possible to use a heat pump for such an application.

Heat pumps operate on the vapour compression cycle. They are similar to cooling systems and have an evaporator, condenser, compressor and an expansion valve as shown in Figure 4.12. The main purpose of a heat pump is to extract the heat produced in the condenser. The cooling produced at the evaporator can be used for cooling applications or discarded to the environment.

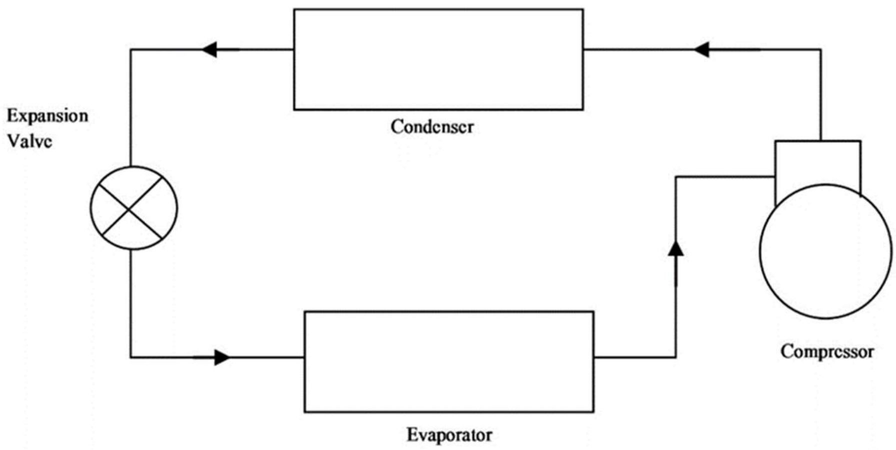


Figure 4.12 Vapour compression cycle used in hot water heat pumps

Most heat pumps are able to produce hot water up to a temperature of about 60°C. They are normally more energy efficient and have a Coefficient of Performance (COP) of 4 to 5. COP refers to the heat produced in kW to the electrical power input to the compressor motor in kW. Therefore, heat pumps are able to produce 4 to 5 kW of heating using 1 kW of electrical power.

Example 4.13

A diesel fired boiler is used to produce hot water at 60°C. The average daily hot water demand is 5,000 kWh. The boiler operates 365 days a year. The boiler efficiency is 80%.

Compute the reduction in energy consumption (kWh) that can be achieved if a heat pump with a COP of 4.0 is used for the same application.

If the calorific value of diesel is 12 kWh/litre and the cost of diesel and electricity are $1.5/litre and $0.10/kWh respectively, compute the annual cost savings that can be achieved.

Solution

Daily energy usage for diesel = 5,000 / 0.8 = 6,250 kWh

Daily electrical energy usage for a heat pump with COP of 4.0
= 5,000 / 4.0 = 1,250 kWh

Reduction in energy usage = (6,250 – 1,250) kWh/day
= 5,000 kWh/day
= 5,000 x 365 kWh/year
= 1,825,000 kWh/year

Diesel usage = 6,250 / 12 litres/day
= 520.83 litres/day
= 520.83 x 365 litres/year
= 190,103 litres/year

Cost of diesel = 190,103 x $1.5
= $285,154 /year

Cost of electricity = 1,250 x $0.1 /day
= $125 /day
= $45,625 /year

Cost savings = $(285,154 – 45,625)
= $239,529 /year

Summary

This chapter described the most common energy saving measures that can be implemented to optimise the operation and energy efficiency of boilers and hot water systems. Various examples were used to illustrate the energy savings achievable for each energy saving measure.

References

1. Improving Steam System Performance: A Source Book for Industry. US Department of Energy, Industrial Technologies Program, Washington DC, 2012
2. Jayamaha, Lal, Energy-Efficient Building Systems, Green Strategies for Operations and Maintenance, McGraw-Hill, New York, 2006.
3. Jayamaha, Lal, Energy-Efficient Industrial Systems, Evaluation and Implementation, McGraw-Hill, New York, 2016.
4. The Steam and Condensate Loop, Spirax Sarco Limited, England 2008.

5.0 ENERGY AUDIT OF STEAM SYSTEMS

Boilers and steam systems are significant energy users in various manufacturing plants, ranging from petrochemical to food manufacturing. Normally, they account for more than 80% of the energy consumed in such plants.

From the previous chapter of this reference manual, it is clear that many measures are available to optimise the operation of boilers and steam systems to minimise energy usage.

However, before optimising a boiler or steam system, it is necessary to first assess its performance so that suitable improvements can be identified and implemented. Such a system assessment is usually called an "Energy Audit". This chapter describes the key energy performance indicators (EnPIs) used for assessing the performance of boilers and steam systems followed by how the required parameters are to be obtained.

Learning Outcomes:

The main learning outcomes from this chapter are to understand:

1. The key EnPIs for boilers and steam systems
2. The parameters required
3. Measurements required
4. Case study of a boiler energy audit

The main EnPIs used for assessing the performance of boilers and steam systems are explained in the subsequent sections of this chapter.

5.1 Boiler overall efficiency

The overall efficiency of the boiler accounts for all the losses, which include individual losses due to combustion, blowdown, convection, radiation and exhaust.

Overall boiler efficiency = $\left(\frac{\text{Heat output}}{\text{Heat input}}\right)$ x 100

The overall boiler efficiency can be measured using either the direct or indirect method as explained in chapter 2. Since the direct method is easier to follow and provides a good estimate of the boiler operating efficiency with sufficient accuracy, it is the most common method used in energy audits.

$$\text{Boiler efficiency } (\eta) = \frac{\text{Enthalpy of steam - Enthalpy of feedwater}}{\text{Heat released in boiler}}$$

The parameters required for the above computation are summarised in Table 5.1

Parameter	**How obtained**	**Typical instruments used**	**Units of measure**
Quantity of steam generated	Measurement using a steam flow meter	Vortex flow meter, thermal mass flow meter, orifice plate	kg/s
Specific enthalpy of saturated steam	From pressure of steam	Pressure gauge or pressure transmitter	bar
Quantity of fuel used	From measurement	Turbine flow meter, thermal mass flow meter	kg/s, l/s
Quantity of feedwater	Measurement using a liquid flow meter	Magnetic flow meter, ultrasonic flow meter	kg/s
Specific enthalpy of feedwater	From temperature of feedwater	RTD	^{o}C
Gross calorific value	From specifications / reference data	N/A	kJ/kg

Table 5.1 Parameters required for estimating boiler efficiency (direct method)

Since the feedwater temperature and pressure of steam produced don't normally vary during steady-state operation of a boiler, spot measurements or average values can be used to obtain average enthalpy values for feedwater and steam (from steam tables).

The amount of fuel used can be obtained from meters for liquid and gaseous fuel. In the absence of a fuel meter, liquid fuel usage can also be obtained by taking measurements of the tank level. When solid fuels are used, a weighing scale would need to be used.

The calorific value of the fuel can be obtained from the specifications provided by the supplier of the fuel, using reference data or laboratory measurement of a sample of fuel.

Therefore, the main measurements required are for the quantities of steam output and feedwater input. In some installations, instruments may be available to measure both parameters and this would provide the most accurate estimation of the boiler overall efficiency.

If only a steam flow meter is installed for the boiler, then an ultrasonic liquid flow meter can be installed to measure the feedwater flow rate during the audit.

However, if a steam flow meter is not available, since it would not be practical to install a temporary steam flow meter during the audit (flow meter installation will be intrusive), a temporary liquid flow meter can be used for measuring only the feedwater. In such a case, the steam output can be estimated by accounting for the boiler blowdown and subtracting it from the feedwater flow rate as shown below.

Under steady-state operation of a boiler,

$$m_s = m_f - m_{bd} \tag{5.1}$$

where,

m_s = steam mass flow rate

m_f = feedwater mass flow rate

m_{bd} = mass flow rate of blowdown

Since it is not practical to measure the mass flow rate of blowdown, it can be estimated using measurements of total dissolved solids (TDS) levels of the feedwater and boiler water as shown below.

$$B = [S \times F_T] / [M_T - F_T] \tag{5.2}$$

and

$$F = B + S \tag{5.3}$$

where,

B = blowdown mass flow rate (kg/hr)

S = steam production rate (kg/hr)

F = feedwater mass flow rate (kg/hr)

F_T = Feedwater TDS (ppm)

M_T = Boiler water TDS (ppm)

Rearranging equations (5.2) and (5.3), the steam flow rate can be expressed in terms of feedwater flow rate and the TDS values as shown below.

$$S = \frac{F}{\left(\frac{F_T}{M_T - F_T}\right) + 1} \qquad (5.4)$$

Therefore, if a steam flow meter is not available, the TDS values of the feedwater and boiler water would need to be measured. Electrical conductivity of water is directly related to the concentration of dissolved solids in the water. Since the ability for water to conduct an electric current depends on the ions from the dissolved solids, TDS can be measured using a conductivity meter. Conductivity provides an approximate value of the TDS concentration, usually to within 10% accuracy.

5.2 Boiler combustion efficiency

As explained in chapter 2, combustion efficiency is an assessment of how close the actual combustion is to Stoichiometric combustion, where the oxygen provided is exactly sufficient for complete combustion of fuel.

It can be measured by sampling the exhaust flue gas to find the composition and temperature using a combustion analyser. Ideally, the combustion efficiency should be greater than 80%.

The flue gas temperature measured to assess the combustion efficiency can also be used to evaluate the potential for heat recovery from exhaust flue gas (if an economiser is not already installed). The measured exhaust temperature value can be compared with the minimum stack temperature recommended for the particular fuel in section 4.6. If there is sufficient potential to reduce the exhaust temperature without reaching the minimum recommended value, then an economiser can be considered.

5.3 Condensate recovery

Since it is not practically possible to directly measure the amount of condensate returned, one of the following options can be used to estimate it:

1. Use data from a meter used for make-up water (if available) as the make-up water will equal the amount of steam not returned as condensate
2. Performing a heat balance for the feedwater, condensate water and make-up water as illustrated below.

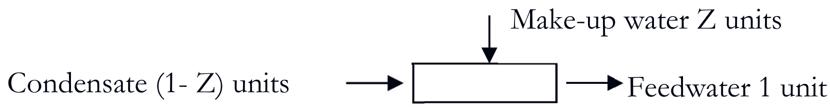


Figure 5.1 Diagram for heat balance to estimate condensate return

T_c = Condensate temperature

T_{mu} = make-up water temperature

T_f = feedwater temperature

Z = units of make-up water added per unit of feed water

Performing a heat balance yields, $T_{mu} \times Z + (1 - Z)\, T_c = 1 \times T_F$ (5.5)

Using measured values for the condensate temperature, make-up water temperature and feedwater temperature, equation (5.5) can be solved to obtain the value of Z, which provides a measure of the make-up water provided per unit of feedwater or unit of steam produced. A high value of "Z" indicates a low rate of condensate recovery.

5.4 Blowdown rate

The measured values or estimated values of the quantity of steam generated and the amount of feedwater provided can be used to quantify the amount of blowdown (the difference between the two values). This value together with the blowdown temperature (which is the saturation temperature of steam at the particular operating pressure) can be used to estimate the potential for heat recovery from blowdown.

5.5 Steam leaks

Visual inspection of steam distribution piping, valves, fittings and steam consuming process systems can help to identify steam leaks. Since steam leaks cannot usually be rectified during operation, leaks should be tagged so that they can be rectified during a system shut-down.

Similarly, observation of steam traps can also help to identify leaks. If steam traps are directly connected to condensate return piping, then it would not be possible to identify leaking traps by observation. In such cases, ultrasonic leak detection systems can be used to identify leaking steam traps.

5.6 Convective and radiative losses

During an energy audit, the surface temperature of the boiler outer surface at various locations should be measured to check the adequacy of the boiler insulation. An infrared temperature detector can be used for this purpose. Similarly, the surface temperature of the feedwater tank, condensate return tank, deaerator, steam headers, heat exchangers and other process equipment using steam should also be measured.

5.7 Case study of a boiler energy audit

Information on the system

In a manufacturing plant, three boilers are usually in operation. Diesel is used as the primary fuel for the boilers. The steam pressure is maintained between 8.9 to 9.0 bar. Steam produced by the boilers is supplied to the production areas and is mainly used for process heating. An economiser is installed to recover heat from the flue gas and pre-heat the make-up water before going into the feedwater tank. In addition, a condensate recovery system is used to recover condensate from the production equipment and to return back to the feedwater tank. The feedwater temperature is usually maintained at around 50 - 70 °C.

The boiler blowdown is manually performed by the operators based on the TDS indicator on the boiler panel. It is conducted once every 2 hours, to maintain the boiler TDS at 1700 ppm and below.

Data collection and analysis

The boiler efficiency was estimated using daily steam demand and fuel consumption data. Cumulative daily figures for above mentioned parameters were recorded for a period of six days. Using cumulative figures, daily consumption was estimated for the measurement period and these daily figures were used to calculate boiler efficiency. The daily steam flow rate is shown in Figure 5.2 and the average daily boiler operating performance including efficiency is shown in Figure 5.3.

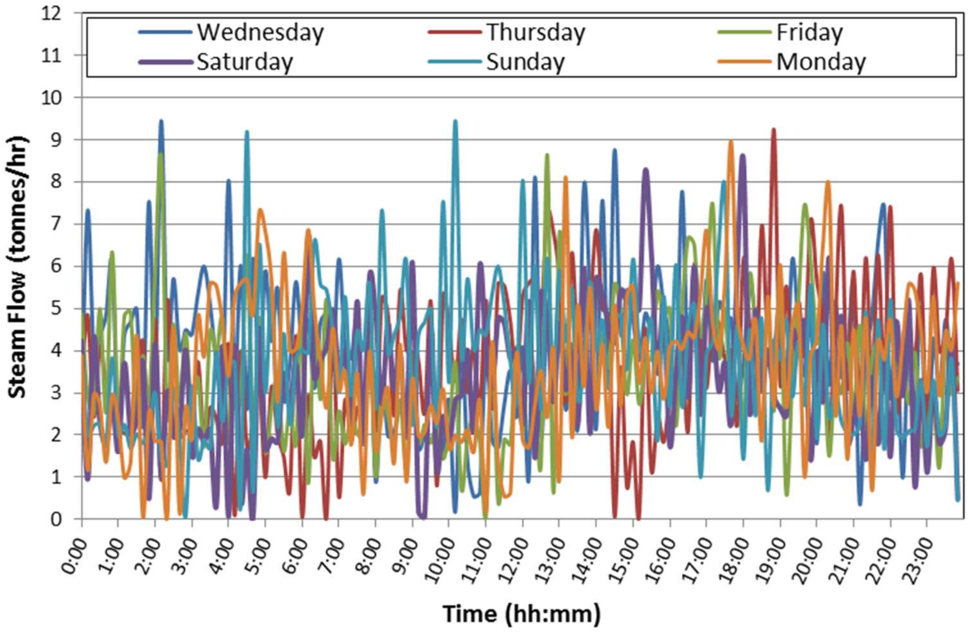


Figure 5.2 Steam flow profile

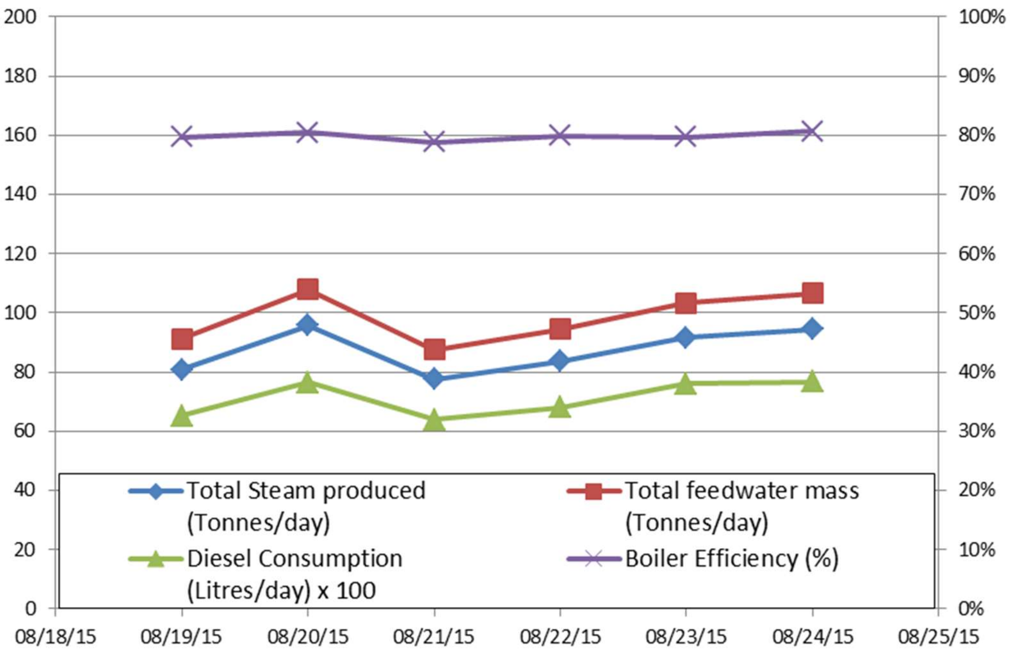


Figure 5.3 Daily average boiler performance data

Boiler operating performance is summarised below:

Descriptions	Design	Measured
Steam Pressure (bar)	9.0	8.9 – 9.0
Steam Temperature (°C)	-	180
Average diesel consumption (Litres/day)	-	7,110
Average feedwater flow rate (kg/hr)	-	4,104
Average feedwater temperature (°C)	-	63
Estimated blowdown rate (%)	-	13.3
Estimated steam generation (kg/hr)	7,200	3,638
Average boiler efficiency (%)	-	79
Condensate recovery (%)	-	88
Benchmark for boiler efficiency (%)	-	77 to 82

Table 5.2 Summary of boiler performance

Results of the boiler flue gas analysis is summarised in Table 5.3.

Measured Parameters for Flue Gas			
Description	Boiler 1	Boiler 2	Combined Stack
CO_2 (%)	9.4	9.6	9.5
CO (ppm)	81	80	80
O_2 (%)	8.2	8.0	8.1
Excess air (%)	64.5	62.0	63.7
Flue gas temperature (°C)	207	206	206
Temperature difference between flue gas and air intake (°C)	175	174	175
Combustion efficiency (%)	83.9	84.1	84.0
Benchmark combustion efficiency (%)	85 to 92	85 to 92	85 to 92

Table 5.3 Summary of Flue Gas

Main findings:

- Steam consumption was largely dependent on the production schedule during the week, and the steam flow varied from 1 to 9 T/hour (average 3.64 T/hour)
- Boiler outer surface temperature ranged from 50°C to 70°C which indicated that the current boiler insulation can be improved
- Flue gas temperature was about 206°C which was much higher than the minimum exit stack temperature of 135°C required for diesel (to prevent condensation)
- Concentration of Oxygen in the flue gas was about 8.1% which was much higher than the recommended value of about 2.2%. Higher concentration of oxygen in the flue gas indicated excess air flow rate was higher than requirements for combustion process.

Main recommendations

The main recommendations for the boiler system are listed below.

- Fine-tune the air to fuel ratio to improve combustion efficiency
- Replace the economiser to achieve a lower stack exit temperature
- Improve insulation of boiler outer surface to reduce heat losses
- Install an automatic blowdown system to reduce blowdown losses

Estimated savings

The approximate energy and cost savings for the recommended measures are tabulated below.

Recommended measure	Energy savings (kWh/year)	Cost savings* ($/year)
Fine-tuning of air to fuel ratio	750,000	37,500
Replacing economiser	400,000	20,000
Improving boiler insulation	300,000	15,000
Automatic blowdown system	200,000	10,000

* Diesel cost is taken as $0.52/litre

Table 5.4 Summary of savings for case study

Summary

This chapter described the key energy performance indicators (EnPIs) used for assessing the performance of boilers and steam systems followed by how the required parameters are to be obtained and how the EnPIs can be computed. Finally, a case study of a boiler energy audit was presented.

References

1. Improving Steam System Performance: A Source Book for Industry. US Department of Energy, Industrial Technologies Program, Washington DC, 2012.
2. ISO 50002:2014, Energy Audits – Requirements with guidance for use.
3. Jayamaha, Lal, Energy-Efficient Building Systems, Green Strategies for Operations and Maintenance, McGraw-Hill, New York, 2006.
4. Jayamaha, Lal, Energy-Efficient Industrial Systems, Evaluation and Implementation, McGraw-Hill, New York, 2016.
5. The Steam and Condensate Loop, Spirax Sarco Limited, England 2008.

6.0 COMPRESSED AIR SYSTEMS

Compressed air is widely used in industrial applications such as pneumatic tools, control systems, operation of machinery, and manufacturing processes. Compressed air is essential for the operation of industrial plants and is often considered the “fourth utility” after electricity, water, and steam.

Compressed air systems are highly energy intensive and account for a significant portion of energy consumed in many plants. Most of the energy used in compressed air systems is wasted as heat and only about 10% of the input energy is delivered in usable form as compressed air. The typical energy flow for a compressed air system is shown in Figure 6.1.

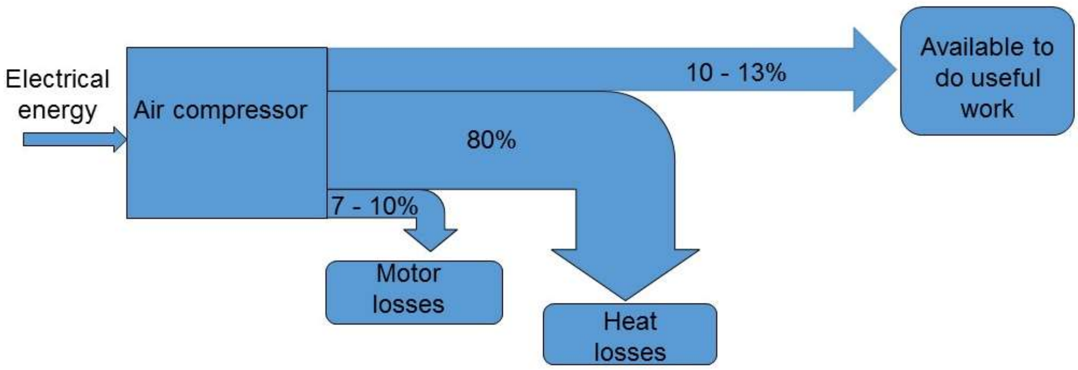


Figure 6.1 Energy flow for a compressed air system

Although only a small fraction of the input energy is available to perform useful work, part of this energy is also lost due to other reasons such as:

- Leaks
- Artificial demand
- Inappropriate usage
- Dynamic losses

Figure 6.2 shows how the 10 – 13% of the input energy available in the compressed air to perform useful work is normally utilised in industrial plants. Typically, only about half is used as the remaining amount is wasted due to leaks, losses and inappropriate usage.

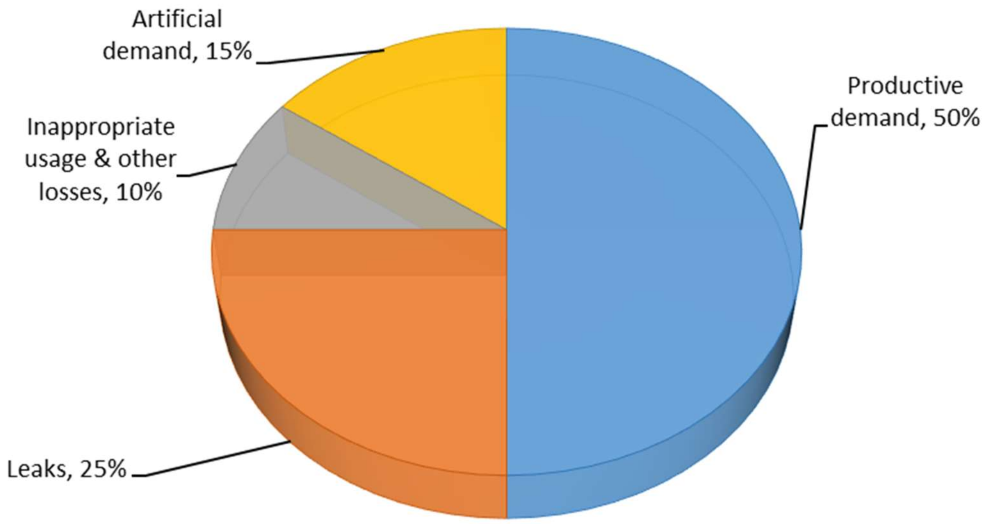


Figure 6.2 Typical breakdown of compressed air usage (Ref. 4)

Normally, the cost of operating an air compressor is more significant than the first cost (capital cost) of the actual air compressor. Figure 6.3 shows the approximate breakdown of the lifecycle cost for a compressor operating over a 10-year period. As can be seen from the figure, the operating energy cost is the highest single cost and accounts for about 80% of the total cost of owning and operating an air compressor. Hence, it is essential to optimise the design of compressed air systems so that they can operate efficiently to minimise operating costs.

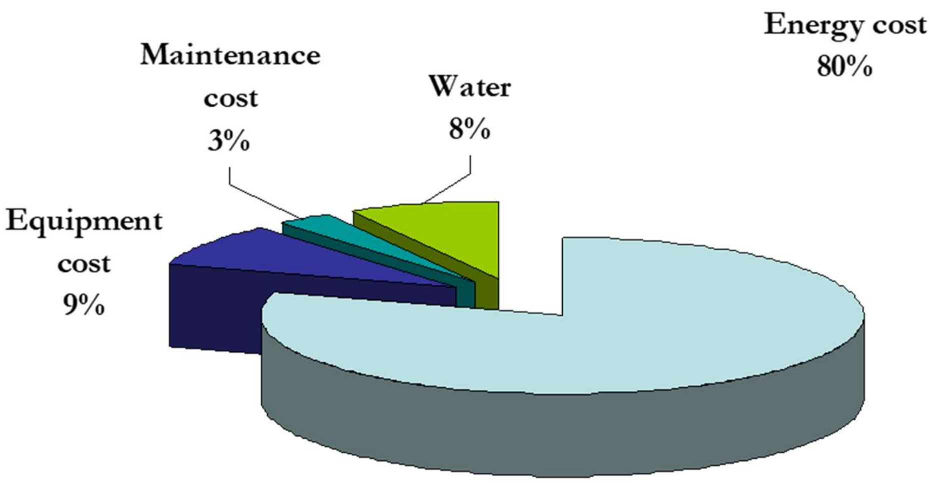


Figure 6.3 Typical lifecycle cost breakdown for an air compressor

Learning Outcomes:

The main learning outcomes from this chapter are to understand:

1. The different components used in compressed air systems
2. Standard conditions used for rating compressed air systems
3. The concept of free air delivery, utilisation factor and duty cycle
4. Types of compressors

6.1 Typical compressed air system components

Compressed air systems comprise air compressors, filters, dryers, storage receivers, distribution systems, and end users, as illustrated in Figure 6.4. The various components can be broadly divided into supply-side and demand-side components where the supply-side consists of air compressors, filters, dryers, and receivers, while the demand-side comprises the distribution system, storage, and the end users. Compressed air systems are often called “CDA” systems where the acronym stands for “compressed dry air”.

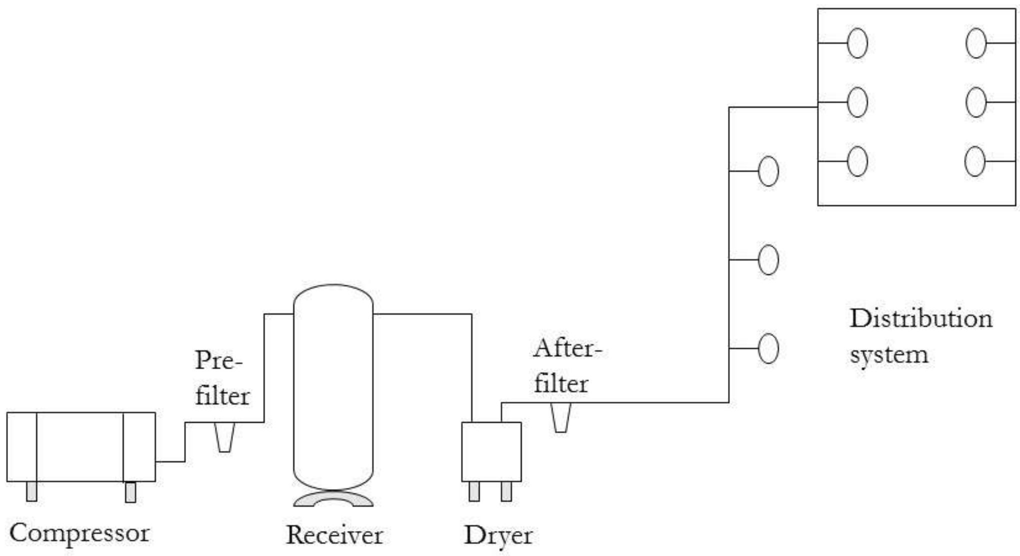


Figure 6.4 Arrangement of a typical compressed air system

The main objectives of a compressed air system are to:

- Provide required compressed air quantity to users
- Maintain a minimum pressure
- Minimise variation in system pressure
- Match supply to demand
- Maintain quality of compressed air to meet user requirements (moisture, oil and particulates).

The main components in a compressed air system are the compressors which produce compressed air. The various types of compressors and their operating characteristics are described later in this chapter. The other main components in compressed air systems such as the dryers and receivers are described in chapter 7.

6.2 Free Air Delivery

Free air delivery (FAD) is a measure of air taken into the compressor at atmospheric temperature and pressure. The capacity of compressors and most end-user equipment is normally rated in FAD, which is a common reference point.

If a compressor is rated 5 m^3/min free air at 9 bar, it means that 5 m^3/min of air is taken at the compressor inlet to produce compressed air at 9 bar pressure.

FAD can be computed from the volume flow rate at the discharge of the compressor (Figure 6.5) and using the ideal gas law as described below.

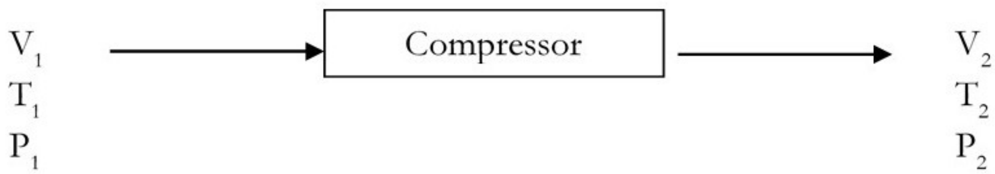


Figure 6.5 Diagram for computing FAD for a compressor

For an ideal gas:

$$\frac{P_1 V_1}{T_1} = \frac{P_2 V_2}{T_2} \qquad (6.1)$$

where,

P_1 and P_2 = the absolute pressures of air at the inlet and outlet of the compressor

V_1 and V_2 = volume of air at the inlet and outlet of the compressor

T_1 and T_2 = absolute temperatures of air at the inlet and outlet of the compressor

Therefore, for FAD:

$$V_1 = \frac{P_2 \text{ x } V_2 \text{ x } T_1}{T_2 \text{ x } P_1} \qquad (6.2)$$

Example 6.1

The volume flow rate at the discharge of the compressor is measured to be 0.05 m^3/s. The absolute discharge pressure and temperature are 6 bar and 35°C, respectively. Compute the corresponding FAD for the compressor, taking the ambient air temperature and absolute pressure to be 30°C and 1 bar, respectively.

Solution

Free air delivery:

$$V_1 = \frac{P_2 \times V_2 \times T_1}{T_2 \times P_1}$$

$= [0.05 \times 6 \times (273.15 + 30)] / [(273.15 + 35) \times 1]$

$= 0.295\ m^3/s$

6.3 Standard Conditions

Since the volume of air is dependent on the pressure, temperature, and relative humidity (measure of moisture in the air – discussed later), the parameter SCFM (standard cubic feet per minute), which refers to the volume flow rate at a set of standard conditions, is used to quantify the airflow rate.

Some of the commonly used “standard” conditions are summarised in Table 6.1.

Standard/usage	Temperature (°C)	Absolute pressure (bar)	Relative humidity (%)
CAGI & ASME	20	1.01	36
Europe & ISO	20	1.01	0

Table 6.1 Common Standard Conditions Used for Compressed Air

The airflow rate at the actual conditions can be converted into that at standard conditions and vice versa using the ideal gas law.

From Equation (6.1),

$$(ACFM \times P_a) / T_a = (SCFM \times P_s) / T_s \quad (6.3)$$

where,

ACFM = actual airflow rate (usually in ft^3/min, but can be in other units)

P_a = absolute pressure at actual conditions (bar)

T_a = absolute temperature at actual conditions (K)

SCFM = airflow rate at standard conditions (usually in ft^3/min, but can be in other units)

P_s = absolute pressure at standard conditions (bar)

T_s = absolute temperature at standard conditions (K).

Compressed air also contains moisture and equation (6.1) can be modified, as given below, to include the vapour pressure exerted by the moisture in the air, to account for moisture in the air.

$$\frac{(P_1 - P_{v1}) \times v_1}{T_1} = \frac{(P_2 - P_{v2}) \times v_2}{T_2} \qquad (6.4)$$

where,

P_1 and P_2 = absolute pressures at conditions 1 and 2

V_1 and V_2 = volume of air at conditions 1 and 2

T_1 and T_2 = absolute temperatures of air at conditions 1 and 2

P_{v1} and P_{v2} = vapour pressures at conditions 1 and 2

Since, relative humidity, RH, is the ratio of vapour pressure (P_v) to saturated vapour pressure (P_{sv}),

P_v = RH x P_{sv}

where P_{sv} = saturated vapour pressure.

Therefore, Equation (6.4) can be expressed using the RH at the two conditions 1 and 2 (RH_1 and RH_2), as

$$\frac{(P_1 - [RH_1 \times P_{sv1}]) \times v_1}{T_1} = \frac{(P_2 - [RH_2 \times P_{sv2}]) \times v_2}{T_2} \qquad (6.5)$$

where,

P_{sv1} and P_{sv2} = saturated vapour pressure of water at conditions 1 and 2

Hence, if the airflow rate is known at one set of conditions, the airflow rate at a different set of conditions can be computed using equation (6.5).

Values of saturated vapour pressure of water at different temperatures can be obtained from reference tables. Some values are given in Table 6.2.

Temperature (°C)	Saturated vapour pressure (kPa)
10	1.23
20	2.34
30	4.25
40	7.38
50	12.34
60	19.93
70	31.18
80	47.37
90	70.12
100	101.32

Table 6.2 Saturated vapour pressure of water

Example 6.2

0.1 m^3/s of air enters a compressor at 30°C and 1 bar absolute pressure and 70% RH. After the compressor, the pressure, temperature and RH are 8 bar (absolute), 40°C and 5%, respectively.

Compute the volume flow rate of air after the compressor.

Solution

Using equation (6.5),

$$\frac{(P_1 - [RH_1 \times P_{sv1}]) \times v_1}{T_1} = \frac{(P_2 - [RH_2 \times P_{sv2}]) \times v_2}{T_2}$$

P_1 = 1 bar $\quad$ P_2 = 8 bar

RH_1 = 70% $\quad$ RH_2 = 5%

P_{sv1} = 4.25 kPa $\quad$ P_{sv2} = 7.38 kPa

V_1 = 0.1 m^3/s

$T_1 = 303$ K $\qquad$ $T_2 = 308$ K

Therefore, $\frac{(1 - [0.7 \text{ x } 0\ .0425]) \text{ x } 0.1}{303} = \frac{(8 - [0.05 \text{ x } 0.0738]) \text{ x } v_2}{308}$

$V_2 = 0.012$ m^3/s

6.4 Utilisation Factor

The utilisation factor (also called the load factor) for a compressed air user is the ratio of the actual compressed air consumption to the maximum continuous consumption.

$$\text{Utilisation factor } \frac{\text{actual air consumption in 24 h}}{\text{maximum continuous air consumption in 24 h}} \qquad (6.6)$$

Example 6.3

The rated compressed air usage by a machine is 0.01 m^3/s. If the actual compressed air usage by this machine is measured to be 260 m^3/day, calculate the utilisation factor.

Solution

Utilisation factor = 260 / (0.01 x 3600 x 24) = 0.3

Utilisation factor is used to estimate the total compressed air requirements when designing a new plant. Although many industrial plants have a large number of pneumatic tools and users, most of them may not operate simultaneously. Therefore, to prevent oversizing of the system, utilisation factors for the various users can be used to compute the approximate average total compressed air consumption, as illustrated in Example 6.4.

Similarly, the utilisation factor for a compressor can be expressed as the ratio of the actual compressed air output in a period of time to the output when operating at maximum capacity for the same period of time.

Example 6.4

Compressed air requirements for a new system are summarised in Table 6.3. Estimate the FAD for a suitable compressor for this application.

Equipment type	A	B	C
Air demand (FAD)	5 CMH	3 CMH	12 CMH
Working pressure	3 bar	4 bar	6 bar
Quantity	2	1	2
Utilisation factor	50%	30%	20%

Table 6.3 Data for Example 6.4

Solution

The effective air demand for each type of equipment can be computed as follows:

Equipment A = 5 x 2 x 0.5 = 5 CMH

Equipment B = 3 x 1 x 0.3 = 0.9 CMH

Equipment C = 12 x 2 x 0.2 = 4.8 CMH

Total estimated demand = 5 + 0.9 + 4.8 = 10.7 CMH

6.5 Duty cycle

Duty cycle refers to the amount of time a compressor can be operated in a given period of time, at 100 psi, at a standard ambient temperature of 22°C.

Duty cycle is commonly expressed in a percentage format as:

$$\text{Duty cycle} = \frac{\text{compressor on time}}{\text{compressor on time+compressor off time}} \times 100\% \qquad (6.7)$$

For example, if a compressor is rated for 25% duty cycle, then the compressor can be operated for 10 minutes On and 30 minutes Off.

6.6 Types of Compressors

The two main categories of compressors are positive-displacement and dynamic, as illustrated in Figure 6.6. In positive-displacement compressors, a fixed volume of gas is compressed in a compression chamber where the volume of the gas is mechanically reduced. In such compressors, the gas flow is constant at a particular speed, irrespective of the discharge pressure. Typical positive-displacement compressors are reciprocating, screw, scroll, and rotary sliding vane, based on their respective operating characteristics.

In dynamic compressors, kinetic energy is imparted to a continuous flow of gas by a single rotor or multiple rotors operating at high speed. The imparted kinetic energy is later converted into potential energy in the discharge volute or diffuser of the compressor. Centrifugal and axial-flow compressors are dynamic compressors.

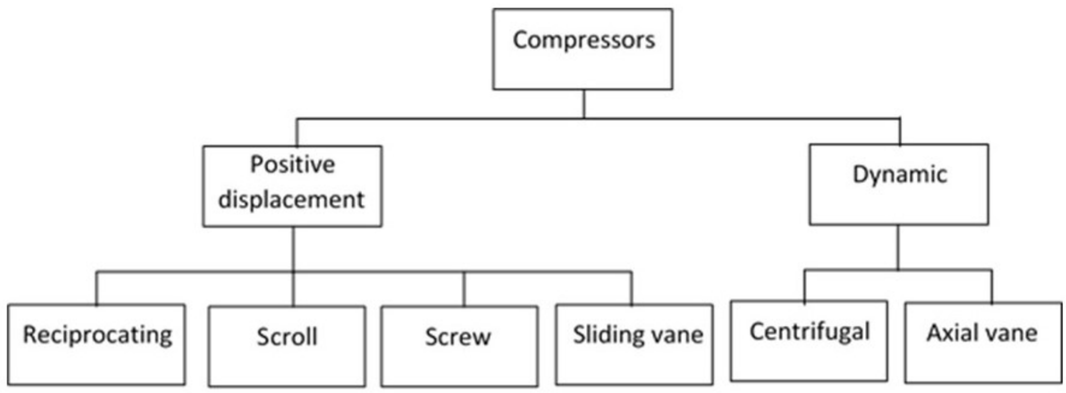


Figure 6.6 Different types of compressors

6.6.1 Reciprocating Compressors

Reciprocating compressors use pistons and connecting rods driven by a crankshaft (Figure 6.7). The crankshaft is driven by a motor. Compression is achieved by reducing the volume within the cylinder due to the movement of the piston. Normally, reciprocating compressors have a number of cylinders in one unit. For high-capacity applications, multistage units with multiple compressors are used. In such compressors, the gas is compressed to an intermediate pressure in the first stage, and then cooled to a lower temperature before being compressed further in the second stage (Figure 6.8).

Reciprocating compressors can be single acting where air is compressed only by one side of the piston (Figures 6.7 and 6.8) or double-acting where compression takes place on both sides of the piston as shown in Figure 6.9.

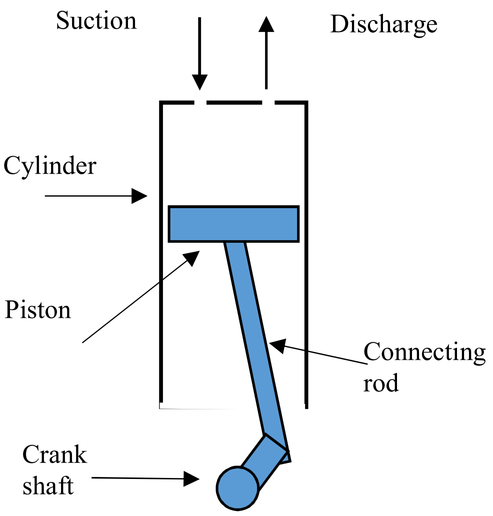


Figure 6.7 Image of a single-stage reciprocating compressor.

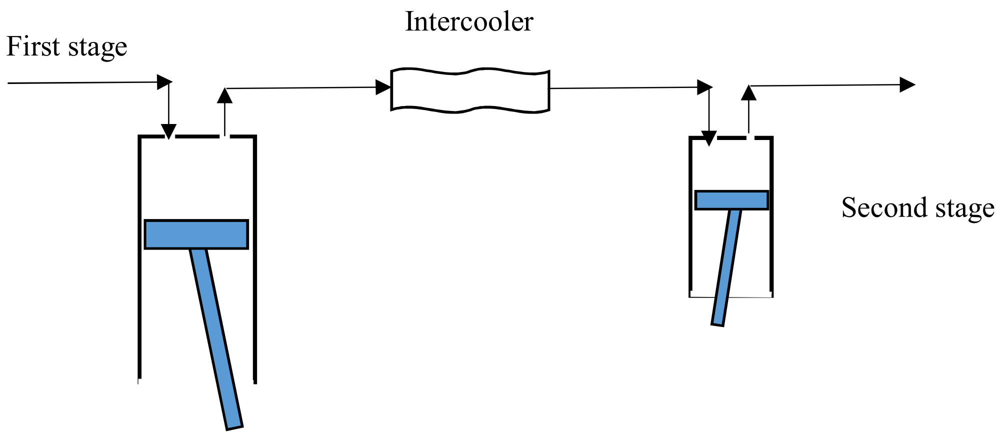


Figure 6.8 Arrangement of a two-stage reciprocating compressor.

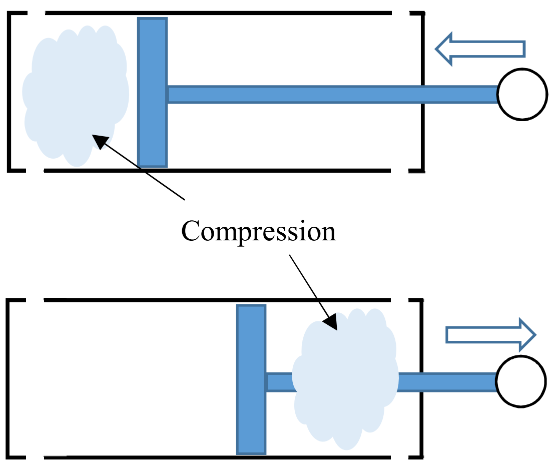


Figure 6.9 Arrangement of a double-acting reciprocating compressor

The compression process showing the relationship between pressure and volume of air for a reciprocating compressor is shown in Figure 6.9.

The different stages of the compression process are:

1 to 2 is where air enters the compression chamber and gets compressed due to the movement of the piston

2 to 3 is where the compressed air is discharged through the discharge port due to the movement of the piston

3 to 4 is where the air trapped in the clearance volume expands (prior to the opening of the suction port)

4 to 1 is the intake stroke where air enters the compression chamber through the suction port

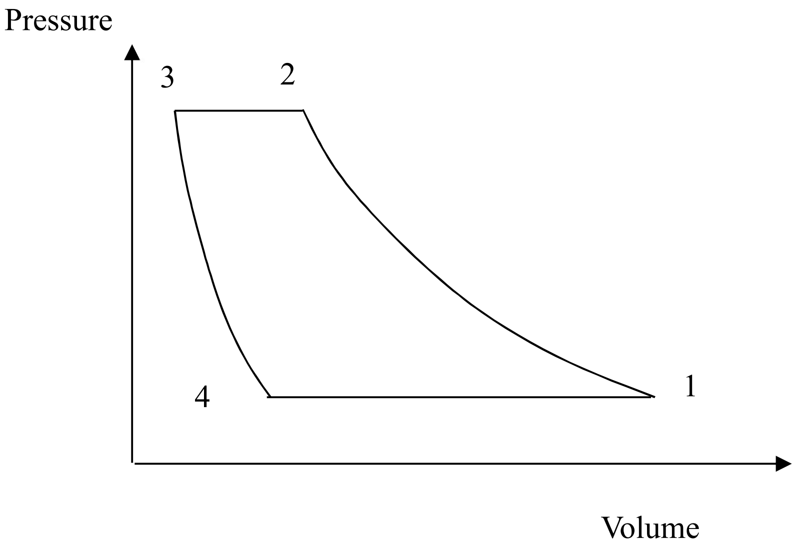


Figure 6.9 Compression process on a PV diagram

Single-acting compressors are used up to about 170 CMH (100 cfm) and double-acting from about 760 CMH (450 cfm). Single-stage is used for pressure from 4.8 barg to 7 barg for single-stage and 7 to 17 barg for two-stage systems (three-stage for higher pressures).

Approximate specific power (defined in section 8.2) at 7 barg:

Single-acting, single-stage	24 kW/100 cfm
Single-acting, two-stage	19 - 21 kW/100 cfm
Double-acting, two-stage	15 - 16 kW/100 cfm
Oil less type	19 – 26 kW/100 cfm

6.6.2 Scroll Compressors

Scroll compressors use two inter-fitting, spiral-shaped scroll members as shown in Figure 6.10. One scroll rotates while the other remains stationary. Due to the profile of the scrolls, the gas drawn in through the inlet port is compressed between the scrolls during rotation and then discharged at the discharge port.

Figure 6.10 Arrangement of a scroll compressor

6.6.3 Screw Compressors

Screw compressors consist of a set of male and female helically grooved rotors and compression is achieved by direct volume reduction due to the rotation of the rotors. The gas is taken in at the inlet port and then compressed during the rotation of the rotors and finally discharged at the discharge port. The cut-away image of a typical screw compressor is shown in Figure 6.11.

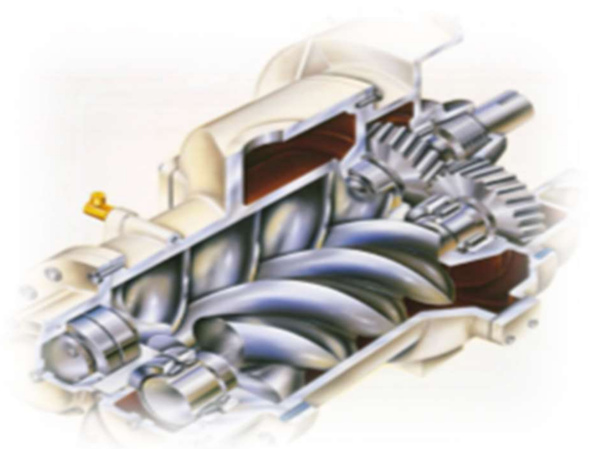


Figure 6.11 Cut-away of a screw compressor (courtesy of Ingersoll Rand)

The compressor consists of intermeshing male and female rotors inside a stator housing. Most common design is with 6 flutes on the female rotor and 5 lobes on the male rotor. When the female rotor makes 5 rotations, the male rotor makes 6 rotations. When a female flute passes the inlet port, air fills up the space of the flute. During rotation, a male lobe engages the flute and traps the air in the flute. As the rotation continues, the male lobe moves along the flute and reduces the volume of the trapped air. When the end of the female flute is exposed to the discharge port, the trapped air is expelled.

A screw compressor usually comes as a package unit as shown in Figure 6.12.

Figure 6.12 Image of a screw compressor package (courtesy of Ingersoll Rand)

The capacity of a screw compressor depends on the rotor diameter, rotor length and the rotational speed. Volumetric efficiency is dependent on the clearance between the rotor and the housing, which creates a leakage path. The bigger the rotor diameter, the higher the leakage path and therefore the lower the volumetric efficiency. Therefore, twin screw arrangements with two sets of screws in parallel are used for higher capacities.

For high-pressure applications, multi-stage compressors with screws in series are used. Rotary screw compressors can be oil-injected or oil-free, and water-cooled or air-cooled.

Oil-free systems can have intercooling between stages. For oil-injected systems, intercooling cannot be used since condensate formed during intercooling will contaminate the lubricant. Therefore, for single stage applications, the oil-injected type is more efficient while for multi-stage, the oil-free type with intercooling can be more efficient.

They are normally used for pressures up to about 10 barg. The specific power for rotary screw compressors is about 18 to 23 kW/100 cfm.

6.6.4 Sliding-Vane Compressors

Sliding-vane compressors consist of a rotor with slots and sliding vanes placed in them as shown in Figure 6.13. The rotor is eccentrically arranged within the housing, providing a crescent-shaped swept area between the intake and discharge ports. The rotor vanes are forced out from the slots up to the housing wall by centrifugal force during rotation. Gas enters through the intake port, and gets trapped between two sliding vanes, the rotor, and the housing. As the rotor turns, the volume occupied by the trapped gas reduces to a minimum where it is exhausted at the discharge port.

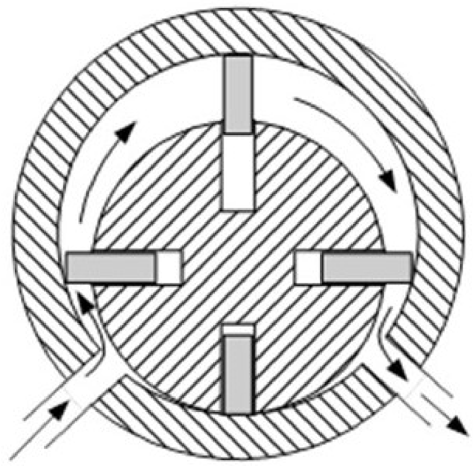


Figure 6.13 Cut-away of a rotary sliding-vane compressor

6.6.5 Centrifugal Compressors

Centrifugal compressors consist of a single impeller or a number of impellers mounted on a shaft and rotating at high speed inside a housing. The gas to be compressed enters the impeller in the axial direction and is discharged radially at high velocity. The velocity pressure is then converted into static pressure in the diffuser. Since the part-load efficiency of centrifugal compressors is low, they are normally used for base-load applications. The image of a centrifugal compressor is shown in Figure 6.14.

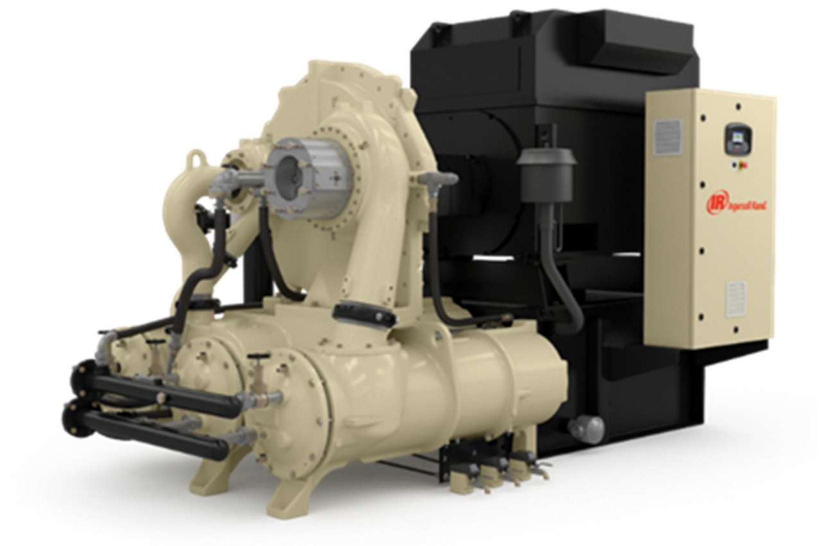


Figure 6.14 Image of a centrifugal compressor (courtesy of Ingersoll Rand)

Centrifugal compressors can be single-stage or multi-stage where compressors are arranged in series with intercooling between stages. Their operating capacity is controlled using inlet guide vanes. They are of the oil-less type and are more reliable because they have fewer rotary parts.

Centrifugal compressors are used for high capacity applications (1300 CMH to 34,000 CMH). They are used for constant load and 24 hour applications (based load applications).

The best efficiency of centrifugal compressors is when operating at 80% to 100% of their rated capacity. Specific power is about 17 to 21 kW/100 cfm.

6.6.6 Axial-Flow Compressors

Axial-flow compressors consist of a number of rows of rotating aerofoil blades (rotors) and stationary blades (stators) placed axially as shown in Figure 6.15. The gas to be compressed passes parallel to the axis of rotation through each stage of rotors rotating at high speed, where velocity pressure is imparted. This is later converted to static pressure by diffusion when passing through the stators.

Since the differential pressure that can be generated per stage is limited, typically 5 to 10 stages are used. They are suitable for very high demand applications and are normally rated over 750 kW.

Some typical applications of axial-flow compressors are:

- FCC (fluid catalytic cracking) air blowers
- Blast furnaces
- Air separation plant
- Sewage treatment
- Wind tunnels

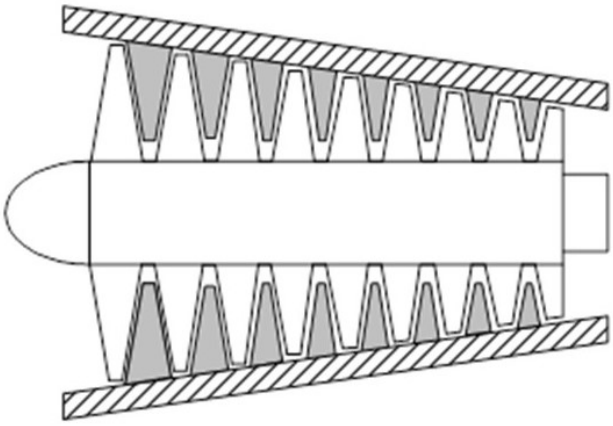


Figure 6.15 Cut-away of an axial-flow compressor

Although there are many types of air compressors available in the market, the main types commonly used in industrial facilities are reciprocating, rotary screw (oil-injected and oil-free) and centrifugal. The usages based on type of application are summarised in Table 6.4.

Application	**Type of compressor**
General manufacturing	Reciprocating or oil-injected screw (rotary) compressors
Industry where air quality requirements are not very stringent	Oil-injected screw (rotary) compressors
High tech. industry (semiconductor, pharmaceutical and food manufacturing etc.)	Oil-free screw and centrifugal compressors

Table 6.4 Usage of compressors for different applications

The selection of the type of air compressor also depends on the operating pressure and capacity. Figure 6.16 shows the operating range for reciprocating, rotary screw and centrifugal compressors.

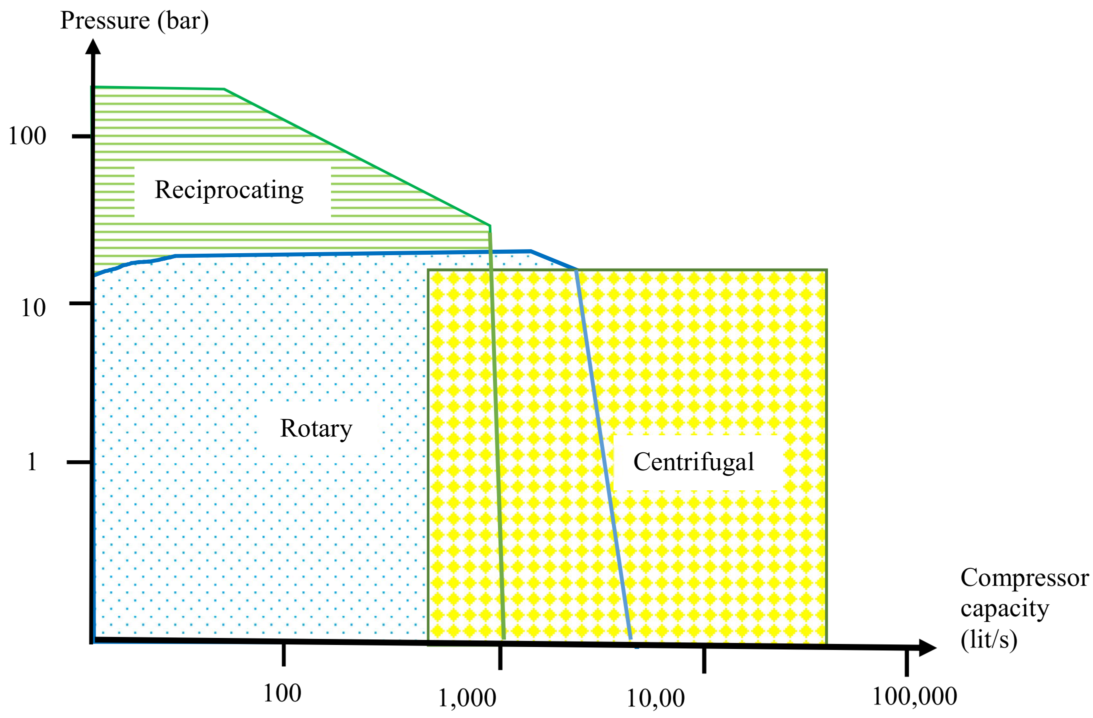


Figure 6.16 Operating range for common compressor types

Summary

The main components used in compressed air systems were described followed by standard conditions used for rating such systems. Some commonly used terms such as free air delivery, utilisation factor and duty cycle were explained followed by the different types of compressors and their main applications.

There is significant potential to enhance the operating efficiency of compressed systems. This will be described in detail in the next few chapters of this reference manual.

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7.0 AUXILIARY EQUIPMENT IN COMPRESSED AIR SYSTEMS

The basic objectives of a compressed air system are to:

- Provide the required compressed air quantity to users
- Maintain a minimum pressure
- Minimise variation in system pressure
- Match supply to demand
- Maintain quality of air to meet user requirements (moisture, oil etc.)

Therefore, in addition to the air compressors, a variety of other auxiliary equipment is used in compressed air systems. Some of the main auxiliary equipment used in compressed air systems are:

- Dryers
- Receiver tanks
- Filters
- After coolers
- Distribution system

This chapter describes the main auxiliary equipment used in compressed air systems.

Learning Outcomes:

The main learning outcomes from this chapter are to understand:

1. The main types of auxiliary equipment used in compressed air systems
2. The various applications of auxiliary equipment based on the various requirements of the end-users of compressed air

7.1 Dryers

Air contains moisture and when compressed, the same amount of moisture is then contained in a lower volume of air. This results in condensation which needs to be removed as otherwise it could damage equipment, cause corrosion and even affect the performance of systems.

Moisture therefore needs to be removed from compressed air before being supplied to most end users. The aftercooler that is used to cool the air after compression to bring it down to a usable temperature removes a significant amount of moisture from the air. In addition, dedicated dryers are used to remove further moisture from the compressed air.

The dryness of the air after the dryer is normally indicated as “dew point”, which is the temperature to which the air needs to be cooled before the moisture in the air condenses. Therefore, air at a low dew point needs to be cooled to a lower temperature before moisture can condense indicating that the moisture content of the air is low.

The dew point is pressure-dependent. The dew point of air can be stated at atmospheric pressure or at the operating pressure (pressure dew point). Therefore, if the mass of moisture in the air remains constant, the pressure dew point would be higher than the atmospheric dew point. For example, if the atmospheric dew point of a stream of air is -20°C, the respective pressure dew point at a pressure of 7 barg will be +5°C. The relationship between atmospheric dew point and pressure dew point is listed in Table 7.1 for a few common system operating pressures.

Atmospheric dew point (°C)	**Approximate pressure dew point at 3 barg (°C)**	**Approximate pressure dew point at 7 barg (°C)**	**Approximate pressure dew point at 15 barg (°C)**
+10	+32	+46	+57
+0	+20	+32	+43
-10	+6	+18	+29
-20	-5	+5	+15
-40	-27	-20	-13
-60	-48	-44	-37
-70	-60	-55	-49

Table 7.1 Pressure dew point temperature values

The compressed air dew point (highest acceptable amount of moisture) is dependent on the process or application served by compressed air. Some of the typical dew point requirements are summarised in Table 7.2.

Application	Typical dew point at 7 barg (°C)
Air motors	+7
Air agitation	-20
Machine tools	+7
Conveying granular materials	-20
Conveying powder materials	-40
Food & beverage industry	-70
Pharmaceutical industry	-70
Electronic industry	-70
Process control instruments	-20

Table 7.2 Pressure dew point requirements

The required dew point dictates the type of dryer that needs to be used to achieve it. Generally, the lower the dew point, the higher will be the energy cost to operate the dryer. Table 7.3 provides a comparison of the achievable dew point temperatures and added energy consumption (to operate the dryer and to overcome pressure losses) for common type of dryers.

Type of dryer	Achievable pressure dew point (°C)	Approximate added energy usage
Refrigerant dryers	0 to +3	5%
Heatless desiccant (air purging) dryers	-40 to -70	20%
Internally heated desiccant dryers	-40 to -70	18%
Externally heated desiccant dryers	-40 to -70	15%
Waste heat / heat of compression dryers	-40 to -70	10%

Table 7.3 Pressure dew point requirements

The main types of dryers used in compressed air systems are described below.

Refrigerant dryers

Refrigerant dryers consist of an evaporator, condenser, compressor and an expansion device and work based on the vapour compression cycle. Liquid refrigerant evaporates in the evaporator. It is then compressed and condensed back at a higher pressure in the condenser. Compressed air is passed over the evaporator coil which cools the air to its dew point, resulting in condensation. An air-to-air heat exchanger is used to pre-cool the saturated air using the cool dry air after the evaporator.

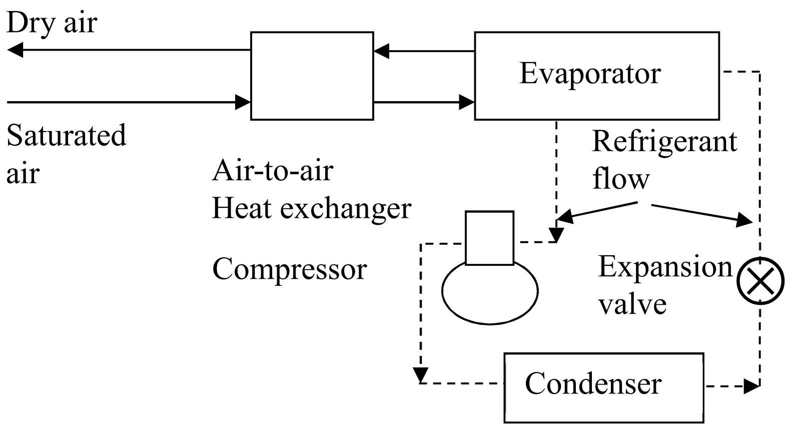

Figure 7.1 Arrangement of a refrigerant dryer

Refrigerant dryers can normally achieve dew point of about +2°C / 3°C and are sufficient for general applications. They generally have low operating cost, low pressure drop (about 0.2 barg) and low maintenance cost as compared to desiccant dryers.

Desiccant dryers

Desiccant dryers consist of desiccant material that absorbs moisture from the air. They are able to achieve very low dew point temperatures such as -70°C (Figure 7.2). These dryers normally use two containers filled with desiccant and air is passed through one of the containers to produce dry air. Once the desiccant in this container is saturated with moisture absorbed from the stream of compressed air, this container is taken out of service and the air is passed through the second container. The absorbed moisture is then removed from the desiccant in the container that is out of service by a “regeneration” process.

Figure 7.2 Image of a desiccant dryer (courtesy of Ingersoll Rand)

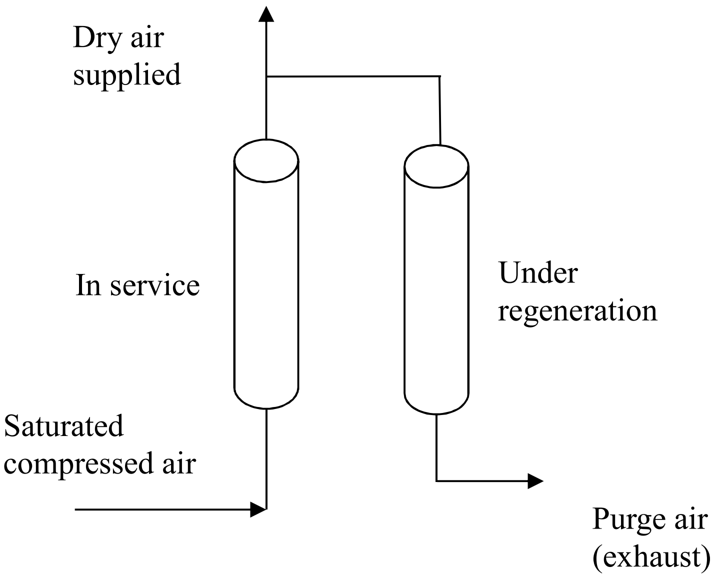

Figure 7.3 Arrangement of a heatless desiccant dryer

One of the basic processes of regeneration is by purging, whereby part of the dry compressed air (taken after the dryer cylinder in operation) is passed through the desiccant in the cylinder that is not in service and this air is exhausted out (Figure 7.3). The amount of air required for purging is about 15 to 20% of the compressed air. Such dryers are therefore not very efficient.

To minimise purging compressed air losses, heaters embedded in the desiccant (internally heated desiccant dryers) are used in some systems where the heaters are turned on during regeneration. Internally heated dryers can reduce the purge air losses by about half.

As an alternative, externally heated desiccant dryers with blowers can be used, whereby air is heated before entering the desiccant container being regenerated which results in the need for much less purging air. The arrangement of an externally heated desiccant dryer is shown in Figure 7.4.

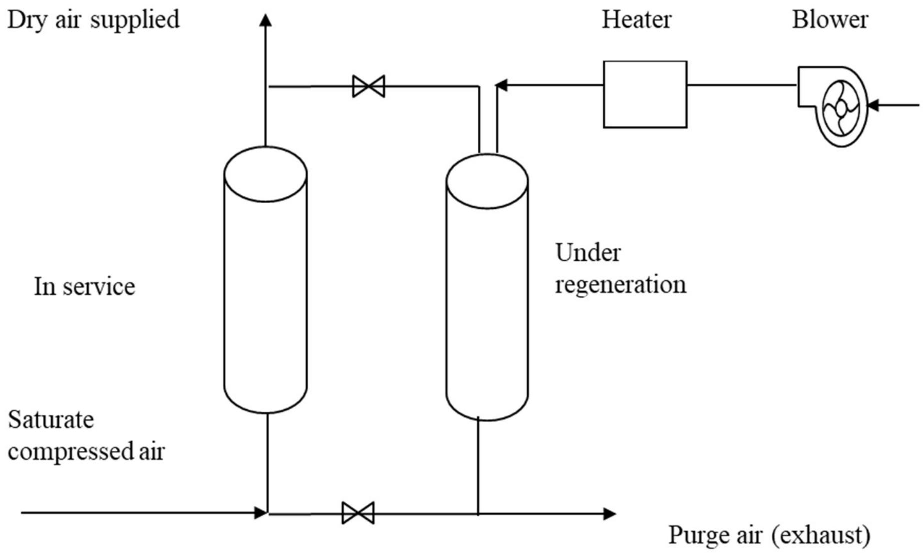

Figure 7.4 Arrangement of a desiccant dryer with external blower & heater

Another method of regeneration is to use the heat produced by the compression process. In such heat of compression desiccant dryers, part of the hot air after the compressor is passed through the desiccant to be regenerated before entering the aftercooler as shown in Figure 7.5.

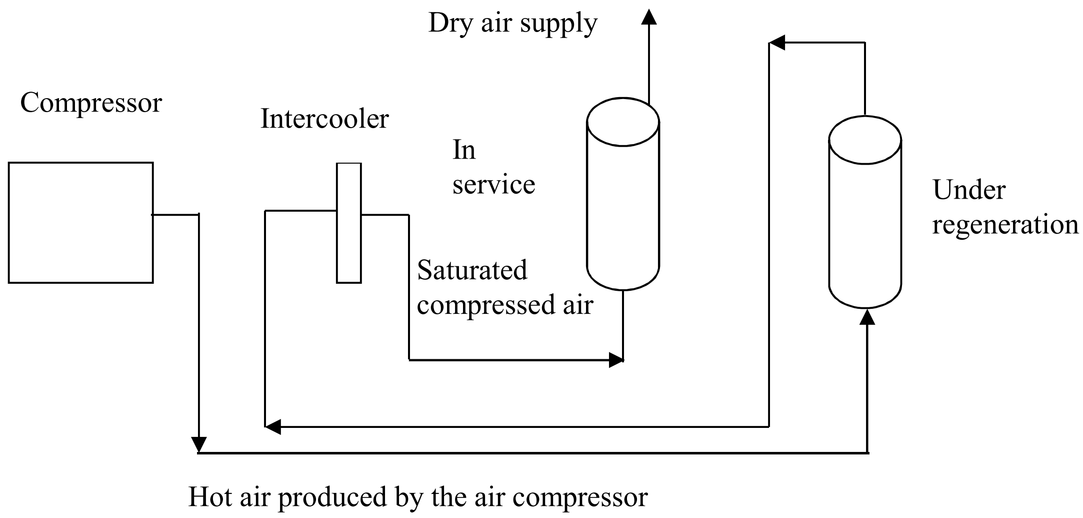

Figure 7.5 Arrangement of a heat of compression desiccant dryer

Deliquescent dryers

These are used for small specialised applications where a continuous supply of dry compressed air is not required. Hygroscopic desiccant material absorbs the water vapour and is dissolved in the liquid formed.

The desiccant is consumed only when moist air is passing through the dryer, and on average desiccant must be added two or three times per year to maintain a proper desiccant bed level. Pressure dew points from single tower desiccant dryers can be as low as -40°C.

A comparison of the energy performance of the different types of dryers is provided in Table 7.4.

Type of dryer	Energy performance (kW/100 CFM)
Refrigerant	0.8
Heatless desiccant	2 to 3
Externally heated desiccant	1 to 2
Heat of compression	0.8
Deliquescent	0.2

Table 7.4 Comparison of the energy performance of dryers

7.2 Receiver tanks

In most compressed air systems, the demand varies between a minimum and maximum value. Since compressors are selected based on a certain capacity, receiver tanks (Figure 7.6) are used to act as storage devices so that the compressors can cycle between set load and unload pressures to provide supply air at a constant pressure to the users. Therefore, they help to minimise pressure fluctuations in the system and allows for sudden demand changes. They also help to remove condensate from the system.

Figure 7.6 Air receiver tanks

The two main types of receivers used are a) primary receivers installed near the air compressors between the aftercoolers and filters, and b) secondary receivers, which are installed near loads with high intermittent demand.

The size (volume) of the required receiver depends on the maximum and minimum pressures set for the compressors to cut out and cut in, the capacity (FAD) of the compressors and the time allowed for pressure to reduce from the maximum to minimum values, and can be expressed as follows:

Receiver volume $V = (t \times FAD \times Pa) / (P_{max} - P_{min})$ (7.1)

where,
V is in m^3
t = time required (seconds) to increase receiver pressure from P_{min} to P_{max}
P_{max} = maximum pressure at which the compressor cuts out (bar)
P_{min} = maximum pressure at which the compressor cuts in (bar)
Pa = atmospheric pressure (bar)
FAD = free air delivery required (m^3/s)

Since in most situations, the size of the receiver tank installed is known, the above relationship can be used to estimate the FAD of the compressor by measuring the time taken (t) for the receiver pressure to increase from P_{min} to P_{max} (with the discharge valve after the receiver closed).

Example 7.1
Compute the FAD of an air compressor if it takes 30 seconds to increase the receiver tank pressure from 7 bar to 9 bar (both absolute pressures). The receiver tank volume is $1m^3$.

Solution
Using relationship (7.1),
Receiver volume V= (t x FAD x Pa) / (P_{max} – P_{min})

Therefore, FAD = V x (P_{max} – P_{min}) / (t x Pa) = 1 x (9 – 7) / 30 x 1 = 0.067 m^3/s

7.3 Filters
Dirt particles, liquid droplets and water vapour can enter compressed air systems through the intake. In addition, other contaminants like oil droplets can be introduced by the air compressor. Depending on user requirements, the contaminants may need to be removed to achieve the acceptable level of air quality.

Therefore, various types of filters are used to remove these contaminants. The three main types of filters used are called:

- Particulate
- Coalescing
- Adsorbing

Figure 7.7 Image of a typical filter (courtesy of Ingersoll Rand)

Particulate filters

Particulate filters use a filter element to trap solid particles up to 1 μm and water and oil droplets up to 0.1 μm.

Coalescing filters

Coalescing filters remove liquid and aerosols (particle size up to 0.01 μm). They can remove water and oil aerosols by coalescing the aerosols into droplets because of tortuous path and pressure drop within the filter.

Adsorbing filters

Adsorbing filters use activated carbon to remove oil vapours and hydrocarbon odours. They are used for breathing air, food, beverage and pharmaceutical applications.

Since higher purity air requires the use of filters with higher pressure drop, if only some users require high quality air, only that stream of air should be filtered to achieve the required purity.

7.4 Aftercoolers

Since the compression process significantly increases the temperature of air, after compression the air has to be cooled to a temperature close to the ambient temperature. Typical air temperatures at the outlet of an air compressor are shown in Table 7.5.

Compressor type	Discharge air temperature (°C)
Rotary oil flooded	95
Rotary oil less	180
Reciprocating	150
Centrifugal	110

Table 7.5 Typical compressor discharge temperatures

Therefore, aftercoolers are installed immediately after the compressor. In oil-free multi-stage compressors, intercoolers (which are similar to aftercoolers) are installed to reduce the heat of air between the compression stages.

Aftercoolers can be either air-cooled or water-cooled. The image of a large air-cooled intercooler used with a multi-stage centrifugal compressor is shown in Figure 7.8.

Figure 7.8 Image of an air-cooled intercooler

Air-cooled aftercoolers are finned tube coils where compressed air passes thorough the tubes while ambient air travels over the outside of the tube and the fins.

Water-cooled aftercoolers are shell and tube heat exchangers where compressed air passes through the tubes while the cooling water passes through the shell.

The water-cooled type results in a lower outlet air temperature compared to air-cooled aftercoolers.

Aftercoolers are selected based on approach temperature (difference between the temperature of air leaving the aftercooler and the cooling medium temperature). Typical approach temperatures are 3 to 12°C. Water-cooled aftercoolers operate at lower end of the range while air-cooled aftercoolers operate at the higher end of the range. Other than reducing the temperature of air, they perform the function of removing water and some oil vapour from the air.

7.5 Distribution systems

Distribution systems consisting of piping and fittings are used to transport the compressed air from the compressors to the users. The main objectives of a distribution system are to:

- Transmit the required volume of air
- Withstand the operating pressure
- Minimise pressure losses

Pressure losses in distribution systems can be categorised as:

- Friction losses
- Dynamic losses

Friction losses

Friction losses can be computed using the Darcy-Weisbach relationship.

$$\Delta P_f = \frac{f \, . \, L \, . \, v^2}{2 \, . \, g \, . \, D} \qquad (7.2)$$

where,

ΔP_f = frictional pressure drop (m)

f = friction factor

L = length of pipe (m)

v = mean velocity (m/s)

g = acceleration due to gravity (m/s^2)

D = inside pipe diameter (m)

By inspecting equation (7.2), it can be seen that friction losses can be reduced by increasing pipe diameter D, which also lowers the velocity v, and by reducing the pipe length L. Ideally, the total friction losses for the piping distribution system should not be more than 0.1 bar.

Designing the piping distribution in the form of a loop as shown in Figure 7.9, (rather than a single distribution pipe), can help to reduce velocity, as air can travel in 2 ways to any point in the system, thereby reducing the flow path.

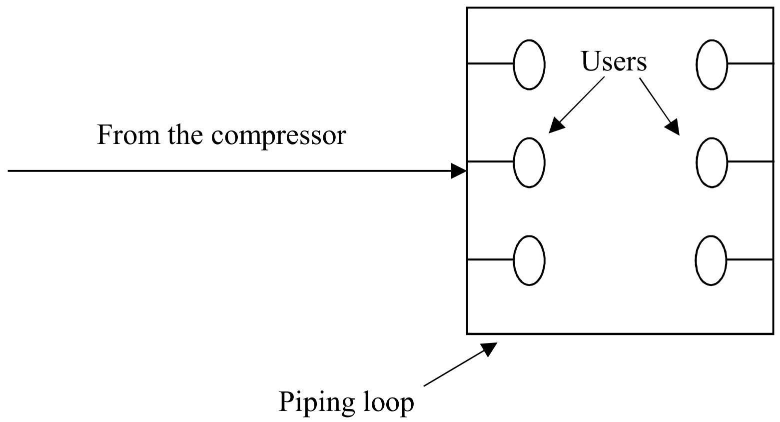

Figure 7.9 Distribution piping in a loop arrangement

Other than increasing pipe diameter, an auxiliary receiver (a secondary storage tank) near an intermittent heavy user, as shown in Figure 7.10, can also be used to reduce pipe velocity. Such a design allows the receiver tank to gradually fill up when the heavy user does not require air, and thereby maintain a lower velocity between the air compressor and the receiver. When the heavy user requires air, it will be supplied from the receiver that is installed adjacent to it and therefore the air will have to travel only a short distance.

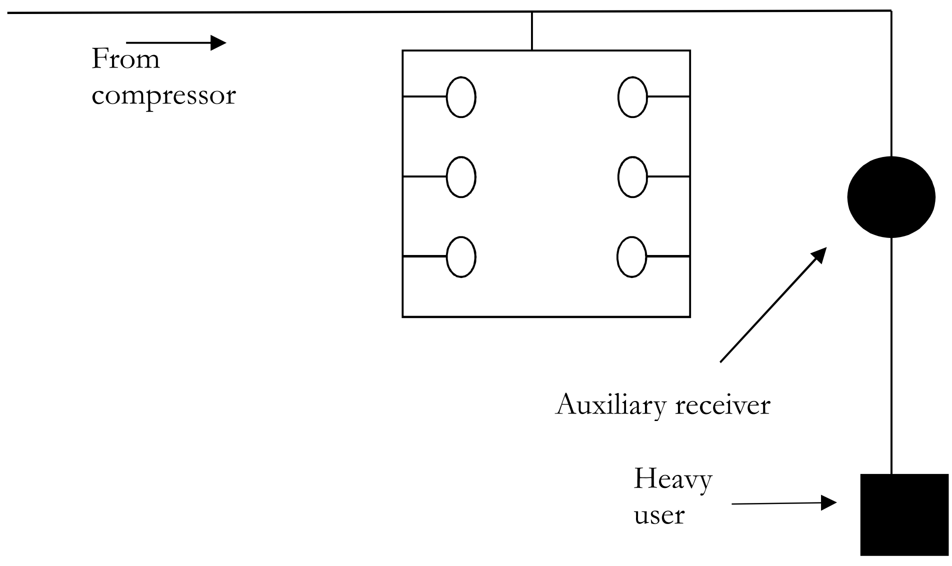

Figure 7.10 Arrangement of an auxiliary receiver to reduce pipe velocity

Dynamic losses

Dynamic losses are due to:

- Change in flow direction caused by fittings such as elbows, tees and bends
- Change in velocity caused by diverging and converging piping sections
- Various devices installed in the system such as filters, valves, receivers and dryers

Some typical pressure losses for common types of equipment are given in Table 7.6. The total dynamic losses should not exceed about 0.4 bar so that the total pressure losses (friction losses and dynamic losses) do not exceed 0.5 bar.

Equipment type	Pressure losses (bar)
Receivers	0.3
Filters	0.07 to 0.2
Refrigerant dryers	0.09
Desiccant dryers	0.2 to 0.3

Table 7.6 Pressure losses for common equipment used in compressed air systems

Summary

This chapter described the main auxiliary equipment used in compressed air systems such as dryers, receiver tanks, filters, aftercoolers and piping distribution systems. The various applications for each type of equipment were also discussed.

References

1. Brian S Elliott, Compressed air operations manual, McGraw-Hill, New York, 2006.
2. Energy saving in the filtration and drying of compressed air, Good practice Guide 216, Department of the Environment, Transport and the Regions, UK, 1998.
3. Improving compressed air system performance: A source book for industry, US DOE, 2003.
4. Jayamaha, Lal, Energy-Efficient Industrial Systems, Evaluation and Implementation, McGraw-Hill, New York, 2016.
5. Kaeser Kompressoren, Designing Your Compressed Air System, Kaeser Kompressoren.

8.0 BASIC THEORY OF COMPRESSION

Compression of a gas such as air involves increasing the pressure of a fixed mass of air while reducing its volume. This compression process also results in an increase in the temperature of the mass of air. The relationship between pressure, volume and temperature during compression depends on the actual compression process. This chapter will explain the different types of compression processes. Thereafter, an expression will be derived to compute the work done or work done per second required for a particular compression process.

Learning Outcomes:

The main learning outcomes from this chapter are to understand:

1. The various compression processes
2. How to compute the work done and work done per second required for the various compression processes
3. How to assess the performance of a compressor by computing the specific power, isothermal efficiency and volumetric efficiency

8.1 Compression process

As air is compressed from a pressure P_1 to a higher pressure P_2, its volume reduces from V_1 to V_2. During compression, the temperature of the air also increases.

The compression process follows the path PV^n = constant (8.1)

As shown in Figure 8.1, the path of compression can be

- "isothermal" where the temperature is maintained constant (ideal case)
- "adiabatic" where heat is not allowed to flow in or out during the compression
- "polytropic" which is generally the case where heat exchange is allowed during compression but temperature is not maintained absolutely constant

In the expression PV^n = constant, value of n is equal to 1 for isothermal, about 1.4 for adiabatic and from 1.2 to 1.3 for polytropic.

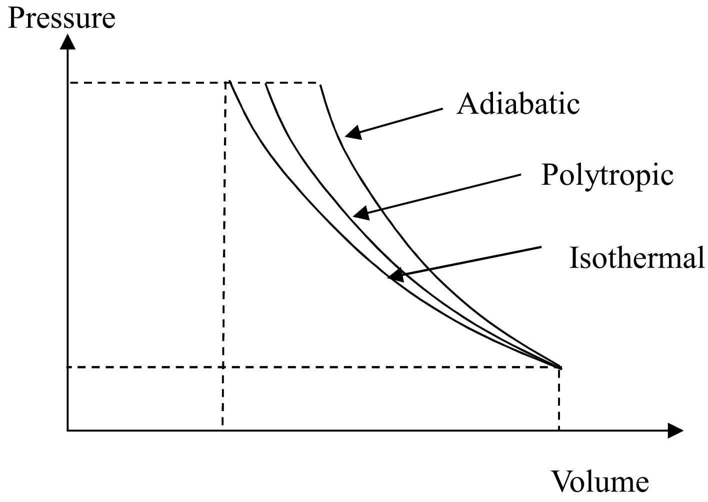

Figure 8.1 Various compression processes

Isothermal compression is the most efficient but is not practical to achieve, while adiabatic compression is the most inefficient.

The actual compression process normally involves four basic stages: intake stage (where air enters the compression chamber), compression stage (where the air gets compressed), discharge stage (where the compressed air is discharged from the compressor) and expansion stage (where prior to the intake, the air trapped inside the compression chamber expands).

A typical compression process is shown in Figure 8.2.

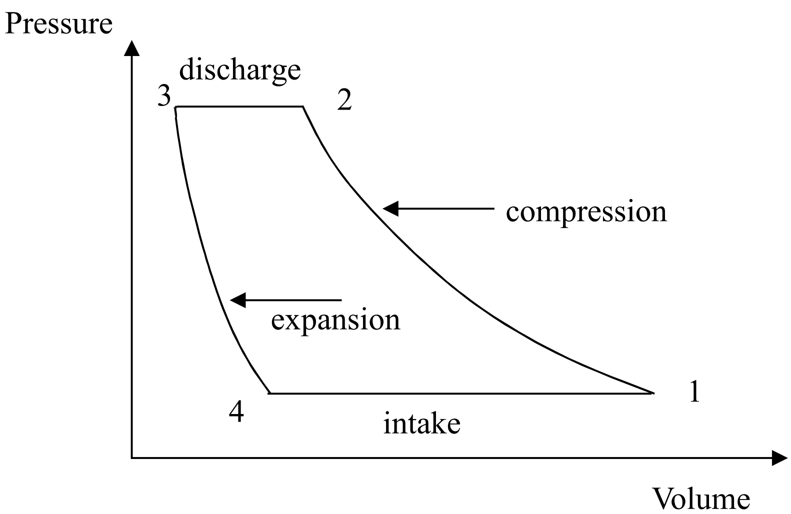

Figure 8.2 Typical Compression process

The work done in compression, $W = \int V \, dP$ (8.2)

The work done is represented by the area enclosed within the P-V diagram shown in Figure 8.3.

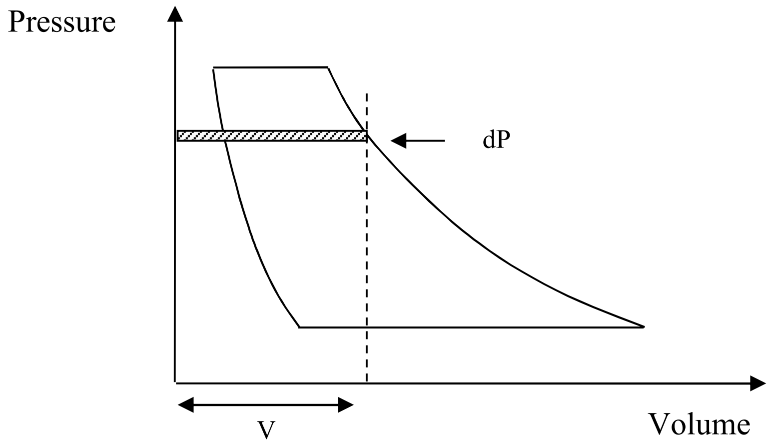

Figure 8.3 Work done in the compression process

Using the relationship (8.1), the work done can be computed by performing the integration represented in equation (8.2) to provide the following two expressions for isothermal compression and polytropic or adiabatic compression.

Work done $W = \int_1^2 V \, dP - \int_3^4 V \, dP$

Since $PV^n = C$, $V = C^{1/n} \times P^{-1/n}$

If the expansion process is neglected, $W = \int_1^2 V \, dP = \int_1^2 C^{1/n} . P^{-1/n} \, dP$

$$= \left[C^{1/n} \frac{P^{(1-1/n)}}{(1-1/n)} \right]_1^2$$

$$= \left[V . P^{1/n} \frac{P^{(1-1/n)}}{(1-1/n)} \right]_1^2$$

$$= \left[\frac{V.P}{(1-1/n)} \right]_1^2 = \frac{n}{n-1} . (P_2V_2 - P_1V_1)$$

$$= \frac{n}{n-1} \cdot P_1V_1 \left(\frac{P_2V_2}{P_1V_1} - 1\right)$$

Since from equation (8.1), $\frac{V_2}{V_1} = \left(\frac{P_2}{P_1}\right)^{-1/n}$

$$W = \frac{n}{n-1} \cdot P_1V_1 \left[\frac{P_2}{P_1} \cdot \left(\frac{P_2}{P_1}\right)^{-\frac{1}{n}} - 1\right] = \frac{n}{n-1} \cdot P_1V_1 \left[\left(\frac{P_2}{P_1}\right)^{\frac{n-1}{n}} - 1\right]$$

Since for an ideal gas PV = mRT, where R = ideal gas constant

$$W = \frac{n}{n-1} \cdot m\,R\,T_1 \left[\left(\frac{P_2}{P_1}\right)^{\frac{n-1}{n}} - 1\right] \tag{8.3}$$

For Isothermal compression, since n=1

$W = \int_1^2 V\,dP = \int_1^2 C.\,P^{-1}\,dP = PV\,[\ln P\,]_1^2$

Therefore,

$$W = mRT_1 \ln\left(\frac{P_2}{P_1}\right) \tag{8.4}$$

where,

W = work done (Joules)

m = mass of air being compressed (kg)

R = specific gas constant for air (287 J/kg.K)

T_1 = absolute temperature at the air compressor intake (K)

P_2 / P_1 = compression ratio in absolute pressure values

If the mass flow rate $\dot{m}$ is used instead of mass, m, the expressions for work done per second will be as follows:

$$\dot{W} = \dot{m}\,RT_1 \ln\left(\frac{P_2}{P_1}\right) \quad \text{for isothermal compression} \tag{8.5}$$

$$\text{and, } \dot{W} = \frac{n}{n-1}\,\dot{m}RT_1 \left[\left(\frac{P_2}{P_1}\right)^{\frac{n-1}{n}} - 1\right] \text{ for polytropic \& adiabatic compression} \tag{8.6}$$

where,
$\dot{W}$ = work done per seconds (Watts)
$\dot{m}$ = mass flow rate (kg/s)

Example 8.1

Compute the mechanical power required to compress 1 m^3/s of dry air at 30°C from 1 bar to 7.5 bar (both absolute pressures) for isothermal, polytropic (n=1.3) and adiabatic (n=1.4) compression. Take the density of air to be 1.2 kg/m^3 and specific gas constant for air R to be 287 J/kg.K.

Solution

$\dot{m}$ = 1 m^3/s x 1.2 kg/m^3 = 1.2 kg/s

Isothermal (from equation 8.5)

$\dot{W}$ = 1.2 x 287 x (273+30) x ln (7.5/1) Watts
= 210 kW

Adiabatic (from equation 8.6)

$\dot{W}$ = 1.2 x 287 x (273+30) x (1.4/0.4) x [$(7.5/1)^{0.4/1.4}$ -1] Watts
= 284 kW

Polytropic (from equation 8.6)

$\dot{W}$ = 1.2 x 287 x (273+30) x (1.3/0.3) x [$(7.5/1)^{0.3/1.3}$ -1] Watts
= 267 kW

8.2 Specific power

Specific power for compressors is a measure of how much power is used to compress a specific quantity of air and is the ratio of power input to the compressor in kW to the free air delivery in m^3/s.

Example 8.2

Compute the specific power in kW per m^3/s and kW per l/s for the three cases of compression in example 8.1.

Solution

Isothermal

Specific power = 210 kW / 1 m^3/s

= 0.21 kW/(l/s)

Adiabatic

Specific power = 284 kW / 1 m^3/s

= 0.28 kW/(l/s)

Polytropic

Specific power = 267 kW / 1 m^3/s

= 0.27 kW/(l/s)

8.3 Isothermal efficiency

Isothermal efficiency is a measure of how close a particular process of compression is to the ideal case of isothermal compression. It is defined as the ratio of isothermal power to the actual power consumed.

$$\text{Isothermal efficiency} = \frac{\text{Isothermal power}}{\text{Actual power consumed}} \qquad (8.7)$$

Example 8.3

Compute the isothermal efficiency of an air compressor operating at 1 m^3/s of dry air at 30^{o}C from 1 bar to 9 bar (both absolute pressures) where the measured power consumption of the motor is 325 kW. The efficiency of the motor and transmission drive between the motor and compressor is 85%. Take the density of air to be 1.2 kg/m^3 and specific gas constant R to be 287 J/kg.K.

Solution

$\dot{m}$ = 1 m^3/s x 1.2 kg/m^3 = 1.2 kg/s

Isothermal power (from equation 8.4)

$\dot{W}$ = 1.2 x 287 x (273+30) x ln (9/1) Watts

= 229.3 kW

Actual power drawn by the compressor shaft = 325 x 0.85 = 276.3 kW

From equation (8.7),

Isothermal efficiency = 229.3/276.3 = 0.83 = 83%

8.4 Volumetric efficiency

Although the theoretical compression capacity of positive displacement machines is the volume available for compression (e.g. cylinder bore size, stroke length and rotational speed for reciprocating compressors), the actual output is normally lower. This is because some of the pressurised air remains inside the 'clearance volume' (as illustrated in Figure 8.4).

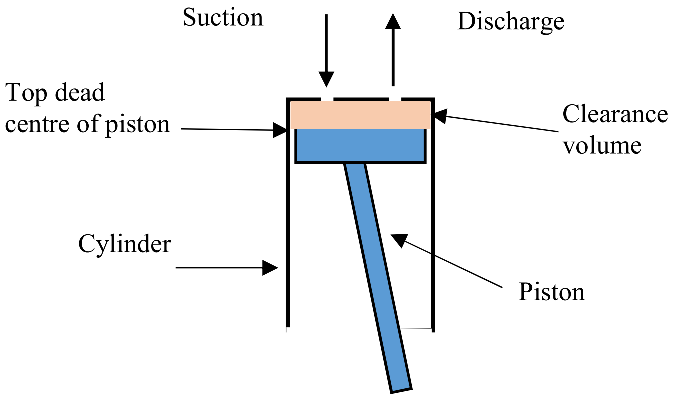

Figure 8.4 Illustration of clearance volume

When the piston begins to move downwards, the suction valve doesn't open immediately because of the air at high pressure. As the piston moves further down, the high-pressure gas in the clearance volume reduces due to expansion, and when the pressure reaches a certain level, the suction valve opens. Therefore, a portion of the air is unused.

Volumetric efficiency is a measure of how much of the volume available for compression is used in the actual compression process. The portion of air that remains unused in a positive displacement compressor leads to a reduction in its volumetric efficiency.

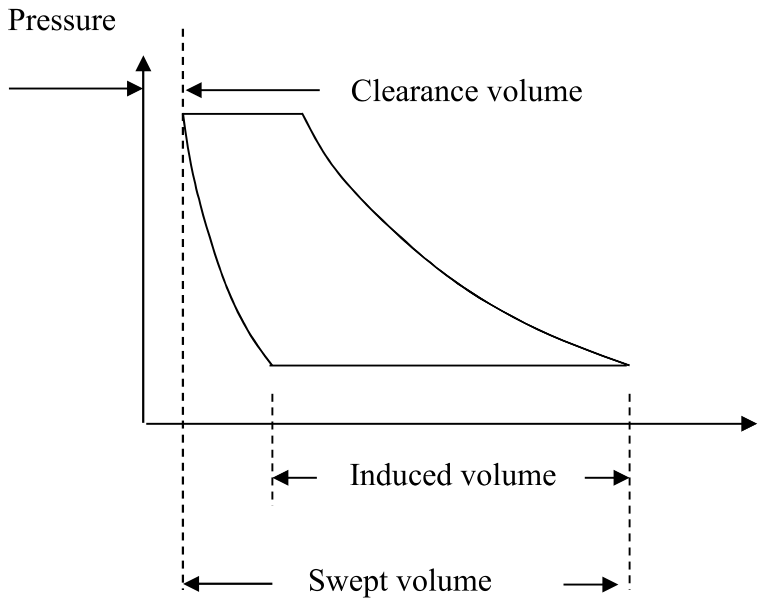

Figure 8.5 Illustration of volumetric efficiency

$$\text{Volumetric efficiency} = \frac{\text{Induced volume}}{\text{Swept volume}} \tag{8.8}$$

As shown in equation (8.8), the volumetric efficiency is defined as the ratio of induced volume (which is the FAD) divided by the swept volume (available volume for compression).

Summary

This chapter explained the different types of compression processes. Thereafter, an expression was derived to compute the work done or work done per second required for a particular compression process. The chapter also described how to evaluate the specific power, isothermal efficiency and volumetric efficiency of a compressor.

References

1. Jayamaha, Lal, Energy-Efficient Industrial Systems, Evaluation and Implementation, McGraw-Hill, New York, 2016.
2. Krati, M, Energy Audit of Building Systems – An Engineering Approach, CHC Press, Boca Raton, FL, 2000.

9.0 COMPRESSOR CONTROLS

Controls in compressed air systems are used to match the supply of compressed air to demand requirements. Since compressed air demand is normally not constant, good control is essential for efficient system performance and operation.

Controls need to be designed to optimise a system, using strategies to reduce the operation of compressors, switching on compressors only when needed, ensuring that the compressors in operation are fully loaded before switching on additional units, and switching on the most efficient unit based on the load.

Compressed air systems are designed to provide varying amounts of compressed air to meet varying load requirements while maintaining a set system pressure. The system pressure is controlled within a band called the control range between a set maximum pressure and a minimum pressure.

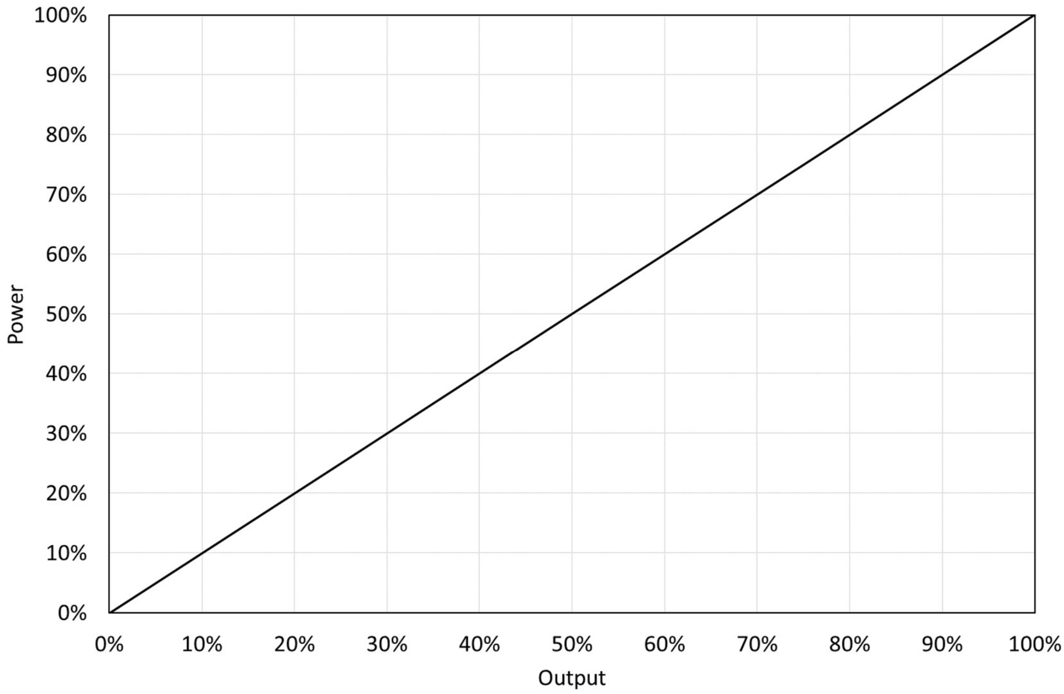

Figure 9.1 Ideal output vs power relationship for an air compressor

The ideal relationship between the power drawn and output for an air compressor is shown in Figure 9.1. As indicated by the figure, when the compressor output reduces from its rated maximum capacity, the power consumed by the compressor should also reduce linearly. Although no compressor control strategy is able to achieve this for the entire operating range from 0 to 100% capacity, some are able to achieve this at some operating points. These will be described in detail in this chapter.

Learning Outcomes:

The main learning outcomes from this chapter are to understand the main air compressor control strategies, which are:

1. On / Off
2. Load / Unload
3. Inlet modulation
4. Variable displacement
5. Variable speed control

9.1 On / Off control

This form of control involves switching on and off the compressor motor to control the amount of compressed air produced to meet the load. This type of control is used only for small compressors that have low duty cycles as otherwise, frequent cycling will lead to overheating of the motor and excessive wear and tear of compressor parts.

The compressor operates under two modes, which are On and Off. These two operating modes can be marked on the ideal relationship between compressor power and output (explained earlier). As shown in Figure 9.2, the two operating modes match the ideal requirements at 0% and 100% output. When the load is less than 100% capacity of the compressor, it will go on and off, and the average value will move along the ideal line. For example, if the load is 50% of the compressor capacity, it will be on for 50% of the time and off during the balance 50% of the time, resulting in an average compressor power of 50% of the power at 100% load.

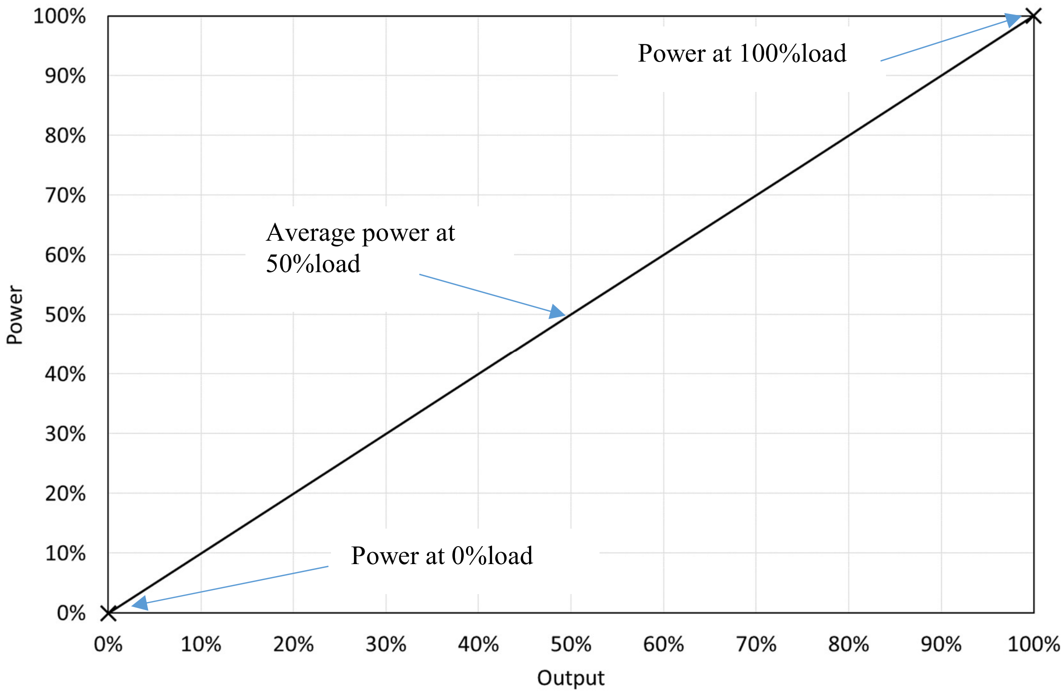

Figure 9.2 Output vs power for on / off control

9.2 Load / unload

This control strategy uses two compressor operating modes: fully loaded and idling (unload). In the fully loaded mode of operation, the compressor runs at a fixed speed and operates at its rated capacity, while under unload mode, the compressor and motor operate at the full-load speed, but the compressor does not produce compressed air.

The main disadvantage of this mode of control is that power is consumed during unload operation. The power consumed during unload operation can be as high as 30% of the full-load power for a screw compressor. However, in the case of reciprocating compressors with multiple cylinders, the unload power can be relatively low as individual cylinders can be unloaded (for example, in a compressor with four cylinders, only one cylinder may be unloaded at 75% load).

Figure 9.3 shows the relationship between compressor power and load. During load operation, a compressor draws 100% of the full-load power. However, during unload operation, reciprocating compressors draw about 15% of the full-load power while screw compressors draw about 25% of full-load power.

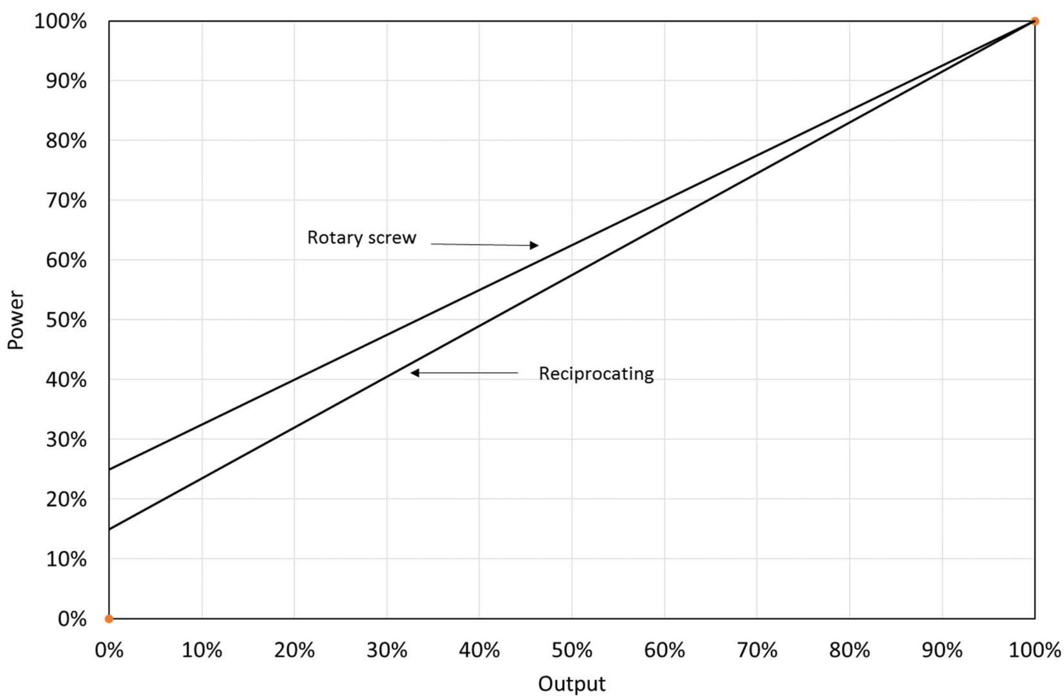

Figure 9.3 Output vs power for load / unload control

For rotary screw compressors, power at unload is required to maintain lubrication and overcome mechanical friction. Most rotary screw compressor designs depend on air pressure in the lubricant sump to circulate lubricant through the system. The pressure differential between vacuum at the compressor air intake and pressure in the lubricant sump circulates lubricant. Therefore, during unloaded operation, it is necessary to maintain minimum pressure in the lubricant sump. As a result, rotary screw compressors consume about 25% of the full load power when unloaded.

In reciprocating compressors, unloading is achieved by keeping the inlet valve open so that during the compression stroke, air is not trapped in the cylinder. Therefore, power is used during unload mainly to overcome mechanical losses.

When the load is less than 100%, the average power is used to represent the power usage in Figure 9.3. For example, if the air demand is 50% of the compressor capacity, the compressor would load 50% of the time and unload 50% of the time. The power consumed at this load is the weighted average of the full load power and the loaded time, and the unloaded power and the unloaded time.

Figure 9.4 shows a compressor power vs time profile when operating at 50% loading (half the time the compressor is loaded and the remaining 50% of the time it is unloaded).

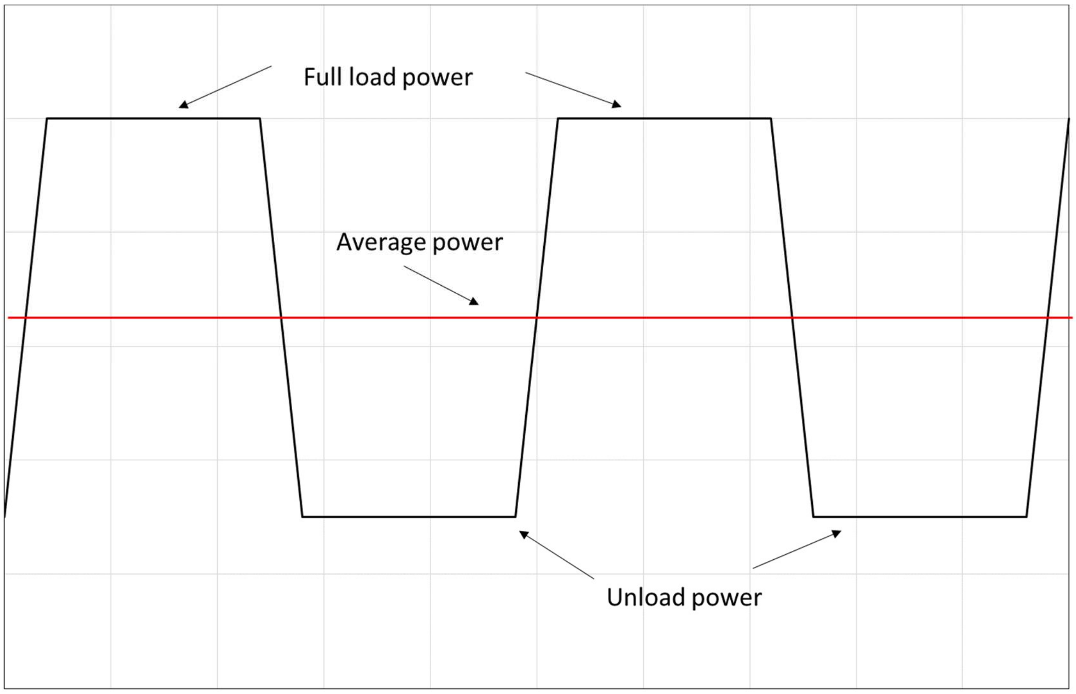

Figure 9.4 Compressor power profile when operating at 50% load

9.3 Inlet modulation

Inlet modulation control involves varying the volume of intake air to match load requirements. At part-load, inlet airflow is reduced by vanes or a modulating valve (Figure 9.5). This reduces inlet pressure and raises the compressor lift, leading to lower compressor efficiency. This mode is generally used for centrifugal compressors to modulate between about 80 to 100% of the capacity.

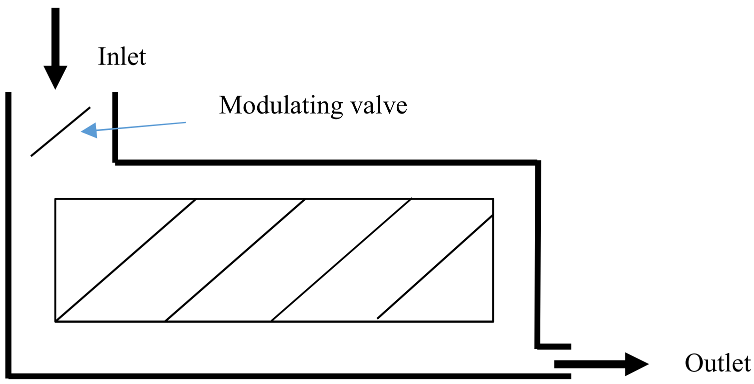

Figure 9.5 Inlet modulation of a screw compressor

With inlet modulation, as shown in Figure 9.6, a 3% power reduction is achieved for every 10% decrease in delivered airflow. Therefore, at 10% output, the compressor can consume as much as 70% of the full load power.

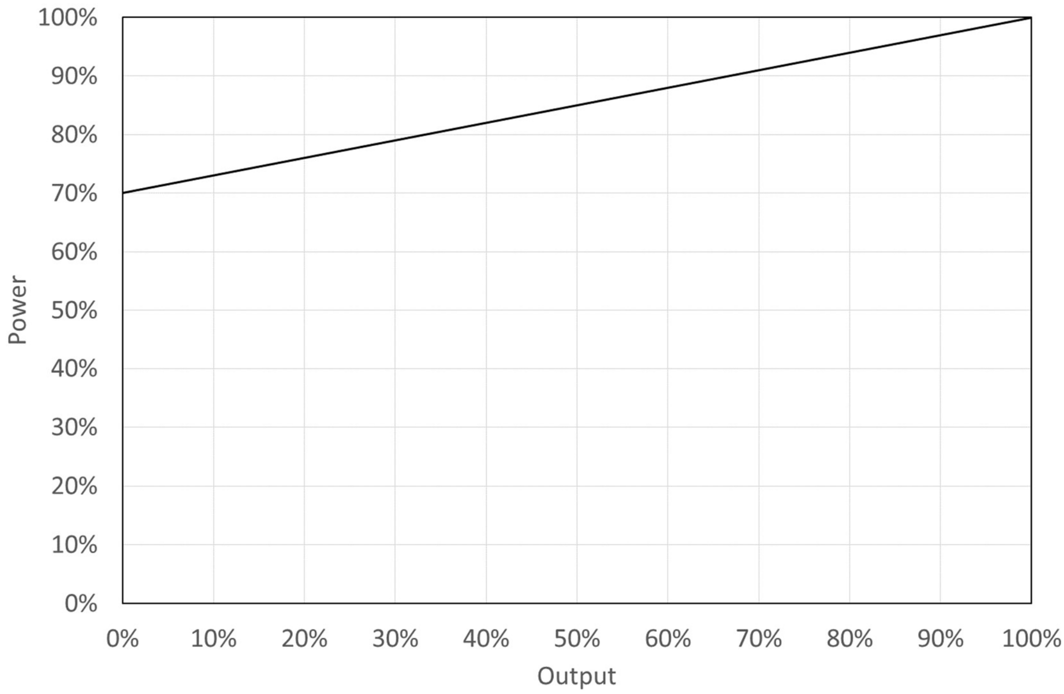

Figure 9.6 Output vs power for inlet modulation control

Due to the extremely poor efficiency at low loads, full range modulating control is rarely used. Often, modulating control is used only for the upper range and is combined with load / unload at lower capacity ranges (Figure 9.7).

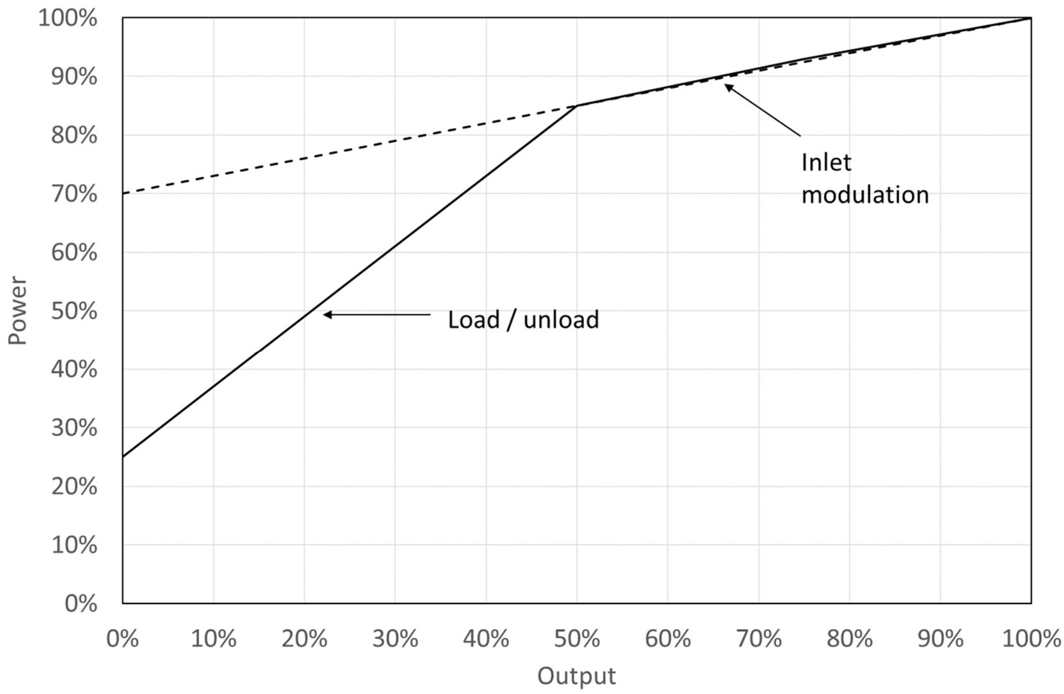

Figure 9.7 Output vs power for inlet modulation followed by load / unload

9.4 Variable displacement

Variable displacement capacity control uses a valve to bypass part of the air back to the inlet without being compressed. Figures 9.8 to 9.9 show variable displacement control at 100%, 75% and 50% loading.

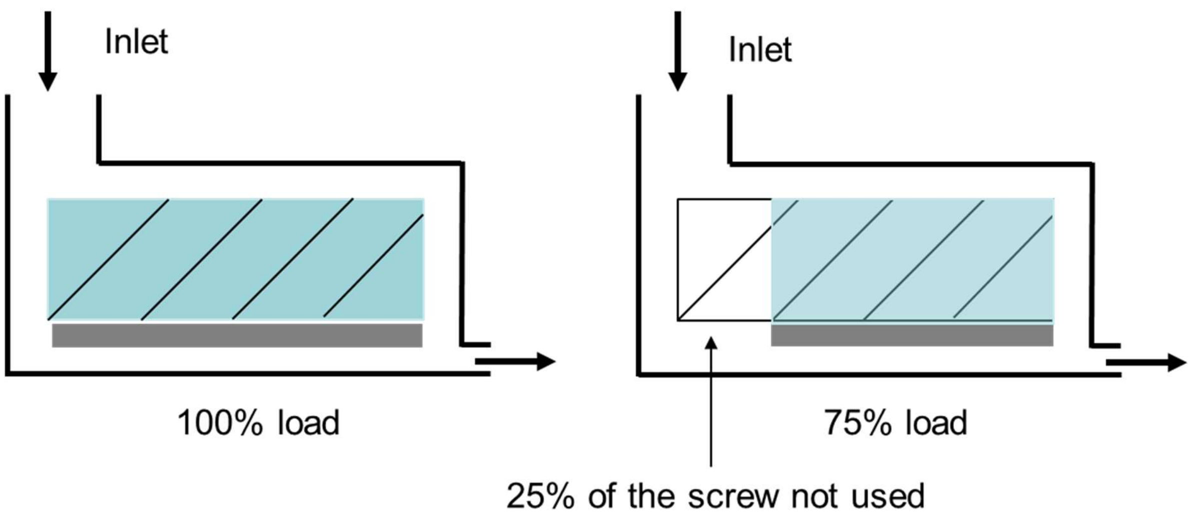

Figure 9.8 Variable displacement control at 100% and 75% loading

Often, variable displacement control is used in conjunction with load / unload control. The comparison of compressor performance using variable displacement and load / unload with the ideal case is shown in Figure 9.9.

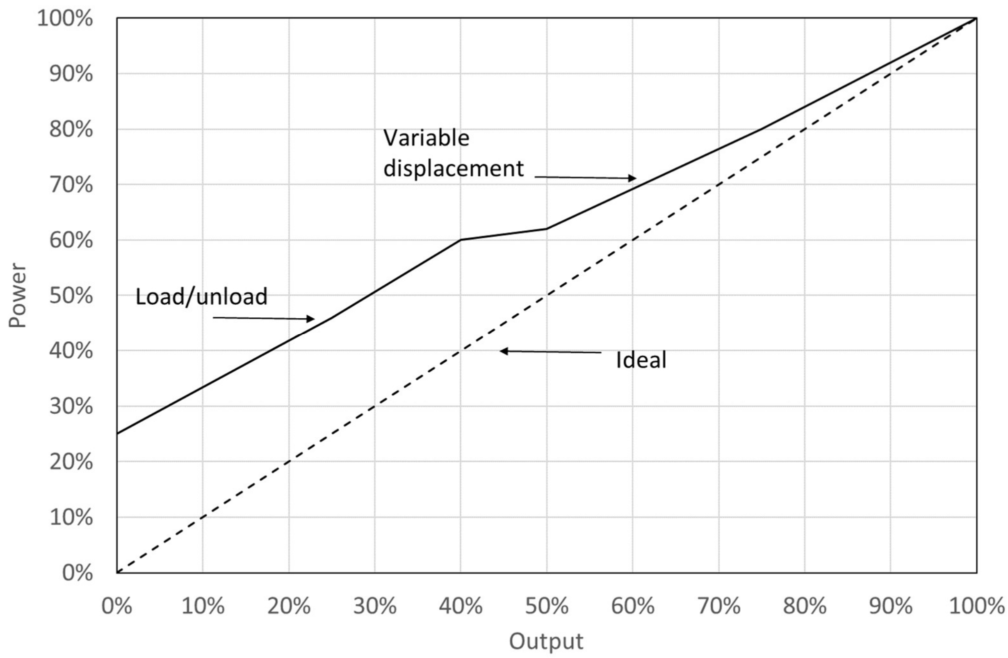

Figure 9.9 Output vs power for variable displacement followed by load / unload

9.5 Variable speed control

This control strategy involves varying the capacity of compressors to match the load by varying the speed of the compressor using variable speed drives (VSDs). At full load, VSD compressors consume about 2 to 4% more power due to losses in the VSDs, but they can maintain good part-load efficiency up to about 50% (i.e. will consume about 50% of full-load power when operating at 50% load).

VSD control is usually used with rotary screw compressors. Due to the higher cost for screw compressors with VSDs and the higher power drawn at 100% speed, VFD compressors should normally be used to meet varying loads, while fixed speed compressors are used to meet the base load.

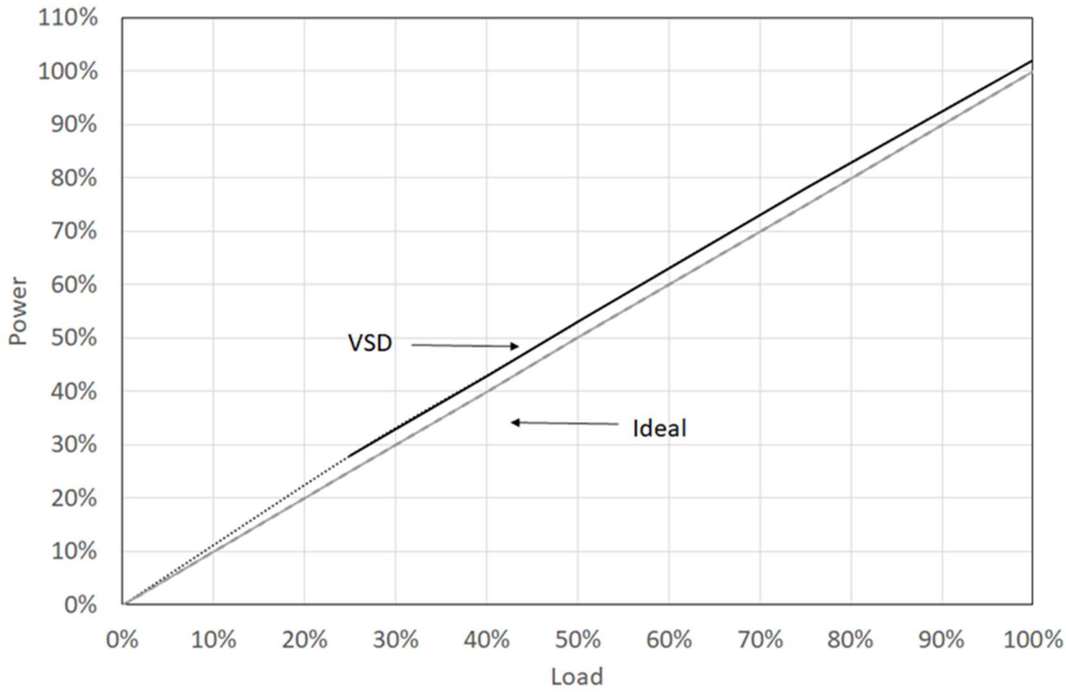

Figure 9.10 Output vs power for VSD compressor

For centrifugal compressors, load / unload, variable displacement and VSD controls are not used. Only inlet modulation using vanes is used to control the capacity of centrifugal compressors between 80% to 100% load. Below 80% capacity, usually blow-off valves are used to release excess output to the environment.

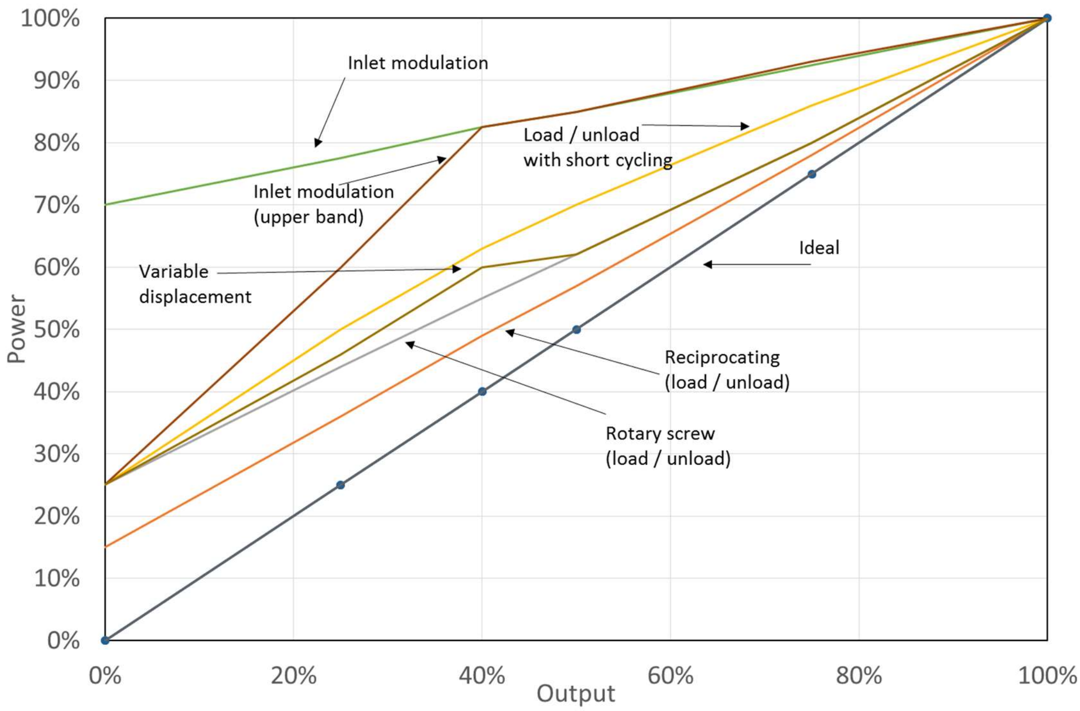

Figure 9.11 Power consumption for different control strategies

A comparison of the power consumption characteristics for the different control strategies is shown in Figure 9.11.

Summary

The main types of control strategies such as on / off, load / unload, inlet modulation, variable displacement and variable speed control, used to vary the capacity of compressors to match their output to the varying load, were explained in this chapter.

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1. Brian S Elliott, Compressed air operations manual, McGraw-Hill, New York, 2006.
2. Improving compressed air system performance: A source book for industry, US DOE, 2003.
3. Jayamaha, Lal, Energy-Efficient Industrial Systems, Evaluation and Implementation, McGraw-Hill, New York, 2016.

10.0 ENERGY SAVING MEASURES FOR COMPRESSED AIR SYSTEMS

As explained in the earlier chapters of this reference manual, compressed air systems are highly energy intensive and account for a significant portion of energy consumed in many plants. Most of the energy used in compressed air systems is wasted as heat, and only about 10% of the input energy is delivered in usable form as compressed air. Further, part of the generated compressed air is wasted due to leaks, artificial demand and inappropriate usage.

However, there are a number of energy-saving measures that can be implemented to reduce the energy usage of compressed air systems. These energy-saving measures can be broadly categorised into those leading to reduction in demand and those improving the compression process. Some of the common energy-saving measures for each category are listed below.

Reduction in demand

- Reducing leaks
- Reducing artificial demand (due to high pressure)
- Eliminating inappropriate usage

Improving compression process

- Reducing pressure ratio
- Multi-staging
- Reducing intake temperature

This chapter describes some of the common energy saving measures that can be implemented for compressed air systems.

Learning Outcomes:

The main learning outcomes from this chapter are to understand:

1. The main energy saving measures for compressed air systems
2. How to estimate the energy savings achievable for each measure

10.1 Rectifying leaks

Since compressed air is produced, stored, distributed and used at a much higher pressure than the surrounding air, compressed air will leak through even the smallest openings, resulting in much energy wastage. In many compressed air systems, leaks account for about 20% to 30% of the total compressor output.

Common sources of leaks are pipe fittings, flexible tubes, couplings, pressure regulators, condensate traps, and pipe joints.

Compressed air leakage depends on the size of the opening, system pressure and the shape of the opening (round or sharp).

Leakage rate can be estimated using Fliegner's formula for flow through an orifice

$$m = \sqrt{(2/R)}\ .\ C\ .\ A \times P \times T^{-1/2} \qquad (10.1)$$

where,

m is in kg/s

C = 0.65 for round holes

A = the orifice size in m^2

P = the pressure in Pa

T = the absolute temperature in K

However, it is not convenient to use the above relationship as the orifice size is normally not known. Therefore, the total leakage rate from a compressed air system can be estimated by switching off all the equipment that uses compressed air, operating the compressor to charge the system to the operating pressure, and measuring the time taken for "load" and "unload" cycles of the compressor continuously for about 10 cycles.

The following expression can be used to estimate the system leakage rate:

$$\%\ \text{of leakage} = T / (T + t) \times 100 \qquad (10.2)$$

and,

$$\text{System leakage } (m^3/min) = (Q \times T) / (T + t) \qquad (10.3)$$

where,

Q = compressor capacity (FAD in m^3/min)

T = average load time (minutes)

t = average unload time (minutes)

Example 10.1

A leak test carried out on a compressed air system with all the compressed air equipment switched off provided the following information :

Average load time = 3 minutes

Average unload time = 10 minutes

If the FAD of the compressor is 0.1 m^3/s, compute the percentage and quantity of leaks.

Taking the specific power for the compressor system to be 270 kW / m^3/s, estimate the energy wastage (in kWh per day) assuming the system operates 24 hours a day.

Solution

Load time = 3 minutes

Unload time = 10 minutes

FAD = 0.1 m^3/s

% leakage = [3 / (3 + 10)] x 100 = 23%

Leakage rate = 0.1 m^3/s x 0.23 = 0.023 m^3/s

Power usage = 0.023 x 270 = 6.21 kW

Energy usage = 6.21 x 24 = 149 kWh / day

10.2 Use of receiver tanks

As explained earlier in chapter 7, receiver tanks are commonly used to store compressed air. Primary receivers installed near the air compressors help to maintain a steady output from the compressors, even when the demand is changing. When the demand is low, the receiver tanks store the excess compressed air while when the demand is high, part of the demand is met by releasing stored air from the tanks.

If receiver tanks are not installed in the system, when the compressed air demand exceeds the capacity of the compressors, the system pressure will drop. To avoid the system pressure dropping below the required minimum pressure at the end-users, the operating pressure is often set at a much higher value than normally required. This results in higher energy consumption by the compressor motor.

Similarly, if there are intermittent high loads, a sudden flow of compressed air through the piping system will result in higher pressure losses. Therefore, if secondary receivers are installed near intermittent high loads, the storage tanks can gradually fill-up when the equipment is not using air, and supply the high flow rate when required, through a much shorter pipe resulting in a much lower pressure drop.

Therefore, installation of receiver tanks helps to operate compressed air systems at the lowest possible pressure. Energy savings resulting from the use of receiver tanks is usually due to the reduction in system pressure and can be estimated as explained later in section 10.5.

10.3 Reducing artificial demand

Different types of equipment need compressed air at different pressures for their operation. Usually the system pressure is set at a value to satisfy all the users. Often, the system pressure is also set to be higher than required to ensure sufficient pressure during sudden high demand, to minimise complaints from end users and to allow for high pressure losses in the distribution system.

Operating the system at a higher than required system pressure not only reduces compressor efficiency (will be explained later), it also results in extra usage called artificial demand.

Compressed air usage by many users can be considered like air flowing through an orifice where, when the upstream pressure increases, the flow through the orifice increases.

Hence, pressure regulators should be installed at the point of use and set at the optimum minimum pressure required for proper functioning. Pressure controllers can also be installed for the distribution system (after the primary receiver) and set to maintain an optimum pressure downstream.

In addition to reducing artificial demand, reducing system pressure will also help to reduce leaks.

Other opportunities for reducing artificial demand are, replacing heatless desiccant dryers (which use purge air for regeneration) with externally heated or refrigerant dryers, repairing faulty dryers that continuously purge and increasing the dew point settings on heatless desiccant dryers.

10.4 Reducing inappropriate usage

Often compressed air is used for applications that can be served by other means. Some such examples are:

- Electrically driven vacuum pumps instead of air powered venturi vacuum systems
- Electric motors for air motors
- Electric actuators for pneumatic actuators
- Using vacuum cleaners for general cleaning
- Using high pressure blowers for tank agitation

<u>Using blowers for tank agitation</u>

Often compressed air is used for tank agitation in applications such as plating to provide movement of the solution in the tank. In such situations, the use of compressed air can be replaced by installing blowers which are much more energy efficient than air compressors.

The blowers should be selected to provide sufficient pressure and air flow rate. The minimum pressure requirement is dependent on the liquid level in tank and can be estimated using equation (10.4).

$$\text{Minimum gauge pressure in bar} = [(\rho \times g \times h) / (1\times10^{-5})] + 0.05 \qquad (10.4)$$

where,

ρ = density of the fluid in the tank (kg/m^3)

g = acceleration due to gravity (m/s^2)

h = height of the liquid in the tank (m)

The air flow rate required for agitation can be expressed by equation (10.5.)

$$\text{The air flow rate required for agitation in SCFM} = A \times F \qquad (10.5)$$

where,

A = tank surface area in m^2

F = agitation factor between 10 to 20 in SCFM/m^2

Example 10.2

A tank contains a solution with a density of 1200 kg/m^3. The liquid level in the tank is 1 m. The surface area of the tank is 2 m^2. It is currently being served by an air compressor having a specific power of 0.2 kW/SCFM.

Compute the required pressure and air flow rate for a suitable blower for this application, taking the agitation factor to be 15 SCFM/m^2. Estimate the reduction in power if the specific power for the blower is 0.08 kW/CFM.

Solution

From equation (10.4),

Gauge pressure requirement = [(1200 x 9.81 x 1) / (1x10^{-5})] + 0.05 = 0.17 bar

The air flow rate required = 2 x 15 = 30 SCFM

Estimated reduction in power = 30 x (0.2 – 0.08) = 3.6 kW

10.5 Reducing system pressure

The system pressure, which is the pressure at which compressed air is supplied by the compressors, depends on factors such as the pressure requirements of the end users, pressure losses in the system, storage capacity and variation in demand.

Power consumed by compressors depends on the discharge pressure as expressed by the following relationship (which was derived earlier in chapter 8).

$$\dot{W} = \dot{m} R T_1 \left(\frac{n}{n-1}\right)\left[\left(\frac{P_2}{P_1}\right)^{\left(\frac{n-1}{n}\right)} - 1\right] \tag{10.6}$$

Therefore, a higher discharge pressure results in a higher compressor power (as the compressor has to work against a higher back pressure). The rule of thumb used in industry is a 7 to 8% increase in compressor power for every 1 bar increase in compressor discharge pressure.

Example 10.3

Compute the reduction in compressor power achievable when the system pressure is reduced from 9 bar to 7 bar (both absolute) for a compressor operating at 0.2 m^3/s of air (FAD). Air intake temperature is 30°C, intake pressure is 1 bar (absolute) and the compression follows $PV^{1.3}=C$. Density of air can be taken as 1.2 kg/m^3 while the specific gas constant R is 287 J/kg.K.

Solution

Using relationship (10.6),

At 9 bar,

$$\dot{W} = 0.2 \text{ x } 1.2 \text{ x } 287 \text{ x } (273+30)\left(\frac{1.3}{0.3}\right)\left[\left(\frac{9}{1}\right)^{\left(\frac{0.3}{1.3}\right)} - 1\right]$$

= 59.7 kW

At 7 bar,

$$\dot{W} = 0.2 \text{ x } 1.2 \text{ x } 287 \text{ x } (273+30)\left(\frac{1.3}{0.3}\right)\left[\left(\frac{7}{1}\right)^{\left(\frac{0.3}{1.3}\right)} - 1\right]$$

= 51.3 kW

Reduction in compressor power = 59.7 – 51.3 = 8.4 kW

(14% reduction in compressor power for a 2 bar reduction in pressure)

Not only does higher system pressure result in higher compressor pressure, it also leads to increased system leaks and increased compressed air consumption by some end users (which in turn results in more compressor power).

Therefore, system pressure should be set at the minimum value required to satisfy equipment requirements. In systems where a single user requires a very much higher pressure than the other users, standalone compressors should be used for the system that requires a higher pressure so that the system serving the other users can operate at a lower pressure.

10.6 Multistage and intercooling

The work required for compression depends on the compression ratio (ratio of discharge pressure to intake pressure) and the volume of air to be compressed. The amount of work required for compression is represented by the area enclosed in the P-V diagram (Figure 10.1 showing pressure vs volume) for the compression process.

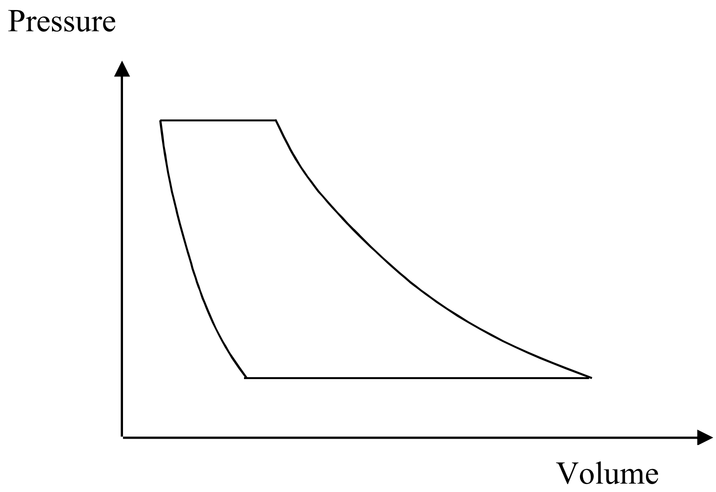

Figure 10.1 P-V diagram for single stage compression

In high-pressure applications, the compression can be divided into two or more stages where the air is compressed to an intermediate pressure and cooled before entering the next stage of compression. This helps to reduce the volume of air to be compressed in the subsequent stage. As shown in the P-V diagram for 2-stage compression (Figure 10.2), the work required, denoted by the area enclosed by the process, is reduced.

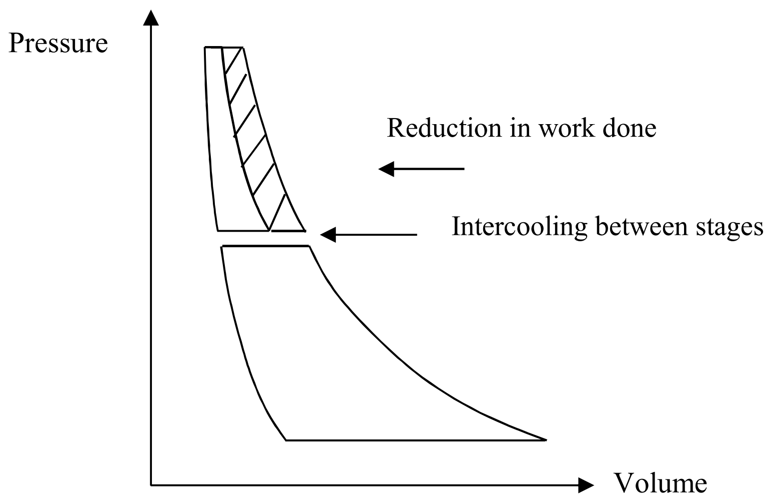

Figure 10.2 P-V diagram for two stage compression

Example 10.4

0.05 m^3/s of air (FAD) is to be compressed from 1 bar (absolute) and 30°C to 36 bar (absolute). Compute the compressor power for single stage and two-stage compression with intercooling which reduces the temperature of the air to 30°C. Estimate the reduction in compressor power for two-stage compression. Take R to be 287 J/kg.K and density of air at 30°C to be 1.2 kg/m^3. The compression pressure ratio for each stage is 1:6 and compression follows the path $PV^{1.3}=C$.

Solution

$\dot{m}$ = 0.05 m^3/s x 1.2 kg/m^3 = 0.06 kg/s

Power for single-stage using equation (10.1)

$\dot{W}$ = 0.06 x 287 x (273+30) x (1.3/0.3) x $[(36)^{0.3/1.3} - 1]$ Watts

= 29 kW

Power for two-stage

$\dot{W} = \dot{W}_1 + \dot{W}_2 = 2 \times \dot{W}_1$ (since mass flow rate and inlet temperature are same for both stages)

= 0.06 x 287 x (273+30) x (1.3/0.3) x $[(6)^{0.3/1.3} - 1]$ x 2 Watts

= 23 kW

Reduction in work done = (29 – 23) = 6 kW (20% reduction)

10.7 Heat recovery

Generally, about 80% of energy used in compression is rejected as heat. In air-cooled screw type package compressors, air used for removing heat from the aftercooler and oil cooler can be ducted and used for space heating where required. Alternatively, heat exchangers can be used to produce hot water.

Example 10.5

The input power to an air compressor motor is measured to be 100 kW (during load operation). The utilisation factor for the compressor is 60%. If 80% of the energy input to the compressor can be recovered as heat, compute the amount of energy that can be recovered daily in kWh (assume no heat recovery when the compressor is off or unloaded for 40% of the time).

Take the motor efficiency to be 90% and the efficiency of the drive system to be 95%.

Solution

Power at compressor = 100 x 0.9 x 0.95 = 85.5 kW

Total energy input per day (load operation) = 85.5 x 0.6 x 24 hours = 1231 kWh

Total amount of heat that can be recovered = 1231 x 0.8 = 985 kWh

Example 10.6

0.05 m^3/s of air (FAD) is to be compressed from 1 bar (absolute) and 30°C to 36 bar (absolute) in 2 stages of 1:6. Intercooling is provided between the two stages to cool the compressed air to 30°C. Compute the amount of heat rejected at the intercooler.

Take R to be 287 J/kg.K, density of air at 30°C to be 1.2kg/m^3 and specific heat capacity of air (Cp) to be 1.005 kJ/kg.K). The compression follows the path $PV^{1.3}=C$

Solution

Mass flow rate of air, $\dot{m}$ = 0.05 m^3/s x 1.2 kg/m^3 = 0.06 kg/s

For an ideal gas, $(P_1 \times V_1) / T_1 = (P_2 \times V_2) / T_2$

Also, $PV^n = C$

Therefore, $P_1 \times V_1^n = P_2 \times V_2^n$

$(T_2/T_1) = (P_2 \times V_2) / (P_1 \times V_1) = (P_2/P_1) \times (P_2/P_1)^{-1/n} = (P_2/P_1)^{1-1/n}$

Therefore, $T_2 = (P_2/P_1)^{1-1/n} \times T_1 = (6)^{1-1/1.3} \times 303 = 458$ K (temperature of air leaving the first stage of the compressor which is the temperature of air entering the intercooler)

Heat rejected at the intercooler,

$Q = m \times C_P. (T_{in} – T_{out}) = 0.06 \times 1.005 \times (458 – 303) = 9.35$ kW

10.8 Intake temperature

Compressor power consumption to compress a certain volume of air depends on the intake temperature of the air. If the intake air is hot, since the density of air is lower, the compressor needs to take in a larger volume of air to produce the same output.

Therefore, it is essential to provide adequate ventilation for compressor rooms to minimise the intake air temperature. Ventilation can be provided by natural means or by forced ventilation using fans and ducting systems.

Generally, a 5°C reduction in intake temperature of air results in a 1.5% reduction in compressor power.

Summary

This chapter described some of the common energy saving measures that can be implemented for compressed air systems to reduce the usage of compressed air and to improve the efficiency of the compression process. Various examples were used to illustrate the energy savings achievable for the saving measures.

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11.0 ENERGY AUDIT OF COMPRESSED AIR SYSTEMS

The main objectives of a compressed air system audit are to:

- Assess system performance
- Establish the energy baseline
- Identify opportunities for improvement

The audit should cover the air compressors, dryers, distribution system and the end users. The audit will usually consist of two parts, which are the measurements and site observations. The main aspects of the measurements and observations are listed below.

<u>Measurements</u>

- Compressed air demand (FAD)
- Pressure profile
- Compressed air temperature and dew point
- Compressor power
- Dryer power

<u>Site observations</u>

- Leak observation
- Pressure drop along distribution piping
- Identifying inappropriate usage
- Identifying wastage
- Identifying areas for improving maintenance
- Assessing potential for heat recovery

Learning Outcomes:

The main learning outcomes from this chapter are to understand:

1. The parameters to be measured in a compressed air system audit
2. How to compute key performance indicators
3. How to identify potential improvement measures

11.1 Measurements

Schematic diagram of a compressed air system showing location of sensors used for measurements is shown in Figure 11.1

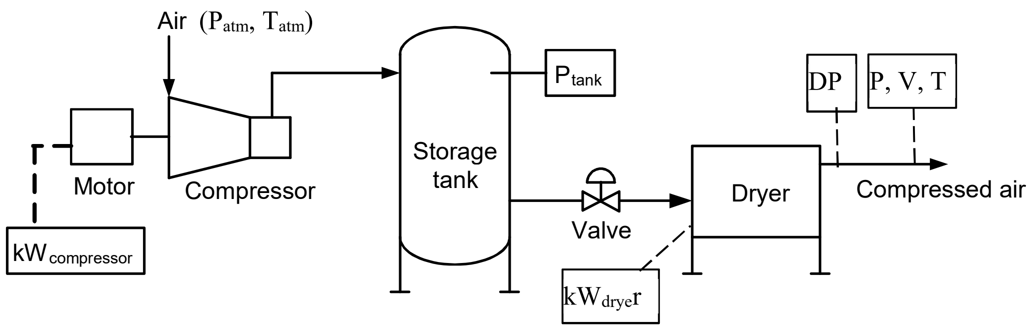

Figure 11.1 Measurement locations for a compressed air system

The different parameters:

V = volume flow rate at the measured point (m^3/s)

P = absolute pressure at the measurement point (bar)

P_{atm} = atmospheric pressure (bar)

P_{tank} = receiver pressure (bar)

T = temperature at the measurement point (K)

T_{atm} = temperature of ambient air (K)

DP = dew point temperature (°C)

$kW_{compressor}$ = compressor motor power (kW)

kW_{dryer} = dryer power (kW)

The values of P_{tank}, P, DP, $kW_{compressor}$ and kW_{dryer} can be plotted against time to obtain their profiles. An example of P_{tank} and $kW_{compressor}$ profiles is shown in Figure 11.2.

Other than plotting the profiles, the measured data can also be used to do the following:

- Estimate the FAD
- Compute the specific power
- Establish the compressor load / unload cycle
- Quantify the leakage rate

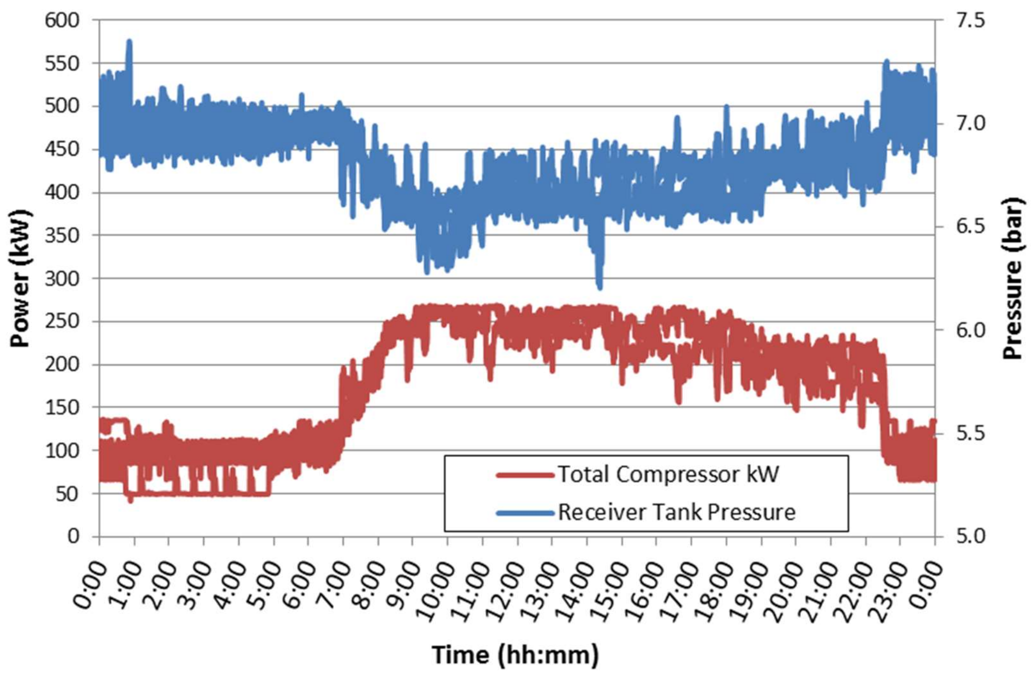

Figure 11.2 Pressure and compressor power profiles

11.2 Calculations

Free Air Delivery

For an ideal gas like air,

$$\frac{P_1 V_1}{T_1} = \frac{P_2 V_2}{T_2} \tag{11.1}$$

where,

P_1 and P_2 = absolute pressure at states 1 and 2

V_1 and V_2 = volume at states 1 and 2

T_1 and T_2 = absolute temperature at states 1 and 2

Using equation (11.1), the measured data can be used to compute the FAD as follows.

$$\frac{P_{atm} \cdot FAD}{T_{atm}} = \frac{P \cdot V}{T} \tag{11.2}$$

where,

P_{atm} = atmospheric pressure (bar)

T_{atm} = temperature of ambient air (K)

FAD = free air delivery (m^3/s)

P = absolute pressure at the measurement point (bar)

V = volume flow rate at the measured point (m^3/s)

T = temperature at the measurement point (K)

Using equation (11.2), the FAD can be calculated as follows.

$$\mathrm{FAD} = \frac{\mathrm{P\,.\,V\,.\,T_{atm}}}{\mathrm{T\,.\,P_{atm}}} \tag{11.3}$$

A typical plot of FAD and compressor power is shown in Figure 11.3.

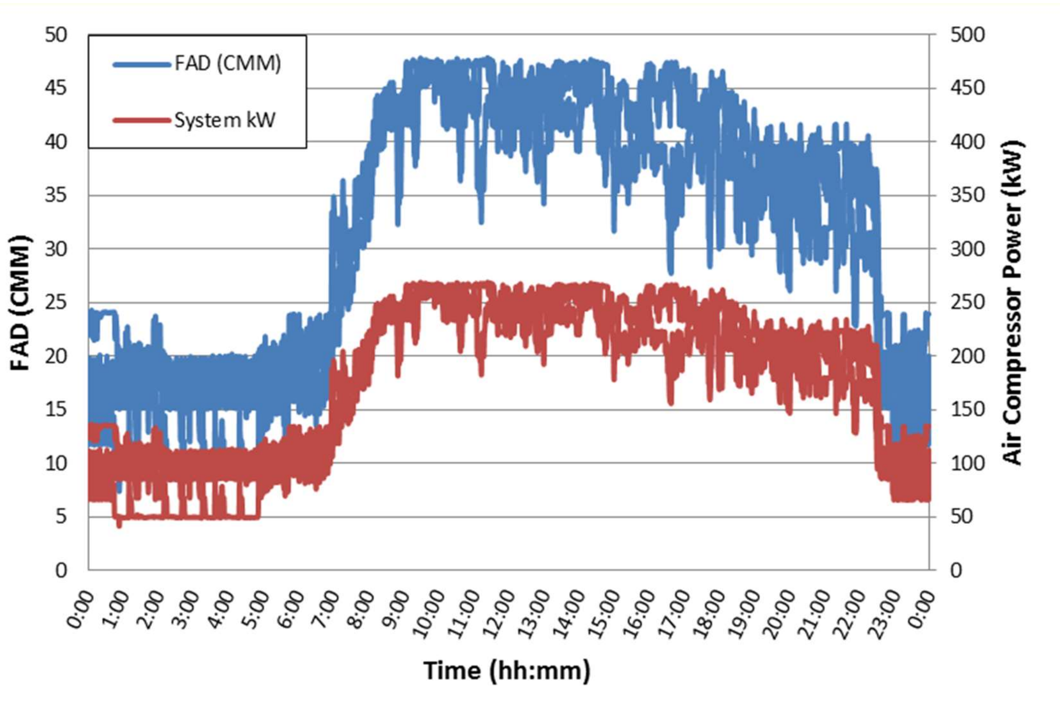

Figure 11.3 Typical FAD and compressor power profiles

Specific power

The specific power for the compressor and the system (including the dryers) can be computed using equations (11.4) and (11.5). For water cooled systems, the power drawn by the cooling tower fans and pumps are usually not considered in the specific power calculation.

$$\text{Compressor specific power} = \frac{\mathrm{kW_{compressor}}}{\mathrm{FAD}} \tag{11.4}$$

$$\text{Compressor system specific power} = \frac{\text{kW}_{\text{compressor}} + \text{kW}_{\text{dryer}}}{\text{FAD}} \quad (11.5)$$

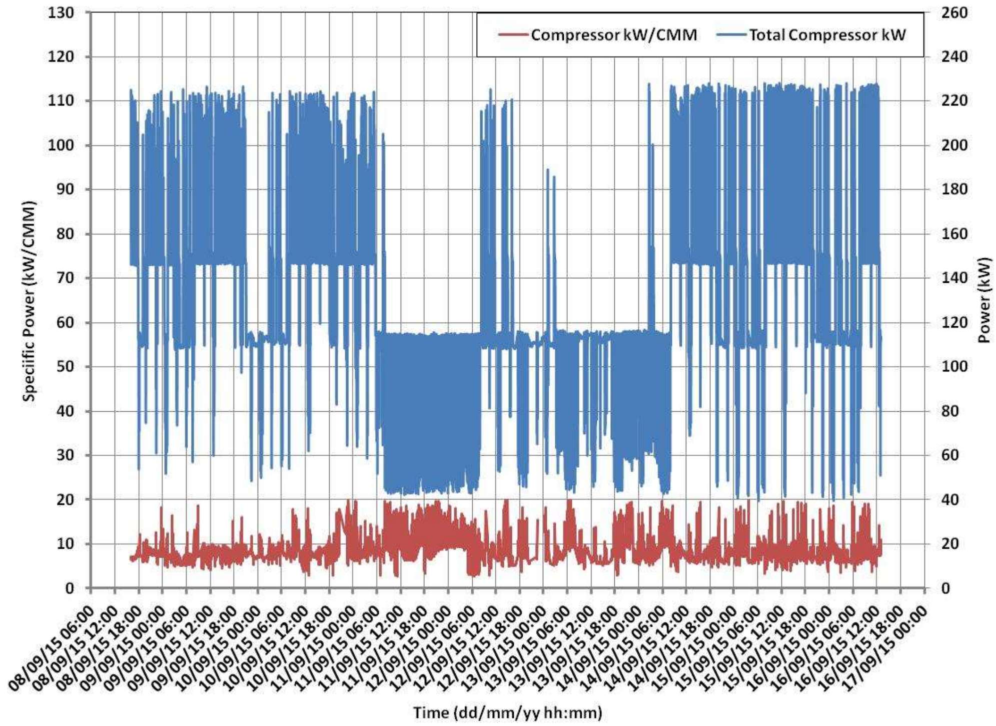

Figure 11.4 Typical plot of compressor power and specific power

Leakage rate

The total leakage rate for the compressed air system can be estimated by operating a compressor when all users of compressed air system are not in operation (during a plant shutdown) and measuring the load and unload time of the compressor.

Usually the time measurement will be carried out for a few load / unload cycles and the average values computed for the load and unload times using equation (11.6).

$$\text{System leakage } (\text{m}^3) = \frac{\text{FAD x T}_\text{L}}{\text{T}_\text{L} + \text{T}_{\text{UL}}} \quad (11.6)$$

Where,

FAD = free air delivery (m^3/min)

T_L = average load time (minutes)

T_{UL} = average unload time (minutes)

Compressor loading

Compressor loading during normal operation can be estimated using the average load and unload times for a compressor using equation (11.7). The load / unload times can be obtained from the compressor profile or the pressure as shown in Figure 11.5.

$$\text{Loading (\%)} = \frac{T_L}{T_L + T_{UL}} \text{ x } 100 \tag{11.7}$$

where

T_L = average load time (minutes)

T_{UL} = average unload time (minutes)

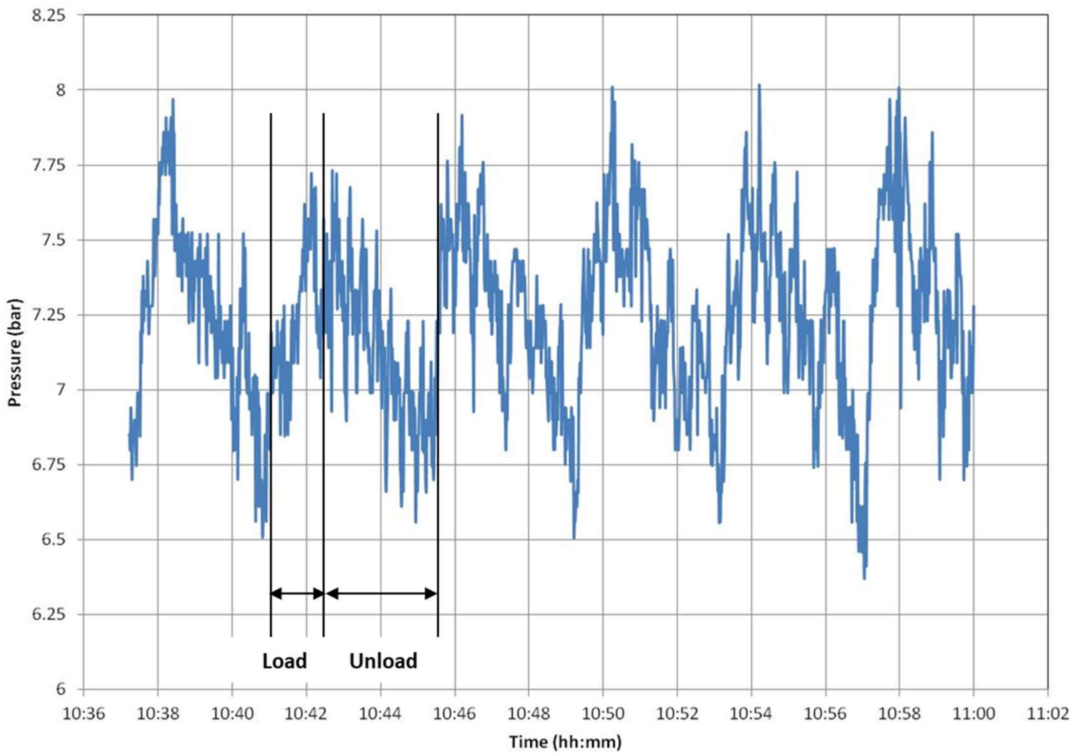

Figure 11.5 Plot showing compressor loading and unloading time

11.3 Observations

Leak observation

Locations of the individual leaks should be identified and tagged so that they can be rectified. Although major leaks can be identified by observation of sound, an ultrasonic leak detector would be required to identify most leaks.

During the audit, a leak detection exercise can be carried out by walking along the main distribution piping and the main users.

Pressure drop along distribution piping

The pressure in the compressed air system at various locations where pressure gauges are installed should be noted. Any significant or unusual drops in pressure should be investigated and action taken to eliminate them.

Identifying inappropriate usage

During the audit, the major compressed air users should be identified and visited. Any inappropriate users and applications such as tank agitations, component drying and blowing through nozzles should be considered for replacement with alternative systems such as blowers and air knives.

Identifying wastage

The most common wastage of compressed air is in cleaning. Since compressed air distribution systems in many manufacturing plants have provision with valves for connecting pneumatic tubes, often, hoses are connected to them and compressed air used for cleaning production machines, work surfaces and floors. Alternatives such as blowers or vacuum cleaners should be considered for such applications.

Identifying areas for improving maintenance

One of the key aspects of maintaining a compressed air system is to ensure that leaks are kept to a minimum. The other area with potential for improvement is the maintenance of filters. Numerous filters are used in compressed air systems to achieve the air quality required by the users. Gradually, contaminants accumulate in these filters and lead to a higher pressure drop. Therefore, maintenance should be carried out to replace or clean filters on a regular basis.

Assessing potential for heat recovery

Compressors generate a lot of heat as more than 80% of the input energy to the compressor motor is converted to heat. This heat can be used for various heating applications, rather than dissipating it to the environment. If hot air is required, air cooled intercoolers or aftercoolers can be used to extract the waste heat and produce hot air. For other applications where liquid heating is required, water cooled intercoolers and aftercoolers can be used.

Therefore, during the audit, the different heating applications in the plant electrical heating or fuel should be studied to assess the potential for using recovered heat from the air compressors.

11.4 Case study of a compressed air system energy audit

Information on the system

The compressed air system consists of three compressors with a total capacity of 1,334 CFM. Two out of the three compressors are VSD controlled while the other is fixed speed. Compressed air supply pressure set point for all compressors is set at 7.5 bars. Normally, one VSD compressor operates with the fixed speed compressor. The compressed air system has a receiver tank of 1,500 litres capacity.

Refrigerant type air dryers (operating at a dew point of $3^{0}C$), are installed after the filter and receiver tanks.

Data collection and analysis

The measured compressed air flow rate (free air delivery) in CMM (cubic metres per minute), system pressure profile and compressor power profile are shown in Figures 11.6 and 11.7 respectively.

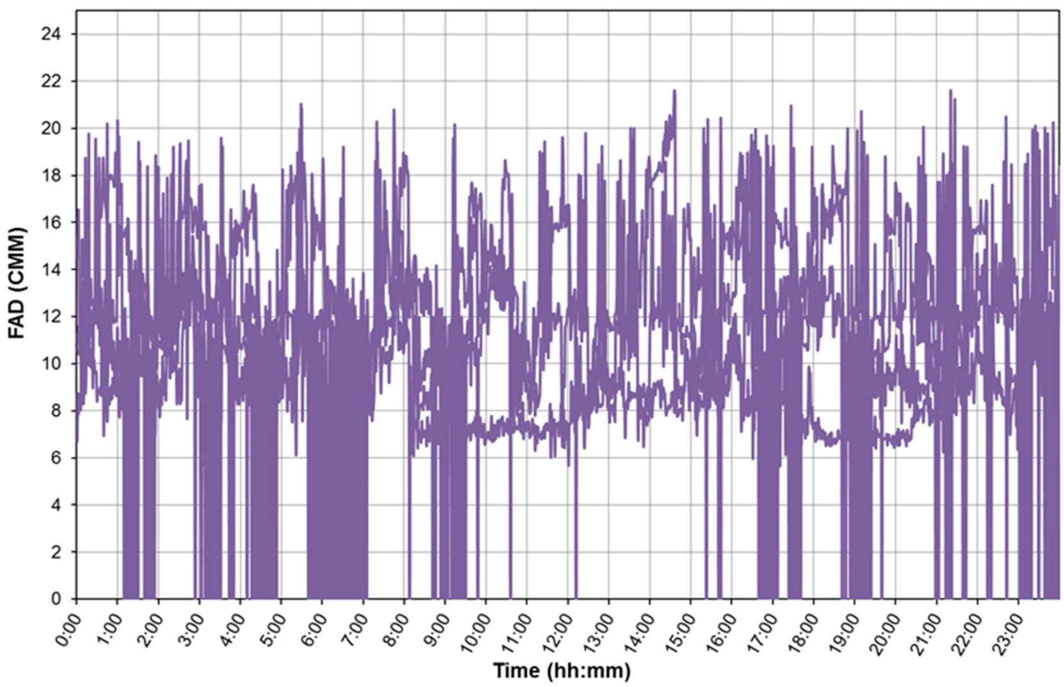

Figure 11.6 Compressed air FAD profile

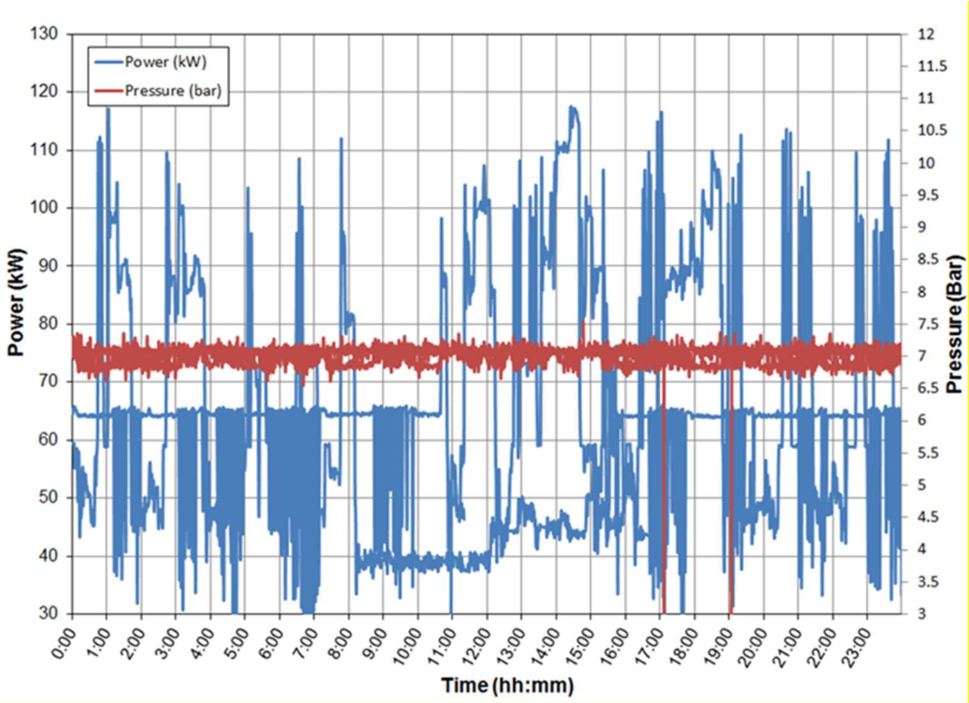

Figure 11.7 Pressure and power profiles

Summary of the measured and computed data is shown in Table 11.1.

Brand/ Model	Power (kW)		Capacity (CMM)	Design Efficiency (kW/ CMM)	Operating Efficiency without dryer (kW/ CMM)	Operating efficiency with dryer (kW/CMM)	Specific energy consumption (kWh/Nm^3)
	Rated	Measured	Rated				
VSD compressor 1	75	40.5	13.8	5.4			
Constant speed compressor	55	58.2	10.2	5.3	5	5.5	0.082
VSD compressor 2	75	N/A	13.8	5.4			

Table 11.1 Summary of data

Other findings:

A compressed air leakage test was conducted for the system and the results are summarised below.

Compressor load time = 50 s

Compressor unload time = 164 s

Capacity of operating compressor = 10.2 CMM

Leakage rate for the compressed air systems = (50/ (50+164)) x 10.2 CMM

= 2.4 CMM

Average compressed air demand = 11.5 CMM (from Figure 11.6)

Leakage rate = (2.4/ 11.5) x 100

= 21%

Main recommendations

As the system operating specific power was close to the design value, the main recommendations were to reduce the compressed air demand which were:

- Rectifying leaks
- Using high pressure blowers for aeration (instead of compressed air).

Estimated savings

The approximate energy and cost savings for the recommended measures are tabulated below.

Recommended measure	Energy savings (kWh/year)	Cost savings* ($/year)
Rectifying leaks	96,000	23,000
Using high pressure blowers for aeration	230,000	55,000

* Electricity cost is taken as $0.239 / kWh

Table 11.2 Summary of savings for case study

Summary

This chapter described the parameters to be measured during an audit of a compressed air system followed by how to compute the various key performance indicators and how to identify common improvement measures for compressed air systems.

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12.0 INTRODUCTION TO WASTE HEAT RECOVERY

Many industrial processes require heating of various fluids, which is achieved by equipment like steam heat exchangers, combustion heaters and electric heaters. All these heating processes require energy input. In the majority of cases, the heated fluids need to be cooled to lower temperatures after completion of the heating process. Cooling of a substance involves removal of heat, which in many instances requires energy. This requirement for heating and cooling is often simultaneous in many industrial processes. Therefore, if the heat removed from one fluid being cooled can be used to heat another fluid that requires the temperature to be increased, the overall energy efficiency of the manufacturing process can be improved.

Learning Outcomes:

The main learning outcomes from this chapter are to:

1. Provide an introduction to the concept of heat recovery
2. Understand the main modes of heat transfer
3. Compute the rate of heat transfer

12.1 Waste heat recovery

Waste heat is available from sources like furnaces, gas turbines, boilers, air compressors and various manufacturing processes and can be used to heat process fluids as shown in Figure 12.1.

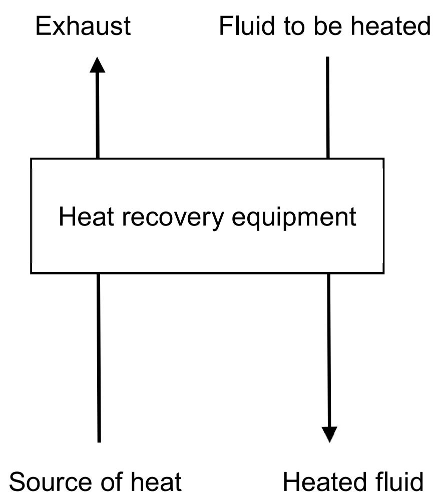

Figure 12.1 Arrangement of a heat recovery system

A very common application of heat recovery is in boilers. The exhaust from boilers is at a considerably high temperature and contains a significant amount of the input energy. This waste heat can be recovered using a heat recovery heat exchanger (economiser) and utilised to pre-heat the boiler feedwater (Figure 12.2) resulting in lower fuel usage by the boiler.

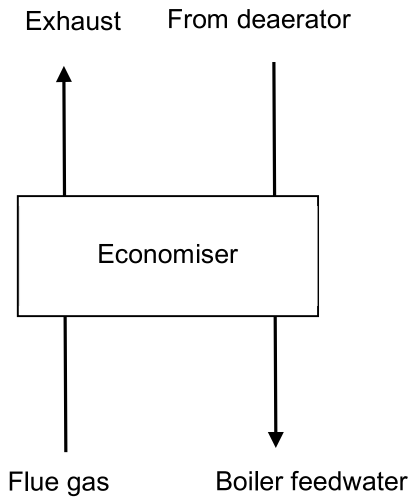

Figure 12.2 Typical heat recovery system for boiler exhaust

Some potential sources of waste heat and common heat recovery applications are summarised in Table 12.1.

Temperature	Source	Use
High (650°C and higher)	Exhaust from direct-fired industrial processes: Cement kiln (dry) 620 to 730°C Steel heating furnaces 925 to 1040°C Glass melting furnaces 980 to 1540°C Solid waste incinerators 650 to 980°C Fume incinerators 650 to 1430°C	Cogeneration
Medium (230 to 650°C)	Exhaust from: Steam boiler 230 to 480°C Gas turbine 370 to 540°C Reciprocating engine 230 to 595°C Heat-treating furnaces 425 to 650°C Drying and baking ovens 230 to 595°C	Steam generation

Low (32 to 230ºC)	Process steam condensate 55 to 90°C Cooling water from: Furnace doors 32 to 55°C Air compressors 27 to 50°C Internal combustion engines 65 to 120°C Hot-processed liquids 32 to 230°C Hot-processed solids 95 to 455°C	Supplementing heating and Preheating

Table 12.1 Potential applications for heat recovery

The main equipment usually required for heat recovery is a heat exchanger. To better understand the operation and selection of heat exchangers for various applications, it is first necessary to have a basic knowledge of the principles of heat transfer. The following sections of this chapter will explain the modes of heat transfer, which are conduction, convection and radiation.

12.2 Conduction

Heat transfer by conduction takes place when there is a temperature gradient across a solid object (as shown in Figure 12.3). The rate of heat transfer depends on the thermal conductivity of the material, its thickness, the temperature gradient, and the surface area available for heat transfer.

The rate of heat transfer by conduction can be expressed using Fourier's law as follows:

$$Q_{\mathrm{cond}} = k A \frac{dT}{dx} \qquad (12.1)$$

where,

Q_{cond} = heat transfer rate by conduction (W)

k = thermal conductivity of the material (W/m.K)

A = area perpendicular to heat flow (m^2)

dT = temperature gradient ($T_1 - T_2$) (K)

dx = thickness (m)

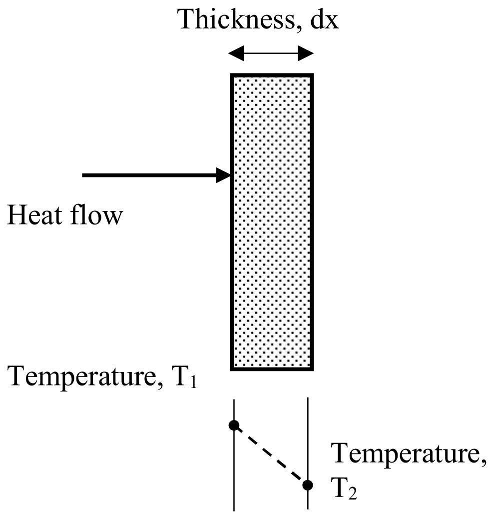

Figure 12.3 Heat transfer by conduction

The above can be compared with Ohm's law, where,
Potential difference V = current (I) x resistance (R)

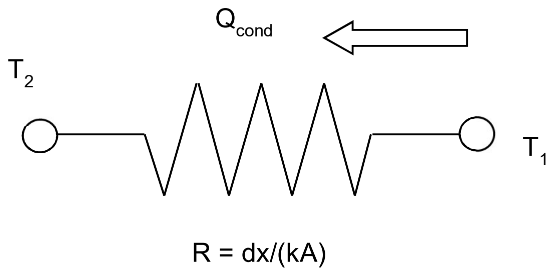

Figure 12.4 Thermal analogy of Ohm's law

The thermal analogy (see Figure 12.4) would be, $(T_1 – T_2) = dT = Q \times R$

Therefore, the thermal resistance R = dx / K A

Thermal conductivity values for some common materials used for constructing heat exchanger components are provided in Table 12.2.

Material	Thermal conductivity (W/m.K)
Aluminium	200 - 250
Copper	380 - 400
Carbon steel (max. 0.5% C)	54
Carbon steel (max. 1.5% C)	36
Stainless steel	16
Titanium	19
Cupro-nickel	30 - 70
Inconel	19
Chrome-moly steel	30 - 40
Chromium	90
Monel	26
Nickel	62
Brass	100
Hastelloy	9 - 10
Zinc	116

Table 12.2 Approximate thermal conductivity values of materials used in heat exchanger construction

Example 12.1

The two surfaces of a 6 mm thick copper plate are maintained at 35°C and 40°C. Compute the conduction heat transfer rate for 1 m^2 of the plate. Take the thermal conductivity of the copper plate as 400 W/m.K

Solution

Using equation (12.1),

$$Q_{cond} = k\,A\,\frac{dT}{dx} = 400 \times 1 \times \frac{(40-35)}{0.006}$$

Q_{cond} = 333,333 W = 333 kW

12.3 Convection

Heat transfer by convection takes place when a fluid comes into contact with a surface at a different temperature (Figure 12.5). Heat transfer by convection can take place at both the inner and outer surfaces of an object.

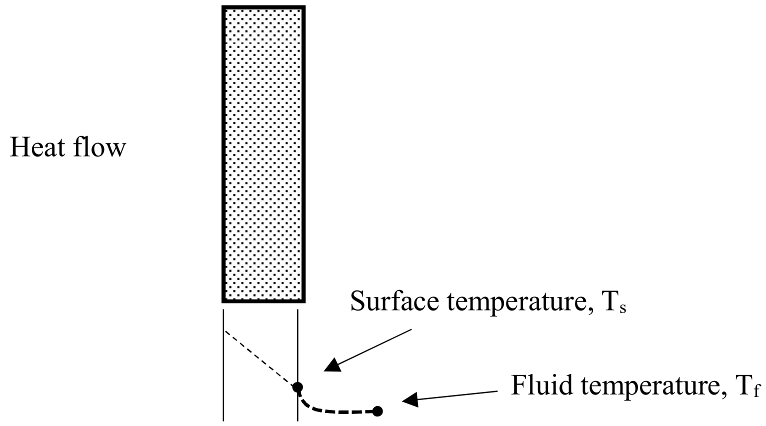

Figure 12.5 Convective heat transfer at the inner surface

Heat transfer by convection can be expressed using Newton's law of cooling, as follows:

$$Q_{conv} = h_c A (T_s - T_f) \quad (12.2)$$

where,

Q_{conv} = heat transfer rate by convection (W)

h_c = surface heat transfer coefficient (W/m^2.K)

A = surface area (m^2)

T_s = surface temperature (K)

T_f = fluid temperature (K)

The amount of convective heat transfer depends on the surface area, temperature difference between the surface and fluid, and the surface heat transfer coefficient. The surface heat transfer coefficient is dependent mainly on the velocity of the fluid motion over the solid surface.

The Ohm's law analogy is shown in Figure 12.6.

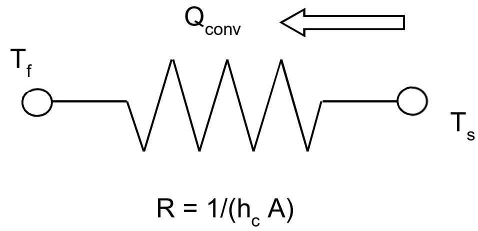

Figure 12.6 Ohm's law analogy applied to convective heat transfer

Typical surface heat transfer coefficient values are provided in Table 12.3.

Condition	**Heat transfer coefficient (W/m^2.K)**
Free convection to air with temperature difference < 100°C (see Figure 12.7)	1 to 10
Free convection to a gas	0.5 to 1000
Free convection to a liquid	100 to 5000
Forced convection to a gas	10 to 1000
Forced convection to a liquid	100 to 10,000
Forced convection to a liquid metal	1000 to 50,000
Nucleate boiling	500 to 50,000
Filmwise condensation	500 to 50,000

Table 12.3 Surface heat transfer coefficient values

For free convection, the heat transfer coefficient for horizontal and vertical plates can be estimated from Figure 12.7.

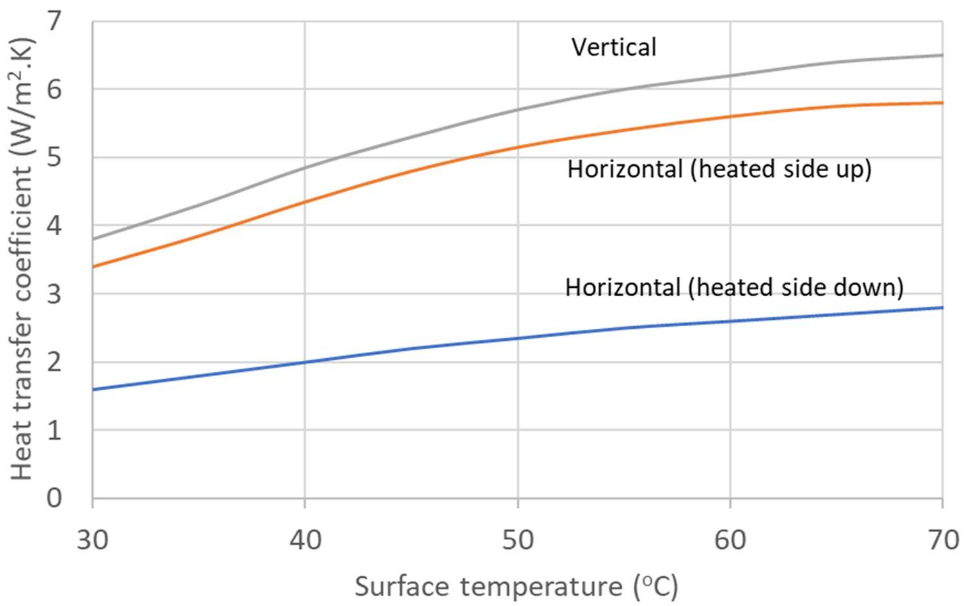

Figure 12.7 Heat transfer coefficient for natural convection

For forced convection, the heat transfer coefficient can be estimated using the following relationship.

$h_c = 10.45 - v + 10\ v^{1/2}$ (12.3)

where,

h_c = convective heat transfer coefficient (W/m^2.K)

v = relative velocity between surface and air (m/s)

Figure 12.8 shows the relationship between air velocity and heat transfer coefficient.

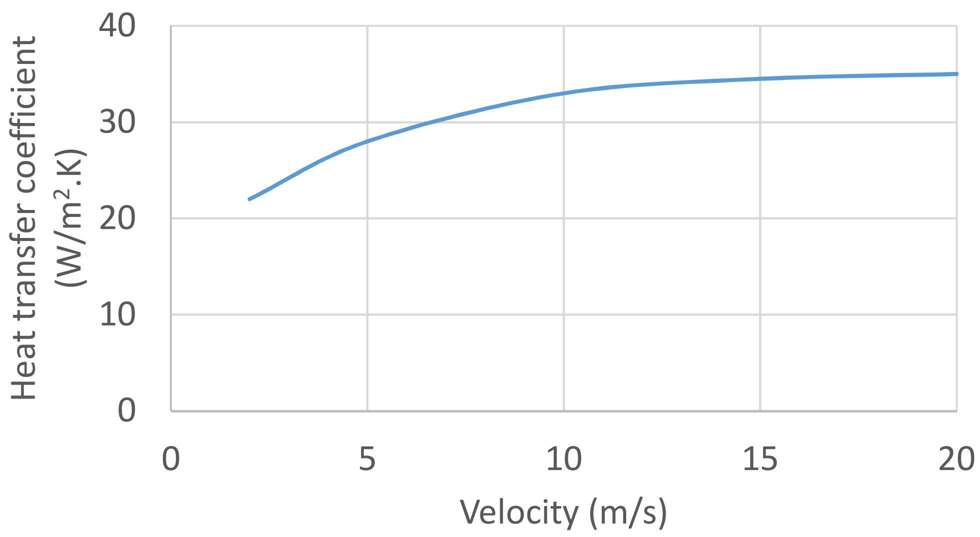

Figure 12.8 Convective heat transfer coefficient values at different air velocities

Example 12.2

The outer surface of a vertical plate in contact with air is maintained at 70°C. The air in contact with the plate is at 30°C. Estimate the heat transfer rate per m^2 of the plate to the air under natural convection and if the air is flowing over the plate at 10 m/s.

Solution

Natural convection

From Figure 12.7, for vertical plate with a surface temperature of 70°C, the convective heat transfer coefficient, h_c is about 6.5 W/m^2.K

From equation (12.2),

$Q_{conv}= h_c\ A\ (T_s - T_f) = 6.5 \times 1 \times (70 - 30) = 260$ W

For forced convection

From equation (12.3), for a velocity of 10 m/s,

$h_c = 10.45 - v + 10\ v^{1/2} = 10.45 - 10 + 10\ (10)^{0.5}$

$h_c = 32$ W/m^2.K

From equation (12.2),

$Q_{conv}= h_c\ A\ (T_s - T_f) = 32 \times 1 \times (70 - 30) = 1280$ W

12.4 Radiation

Heat transfer by radiation takes place due to electromagnetic waves. The amount of radiant heat transmission between two surfaces depends on the absolute surface temperatures of the bodies exchanging heat and the area of the body at the higher temperature. Radiant heat transfer can be expressed as follows:

$$Q_{rad} = \sigma\, \varepsilon_1\, A_1\, (T_1^4 - T_2^4) \qquad (12.4)$$

where,

σ = Stefan-Boltzmann constant = 5.67 x 10^{-8} W/m^2.K^4

A_1 = area of surface 1 (m^2)

ε_1= emissivity of surface 1

T_1 = absolute temperature of surface 1 (K)

T_2 = absolute temperature of surface 2 (K)

Typical emissivity values are given in Table 12.4.

Material	Emissivity
Aluminium sheet	0.09
Cast iron	0.44
Copper (polished)	0.02 – 0.05
Copper (oxidised)	0.8
Copper nickel (polished)	0.06
Nickel polished	0.072
Mild steel (oxidised)	0.79
Mild steel (polished)	0.07
Stainless steel (polished)	0.075
Galvanised steel (new)	0.23

Table 12.4 Typical emissivity values

Radiant heat transfer can also be described by a simple expression using the radiant heat transfer coefficient (h_r), as follows:

$$Q_{rad} = h_r\, A_1\, (T_1 - T_2) \qquad (12.5)$$

Example 12.3

Compute the radiative heat transfer rate (per m^2) from a polished copper surface having an emissivity of 0.03 and maintained at 100°C to the surroundings at 30°C. Assume that all the heat radiated is absorbed by the surroundings.

Solution

T_1 = 273 + 100 = 373 K

T_2 = 273 + 30 = 303 K

From equation (12.4),

$Q_{rad} = \sigma \, \varepsilon_1 \, A_1 \, (T_1^4 - T_2^4)$

$= 5.67 \times 10^{-8} \times 0.03 \times 1 \times (373^4 - 303^4)$

= 18.6 W

12.4 Heat transfer through composite materials

Often, heat transfer takes place by conduction through composite materials made up of a few layers of different materials. In addition, there can be convective heat transfer at the inner and outer surfaces, as shown in Figure 12.7.

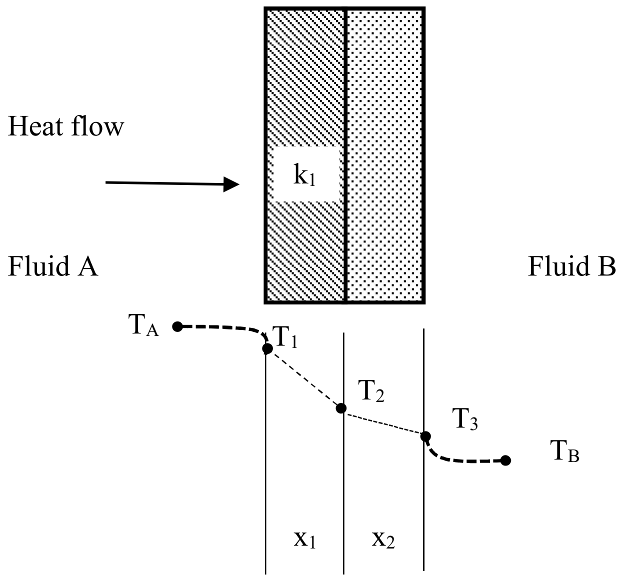

Figure 12.7 Heat transfer across a composite material

The rate of heat transfer, Q

$= h_A \, A \, (T_A - T_1) = (k_1/x_1) \, A \, (T_1 - T_2) = (k_2/x_2) \, A \, (T_2 - T_3) = h_B \, A \, (T_3 - T_B)$

The above can be rearranged and summed which will result in the following:

$$\frac{Q}{A} = \left[\frac{1}{\frac{1}{h_A}+\frac{x_1}{k_1}+\frac{x_2}{k_2}+\frac{1}{h_B}}\right](T_A - T_B) \qquad (12.6)$$

Equation (12.6) can be rewritten as,

$Q = U\ A\ \Delta T$

where the overall heat transfer coefficient,

$$U = \left[\frac{1}{\frac{1}{h_A}+\frac{x_1}{k_1}+\frac{x_2}{k_2}+\frac{1}{h_B}}\right] \qquad (12.7)$$

Example 12.4

Heat is transferring from flue gas of temperature 140°C to water stream of temperature 30°C through a flat steel plate of thermal conductivity 40 W/m K. The steel plate thickness, length and width are 3 mm, 1 m and 1.5 m, respectively. Convective heat transfer coefficients of flue gas and water are 10 W/m^2 K and 100 W/m^2 K respectively. Calculate overall heat transfer coefficient and heat transfer rate.

Solution

k= 40 W/m K

h_A = 10 W/m^2 K

h_B = 100 W/m^2 K

A = 1 x 1.5 = 1.5 m^2

x = 3 mm =3/1000 = 0.003 m

From equation (12.7),

$$U = \left[\frac{1}{\frac{1}{h_A}+\frac{x_1}{k_1}+\frac{x_2}{k_2}+\frac{1}{h_B}}\right]$$

$$U = \left[\frac{1}{\frac{1}{10}+\frac{0.003}{40}+\frac{1}{100}}\right] \quad = 9.9\ \text{W/m}^2.\text{K}$$

Using equation (12.6),

$Q = U\,A\,\Delta T = 9.9 \times 1 \times (140 - 30)$

$= 1089$ W

12.5 Radial heat transfer

Often, heat transfer between two fluids takes place through an annular surface such as across a pipe surface, as shown in Figure 12.8.

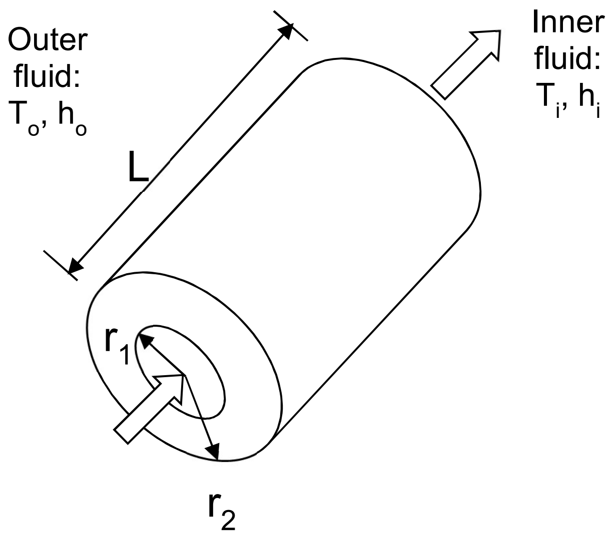

Figure 12.8 Heat transfer through an annular surface

Equation (12.1) can be expressed as

$$Q_{cond} = K\,A\,\frac{dT}{dr} \qquad (12.8)$$

where

k = thermal conductivity of the material (W/m.K)

A = area (perpendicular to heat flow) = 2π r L (m^2)

dT = temperature gradient, ($T_1 - T_2$) (K)

dr = thickness, ($r_2 - r_1$) (m)

rearranging equation (12.7),

$$Q \int_{r_1}^{r_2} \frac{dr}{r} = -2\pi\,k\,L \int_{T_1}^{T_2} dT \qquad (12.9)$$

Therefore, $Q = \frac{2\pi\, k\, L\, (T_1 - T_2)}{\ln\frac{r_2}{r_1}}$

Heat flow at the inner and outer surfaces by convection,

$Q_i = h_i\, 2\pi r_1\, L\, (T_i - T_1)$

$Q_o = h_o\, 2\pi r_2\, L\, (T_2 - T_o)$

Since, $Q = Q_i = Q_o$

$$\frac{Q}{L} = \left[\frac{1}{\frac{1}{2\pi r_1 h_i} + \frac{\ln\frac{r_2}{r_1}}{2\pi k_1} + \frac{1}{2\pi r_2 h_o}}\right](T_i - T_o) \qquad (12.10)$$

Example 12.5

A liquid at a temperature of 80^{o}C flows through a stainless pipe having an inner diameter of 100mm. The thickness of the pipe is 10mm. The thermal conductivity of the pipe wall material is 16 W/m.K. The pipe outer surface is exposed to air at 35^{o}C. Compute the heat loss through the pipe wall to the outside air per metre length of pipe. The convective heat transfer coefficients at the inner and outer surfaces of the pipe are 50 and 5 W/m.K, respectively.

Solution

T_i = 80^{o}C

T_o = 35^{o}C

r_1 = 0.05 m

r_2 = 0.06 m

k = 16 W/m.K

h_i = 50 W/m^2.K

h_o = 5 W/m^2.K

Using equation (12.10)

$$\frac{Q}{L} = \left[\frac{1}{\frac{1}{2\pi r_1 h_i} + \frac{\ln\frac{r_2}{r_1}}{2\pi k_1} + \frac{1}{2\pi r_2 h_o}}\right](T_i - T_o) = \left[\frac{1}{\frac{1}{2\pi 0.05 \text{ x } 50} + \frac{\ln\frac{0.06}{0.05}}{2\pi \text{ x}16} + \frac{1}{2\pi 0.06 \text{ x } 5}}\right](80 - 35)$$

= 378 W/m

Summary

The various modes of heat transfer were explained in this chapter. The key parameters that control heat transfer were identified and the heat transfer rate was computed for different situations. The application of these concepts in heat recovery systems will be discussed in the subsequent chapters of this reference manual.

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13.0 HEAT EXCHANGERS

Heat exchangers are devices used to transfer heat from one fluid to another. There are many types of heat exchanger design available to suit the various heat transfer requirements. In general, two fluids are separated by a wall and heat flows through the wall from the fluid at the higher temperature to the other fluid. Heat transfer through the wall is by conduction while convective heat transfer takes place between the surface and the fluid. If the temperature difference between the surface and the fluid is large, heat transfer by radiation may also occur.

Heat exchangers can be classified according to the flow arrangement of the two fluids exchanging heat. This chapter describes the main types of heat exchangers and their performance characteristics.

Learning Outcomes:

The main learning outcomes from this chapter are to understand:

1. The different types of heat exchangers
2. The main characteristics of heat exchangers

13.1 Parallel-flow heat exchangers

In parallel-flow heat exchangers, both the fluids enter the heat exchanger at the same end and flow parallel to each other before being discharged at the opposite end. The arrangement and temperature profile for a concentric tube (double pipe) parallel-flow heat exchanger are shown in Figures 13.1 and 13.2. One main disadvantage of this type of heat exchanger is that the temperature difference between the two fluids, which is the driving potential for heat transfer, reduces with distance from the inlet to the outlet.

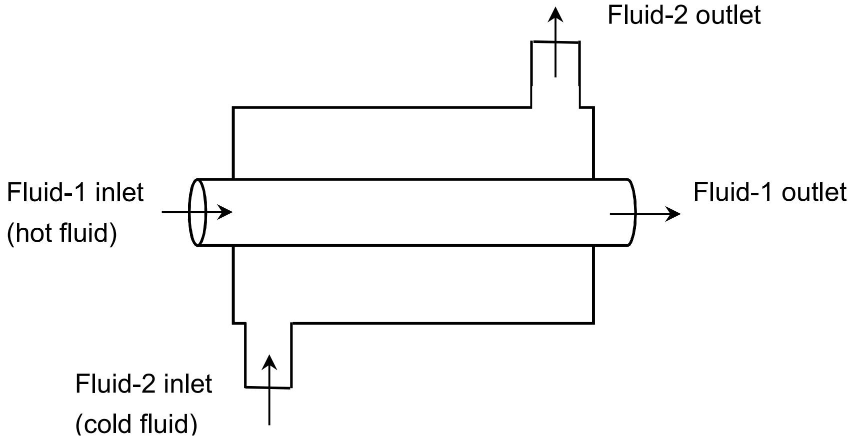


Figure 13.1 Arrangement of a parallel-flow heat exchanger

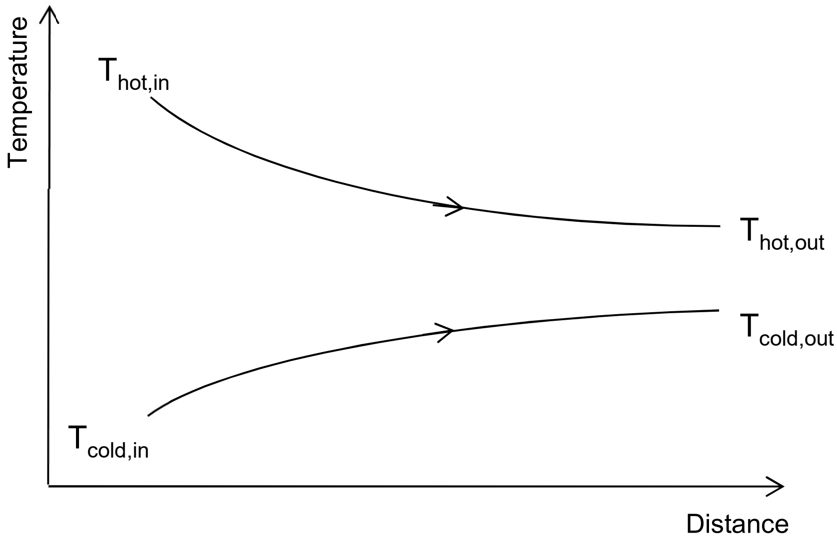


Figure 13.2 Temperature profile for a parallel-flow heat exchanger

13.2 Counterflow heat exchangers

In counterflow devices, the two fluids enter at the two opposite ends of the heat exchanger and, as a result, are able to maintain a relatively better temperature difference when compared to parallel-flow exchangers. Therefore, for the same heat transfer capacity, a counter flow heat exchanger will require a shorter length than a parallel type one.

The typical arrangement and temperature profile for a counterflow heat exchanger are shown in Figures 13.3 and 13.4.

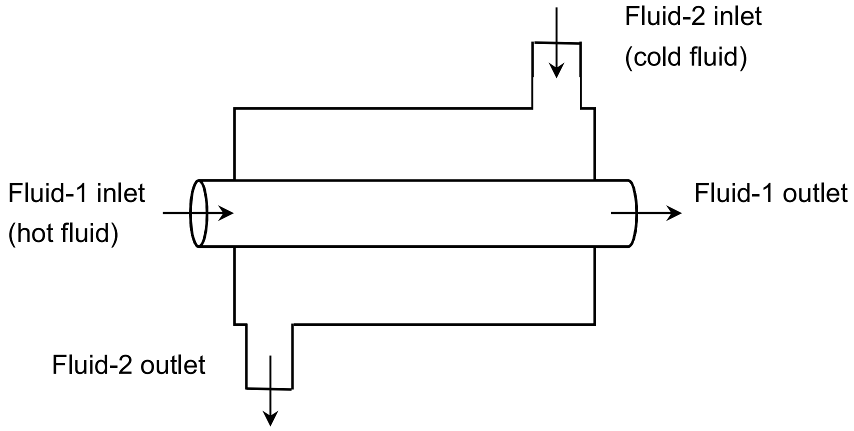


Figure 13.3 Arrangement of a counter flow heat exchanger

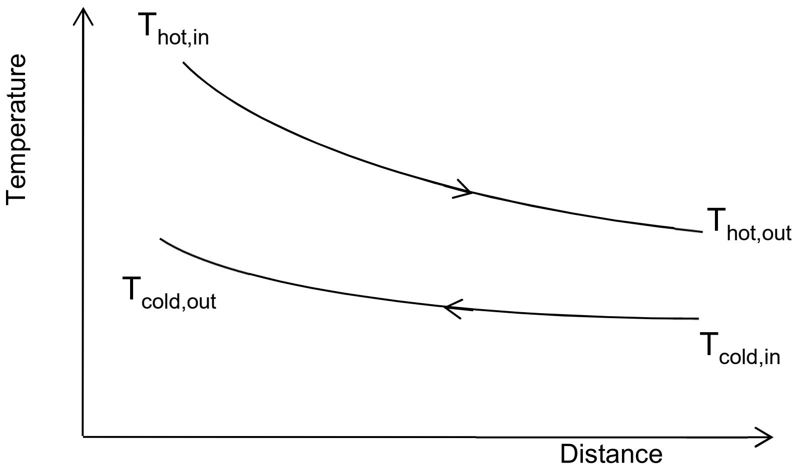


Figure 13.4 Temperature profile for a counter flow heat exchanger

13.3 Cross-flow heat exchangers

In cross-flow heat exchangers, the two fluids generally flow perpendicular to each other. The typical arrangement of a cross-flow heat exchanger is shown in Figure 13.5. They are commonly used to exchange heat between a liquid and a gas.

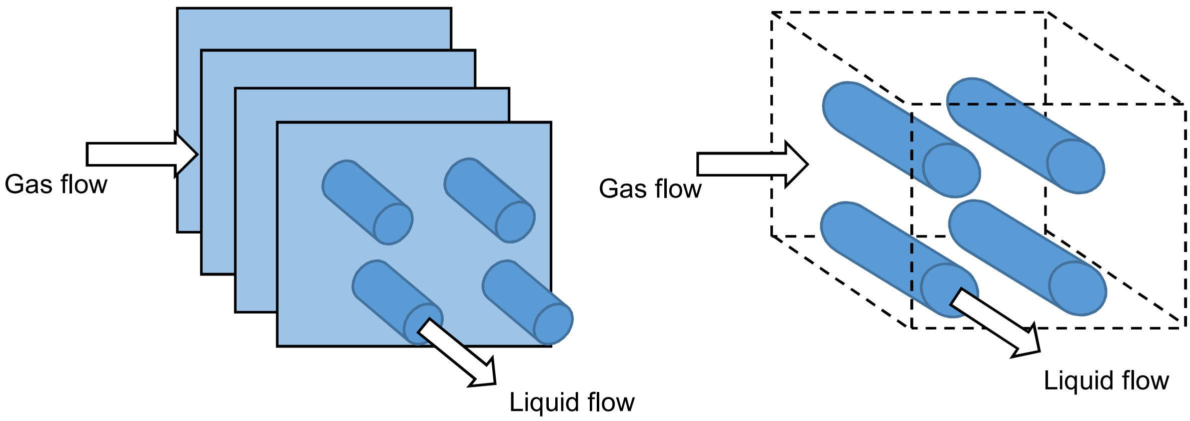


Figure 13.5 Arrangement of a cross-flow heat exchanger

13.4 Performance of heat exchangers

Without phase change

If we consider the heat exchangers shown in Figures 13.1 and 13.3, the heat transfer rate from the hot fluid and cold fluid can be expressed as follows

$$Q = m_{hot} \times Cp_{hot} \times (T_{hot\ in} - T_{hot\ out}) = m_{cold} \times Cp_{cold} \times (T_{cold\ out} - T_{cold\ in}) \qquad (13.1)$$

where,

m_{hot} = mass flow rate of the hot fluid (kg/s)

Cp_{hot} = specific heat capacity of the hot fluid (kJ/kg.K)

$T_{hot\ in}$ = inlet temperature of the hot fluid (K)

$T_{cold\ in}$ = inlet and outlet temperature of the cold fluid (K)

m_{cold} = mass flow rate of the cold fluid (kg/s)

Cp_{cold} = specific heat capacity of the cold fluid (kJ/kg.K)

$T_{hot\ out}$ = outlet temperature of the hot fluid (K)

$T_{cold\ out}$ = outlet temperature of the cold fluid (K)

The specific heat capacities of some common fluids are shown in Table 13.1.

Material	Specific heat capacity (J/kg.K)
Water (20°C)	4180
Water (100°C)	4220
Air (20°C)	1005
Air (100°C)	1010
Crude oil	≈ 2000
Steam	1900
Alcohol	2500
Fuel oil	1700 - 2000

Table 13.1 Specific heat capacity values of common fluids

Example 13.1

In a water to water heat exchanger, hot water from an industrial cooling process is used to heat cold water used in another process. The hot water enters the heat exchanger at 90°C and exits it at 45°C. If the hot water flow rate is 5 kg/s, compute the total amount of heat transferred from the hot water to the cold water. If the cold water flow rate is also 5 kg/s and enters the heat exchanger at 30°C, estimate the temperature of this stream of water at the exit of the heat exchanger. Take the specific heat capacity of water to be 4.19 kJ/kg.K.

Solution

Using equation (13.1),

$Q = m_{hot} \times Cp_{hot} \times (T_{hot\ in} - T_{hot\ out})$

$Q = 5 \times 4190 \times (90 - 45) = 942.75$ kW

Also, $Q = m_{cold} \times Cp_{cold} \times (T_{cold\ out} - T_{cold\ in})$

Therefore, $942{,}750 = 5 \times 4190 \times (T_{cold\ out} - 30)$

$T_{cold\ out} = 75^{o}C$

With phase change

If one or more of the fluids change phase, such as steam condensation or evaporation of water, the total heat transfer rate will be the sum of the sensible and latent heat transfers and equation (13.1) has to be modified to include latent heat as expressed in equation (13.2).

$$Q = (m \times Cp \times \Delta T) + (m \times h_{fg}) \qquad (13.2)$$

where,

ΔT = change in temperature of the fluid

h_{fg} = latent heat of vapourisation or condensation

Example 13.2

Exhaust flue gas from a furnace is used to heat water using a heat recovery heat exchanger. Water enters the heat recovery unit at of 45°C and the flow rate is 1 m^3/hour. Compute the heat recovery rate for the following two cases:

a) Water exits the heat exchanger in liquid state at 100°C

b) Water exits the heat exchanger as saturated vapour (low-pressure steam) at 100°C.

Take the specific heat capacity of water to be 4.195 kJ/kg.K, density of water to be 1000 kg/m^3 and the latent heat of vapourisation of water at 100°C to be 2257 kJ/kg

Solution

a) Using equation (13.1),

$Q = m_{cold} \times Cp_{cold} \times (T_{cold\ out} - T_{cold\ in})$

$Q = (1000/3600) \times 4.195 \times (100 - 45) = 64.1$ kW

b) Using equation (13.2),

$Q = (m \times Cp \times \Delta T) + (m \times h_{fg})$

$Q = ((1000/3600) \times 4.195 \times (100 - 45)) + ((1000/3600) \times 2257) = 691$ kW

(note that the heat transfer rate with phase change is many times the value of pure sensible heat transfer).

Concept of LMTD

The total heat transfer rate can also be expressed as:

$$Q = U \times A \times \Delta T_m \qquad (13.3)$$

where,

U = overall heat transfer coefficient for the heat exchanger

A = heat transfer area

ΔT_m = mean temperature difference between the fluids

The temperature difference between the two fluids (ΔT) varies with position and can be expressed as a mean value using the log mean temperature difference (LMTD). The LMTD is a logarithmic average of the temperature differences between the two fluids at each end of the heat exchanger.

$$LMTD = \frac{\Delta T_2 - \Delta T_1}{ln\left(\frac{\Delta T_2}{\Delta T_2}\right)} \qquad (13.4)$$

where

ΔT_2 = temperature difference between the fluids at one end of the heat exchanger

ΔT_1 = temperature difference between the fluids at the other end of the heat exchanger

Equation (13.3) can be written as,

$$Q = U \times A \times LMTD \qquad (13.5)$$

The above relationship applies to both parallel flow and counter flow heat exchangers.

Example 13.3

A double pipe counter flow heat exchanger is used to heat water from 35°C to 90°C. The water mass flow rate is 1 kg/s. The water stream is heated using hot oil from a process at 140°C with a mass flow rate of 2 kg/s. The inner diameter of the heat exchanger pipe is 100 mm. The overall heat transfer coefficient of the heat exchanger (U) is 500 W/m·K and the specific heat capacity of water and oil are 4.19 kJ/kg.K and

1.9 kJ/kg.K. Compute the LMTD and length of the heat exchanger needed to achieve the required heating.

Solution

From equation (13.1),

$Q = m_{cold} \times Cp_{cold} \times (T_{cold\ out} - T_{cold\ in})$

$Q = 1 \times 4.19 \times (90 - 35) = 230$ kW

Also, $Q = m_{hot} \times Cp_{hot} \times (T_{hot\ in} - T_{hot\ out})$

Therefore, $Q = 230 = 2 \times 1.9 \times (140 - T_{hot\ out})$

$T_{hot\ out} = 79.5^{o}C$

From equation (13.4), $LMTD = \frac{\Delta T_2 - \Delta T_1}{ln\left(\frac{\Delta T_2}{\Delta T_2}\right)}$

$\Delta T_2 = (140 - 90) = 50^{o}C$

$\Delta T_1 = (79.5 - 35) = 44.5^{o}C$

$LMTD = (50 - 44.5) / \ln (50/44.5) = 47.2^{o}C$

$Q = U \times A \times LMTD$

$230{,}000 = 500 \times A \times 47.2$

Therefore, the surface area for heat transfer, $A = 9.75\ m^2$

Also, $A = \pi \times D \times Length$

Therefore, $9.75 = 3.14 \times 0.1 \times L$

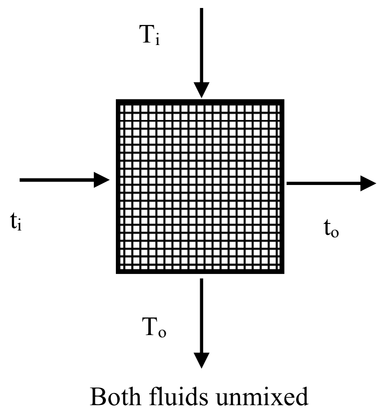
The required heat exchanger length, $L = 31$ m

In the case of cross flow heat exchangers where one fluid is at a fixed temperature at all locations (heat sink), the log mean temperature difference can be expressed using the LMTD for a counter flow heat exchanger multiplied by a correction factor where:

$$LMTD_{cross\ flow} = F \times LMTD_{counter\ flow} \qquad (13.6)$$

Correction factors "F" are available in charts for various configurations.

The correction factors used are dependent on whether a fluid is unmixed (confined in a channel) or unmixed (unconfined and free to contact several heat transfer surfaces). Figure 13.6 provides correction factors when both fluids are unmixed (eg. car radiator) while Figure 13.7 provides correction factors when one fluid is mixed and the other is unmixed (eg. boiler flue gas economiser).



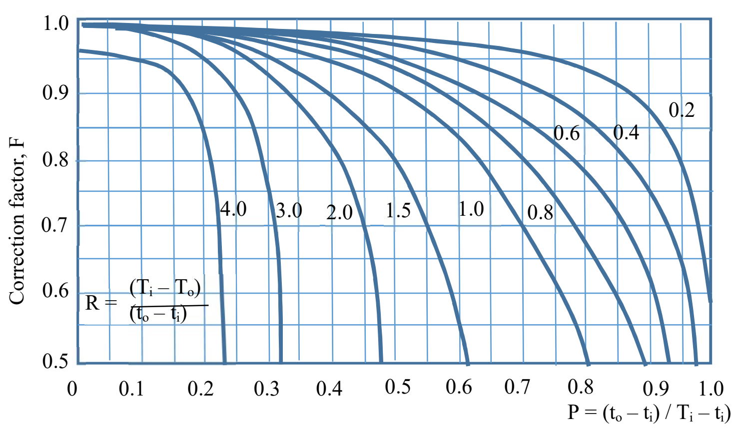
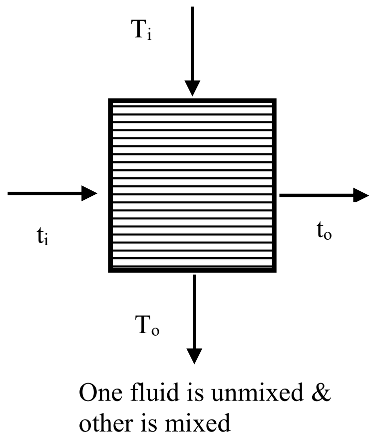


Figure 13.6 Correction factor F when both fluids are unmixed



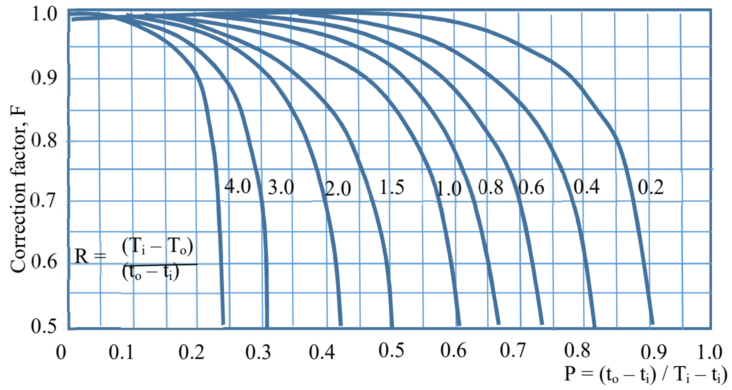


Figure 13.7 Correction factor F when one fluid is unmixed the other is mixed

Example 13.4

In a radiator that is a cross flow water-to-air heat exchanger where both fluids are unmixed, hot water enters the tubes at 80°C at a rate of 0.3 kg/s and leaves at 55°C. The radiator has 50 tubes of internal diameter 0.4 cm and length 50 cm in a closely spaced plate-finned matrix. Air flows across the radiator through the fin spaces and is heated from 25°C to 40°C. Determine the overall heat transfer coefficient U of this radiator based on the inner surface area of the tubes. Take the specific heat capacity of water as 4190 J/kg.K.

Solution

$Q = m \times Cp \times \Delta T = 0.3 \times 4.190 \times (80 – 55) = 31.4$ kW

$$LMTD = \frac{\Delta T_2 - \Delta T_1}{ln\left(\frac{\Delta T_2}{\Delta T_2}\right)}$$

ΔT_2 = (80 – 40) = 40°C

ΔT_1 = (55 – 25) = 30°C

LMTD = (40 – 30) / ln (40/30) = 34.8°C

$LMTD_{cross\ flow}$ = F x $LMTD_{counter\ flow}$

For Figure (13.6),

P = (t_o – t_i) / (T_i – t_i) = 15 / 55 = 0.27

R = (T_i – T_o) / (t_o – t_i) = 25 / 15 = 1.7

F = 0.96

Q = U x A x LMTD x F

Surface area per tube = π x 0.004 x 0.50 = 0.006 m^2

Total surface area (for 50 tubes) = 0.006 x 50 = 0.3 m^2

Therefore, Q = 31.4 = U x 0.3 x 34.8 x 0.96

U = 3140 W/m^2.K

13.5 Extended surfaces

If heat is transferred between two fluids through a wall as shown in Figure 13.8, and heat transfer coefficients at the two surfaces are h_o and h_i, neglecting the resistance of the wall, the overall heat transfer coefficient can be expressed as shown in equation (13.7).

$$\frac{1}{U} = \frac{1}{h_i} + \frac{1}{h_o} \quad (13.7)$$

The overall heat transfer coefficient computed using equation (13.7) will be lower than the lowest of the two heat transfer coefficients.

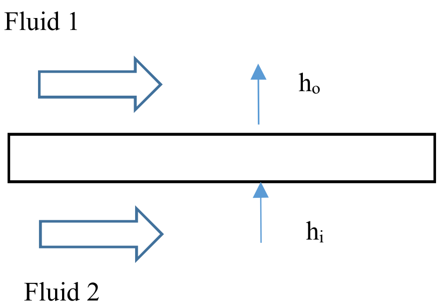


Figure 13.8 Heat exchange between two fluids

For instance, if h_i is 10 W/m^2.K and h_o is 1000 W/m^2.K, the overall heat transfer coefficient will be 9.9 W/m^2.K (which is lower than the lowest value of 10 W/m^2.K).

$$\frac{1}{U} = \frac{1}{10} + \frac{1}{1000}$$

Therefore, U = 9.9 W/m^2.K

Since the lower heat transfer coefficient is the controlling factor, one way of increasing heat transfer is to increase the lower heat transfer coefficient. Since in many cases it is not practical to increase the convective heat transfer coefficient at the heat transfer surface, an alternative is to increase the area available for heat transfer. This can be achieved by adding fins. As shown in Figure 13.9, a fin exposed to the flowing fluid, is heated or cooled by it, thereby increasing the heat conducted from the wall through the fin.

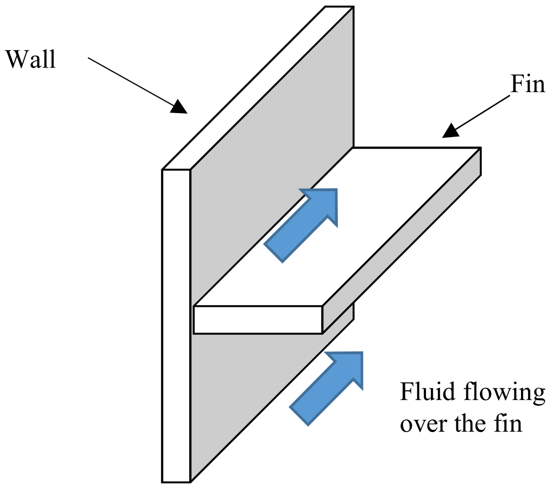


Figure 13.9 Arrangement of a fin

Fin efficiency

Fin efficiency is defined as the ratio of the actual heat transfer rate to the maximum possible heat transfer rate from the fin, if the fin surface is kept at the same temperature as the base temperature.

$$\eta_f = \frac{q_a}{q_{max}} \tag{13.8}$$

where,

η_f = fin efficiency

q_a = actual heat transfer rate

q_{max} = maximum possible heat transfer rate

and, $q_{max} = h \times A \times (T_w - T_\infty)$ (13.9)

where,

h = heat transfer coefficient

A = heat transfer area of the fin

T_w = wall temperature (refer to Figure 13.10)

T_∞ = fluid temperature (refer to Figure 13.10)

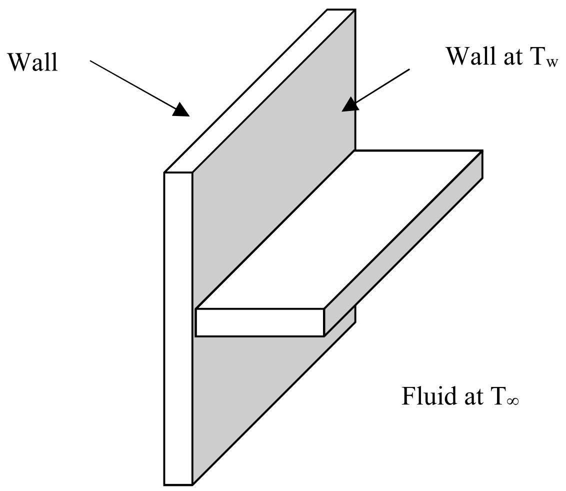


Figure 13.10 Illustration of wall and fluid temperature

The maximum possible heat transfer rate computed using equation (13.9) assumes that the entire fin surface is at the wall temperature (T_w). However, the actual surface temperature of the fin will reduce from T_w to a lower temperature, as indicated in Figure 13.11.

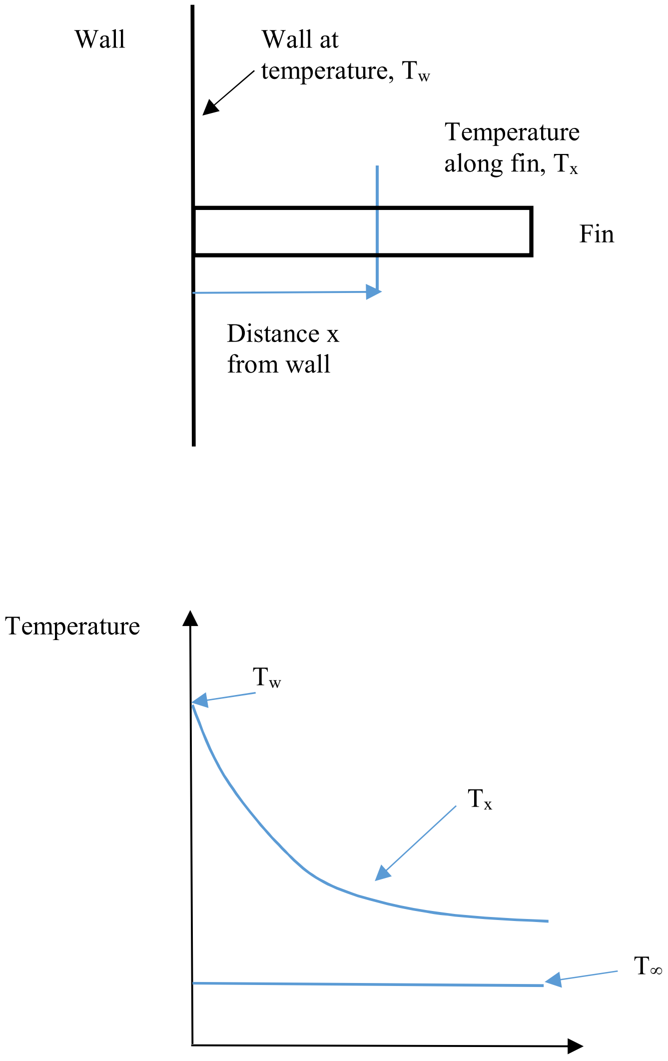


Figure 13.11 Temperature variation along a

Therefore, the actual heat transfer rate (q_a) would be less than the maximum possible heat transfer rate (q_{max}), and the fin efficiency will be less than 1.0.

<u>Fin effectiveness</u>

Fin effectiveness is defined as the ratio of the heat transfer rate with the fin to the heat transfer rate without the fin.

$$\epsilon_f = \frac{q_a}{q_{wof}} \tag{13.10}$$

where,

ε_f = fin effectiveness

q_a = heat transfer rate through the fin

q_{wof} = heat transfer rate without the fin

The heat transfer rate without the fin is the heat transfer rate that would take place from the part of the wall (where the fin is fixed), if the fin was not present (Figure 13.12).

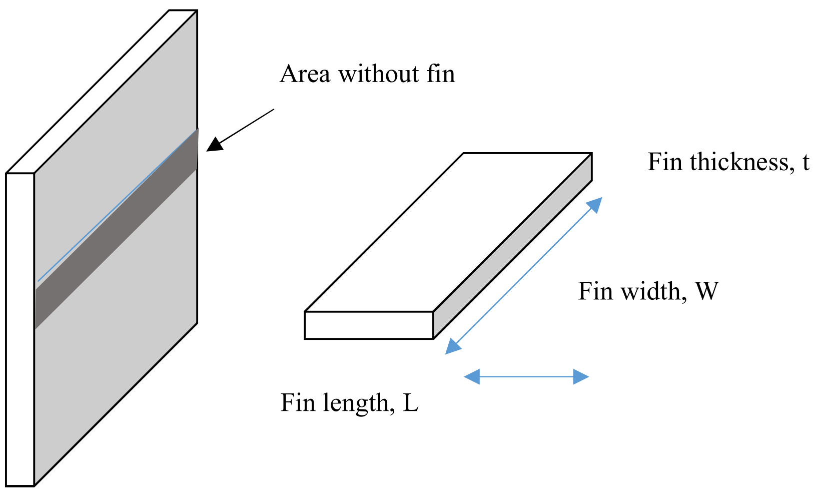


Figure 13.12 Illustration of area without fin

Therefore, q_{wof} can be expressed as:

$$q_{wof} = h \times A \times (T_w - T_\infty) \tag{13.11}$$

where,

h = convective heat transfer coefficient

A = cross-sectional area of the fin

In any rational design, the value of fin effectiveness should be as large as possible. In general, the use of fins may rarely be justified unless it is greater than 2.

Example 13.5

The heat transfer rate from a rectangular fin is estimated to be 50 W. The fin is attached to the wall of a heat exchanger cooled by air. The heat exchanger surface temperature is maintained at 150°C and the ambient air temperature is 30°C. The width and thickness of the fin are 250 mm and 3 mm, respectively. The convective heat transfer coefficient is 10 W/m^2 K and the fin efficiency is 70%.

Compute the length of the fin and the effectiveness of the fin.

Solution

The surface area of the fin, taking the fin length to be L,

= (2 x 0.25 x L) + (0.003 x (2L + 0.25))

= 0.506 L + 0.00075 (the second term can be neglected)

Therefore,

Fin length calculation

$q_{max} = h \times A \times (T_w - T_\infty)$

$= 10 \times 0.506 \times L \times (150 - 30) = 607 L$

$$\eta_f = \frac{q_a}{q_{max}} = 0.7$$

Since q_a = 50 W,

q_{max} = 50 / 0.7 = 71.4 W = 607 x L

Hence, L = 0.118 m (118 mm)

$q_{wof} = h \times A \times (T_w - T_\infty)$

= 10 x 0.003 x 0.05 x (150 – 30)

= 0.18 W

Fin effectiveness calculation,

$$\epsilon_f = \frac{q_a}{q_{wof}} = \frac{50}{0.18} = 277.8$$

13.6 Heat exchanger effectiveness

Similar to fins, the effectiveness of a heat exchanger can also be defined as the ratio of the actual heat transfer rate to the maximum possible heat transfer rate.

$$\epsilon_f = \frac{q_a}{q_{max}} \qquad (13.11)$$

where,

ε_f = fin effectiveness

q_a = heat transfer rate of the heat exchanger

q_{max} = maximum possible heat transfer rate for the heat exchanger

The heat transfer rate, q = m x Cp x ΔT (13.12)

where,

m = mass flow rate of the fluid

C_p = specific heat capacity of the fluid

ΔT = temperature difference of the fluid (between inlet and outlet)

In a heat exchanger where heat is transferred from one fluid to another, the maximum heat transfer rate possible for the heat exchanger is dependent on the characteristics of one of the fluids.

Consider the counterflow heat exchanger shown in Figure 13.3. The typical temperature profiles for the two fluids are shown in Figure 13.13.

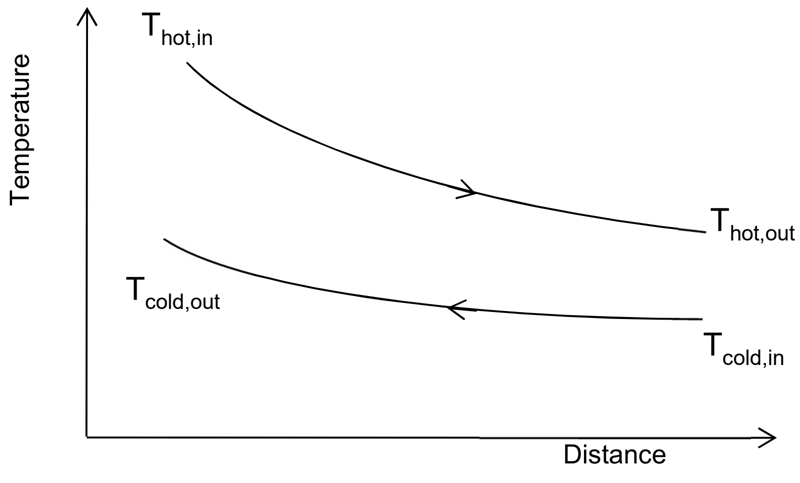


Figure 13.13 Typical fluid temperature profiles for a counterflow heat exchanger

In such a heat exchanger, one of the fluids would experience the maximum possible temperature difference, $T_{hot,in} - T_{hot,out}$ or $T_{cold,in} - T_{cold,out}$.

As shown in Figure 13.14, if the maximum possible heat transfer rate is achieved for the hot fluid (for an infinitely long heat exchanger), then, $T_{hot,out} = T_{cold,in}$.

Similarly, if the maximum heat transfer rate is for the cold fluid, then, $T_{cold,out} = T_{hot,in}$.

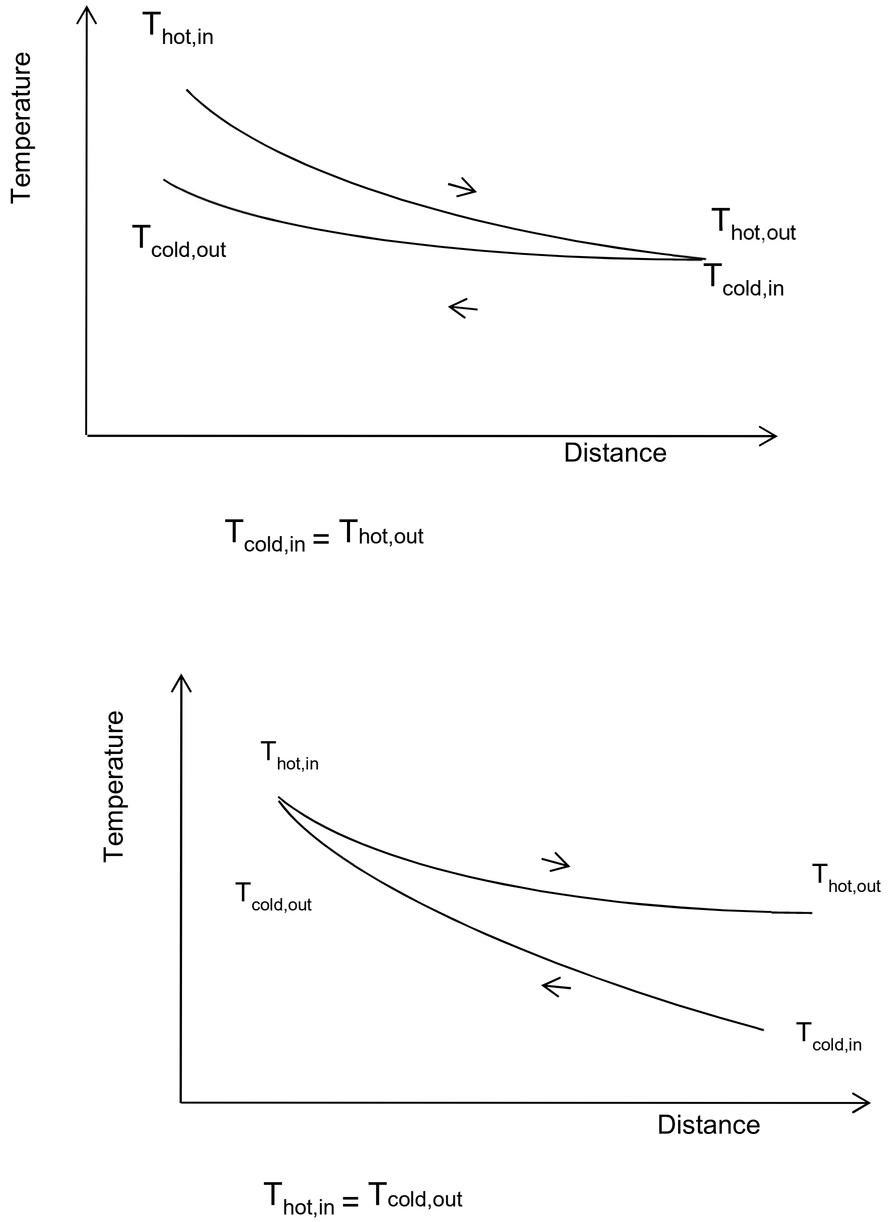


Figure 13.14 Temperature profiles for maximum heat transfer rate

From equation (13.1)

$Q = m_{hot} \times Cp_{hot} \times (T_{hot\ in} - T_{hot\ out}) = m_{cold} \times Cp_{cold} \times (T_{cold\ out} - T_{cold\ in})$

Since the total heat transfer rate for both fluids is equal, the fluid having the maximum temperature difference should have the minimum value for the term m x Cp.

Therefore, the minimum value of m_{hot} x Cp_{hot} or m_{cold} x Cp_{cold} will dictate which fluid can experience the maximum heat transfer rate.

Hence, if m_{hot} x Cp_{hot} is the lower of the two,

$q_{max} = m_{hot} \times Cp_{hot} \ (T_{hot,inlet} - T_{cold,inlet})$

Similarly, if m_{cold} x Cp_{cold} is the lower of the two,

$q_{max} = m_{cold} \times Cp_{cold} \ (T_{hot,inlet} - T_{cold,inlet})$

Therefore,

$$q_{max} = (m \cdot Cp)_{min} \ (T_{hot,inlet} - T_{cold,inlet}) \qquad (13.13)$$

using equation (13.11), the heat exchanger effectiveness can be expressed as,

$$\epsilon = \frac{m_{hot}\ Cp_{hot}\ (T_{hot,inlet} - T_{hot,outlet})}{(m \cdot Cp)_{min}\ (T_{hot,inlet} - T_{cold,inlet})} \qquad (13.14)$$

and

$$\epsilon = \frac{m_{cold}\ Cp_{cold}\ (T_{cold,inlet} - T_{cold,outlet})}{(m \cdot Cp)_{min}\ (T_{hot,inlet} - T_{cold,inlet})} \qquad (13.15)$$

By definition, the effectiveness will be between 0 and 1 ($0 \leq \varepsilon \leq 1$).

The concept of effectiveness is useful because if the hot and cold fluid inlet temperatures are known (which is normally the case), the heat exchanger effectiveness and actual heat transfer rate can be computed as illustrated in the following example.

Example 13.6

In a finned type cross-flow heat exchanger, hot exhaust gases are used to heat water. The inlet and outlet temperatures of the exhaust gases are 250°C and 100°C while the inlet and outlet temperatures of water are 35°C and 130°C. The flow rate of water is 1.5 kg/s. The specific heat capacity of the exhaust gases is 1.1 kJ/kg.K. The gas side overall heat transfer coefficient is 125 W/m^2. Compute the effectiveness of the heat exchanger.

Solution

Since the specific heat capacity of water is known (4.19 kJ/kg.K), using equation (13.1), the mass flow rate of the hot exhaust gases can be computed as follows.

$Q = m_{hot} \times Cp_{hot} \times (T_{hot\ in} - T_{hot\ out}) = m_{cold} \times Cp_{cold} \times (T_{cold\ out} - T_{cold\ in})$

$m_{hot} \times 1.1 \times (250 - 100) = 1.5 \times 4.19 \times (130 - 35)$

$m_{hot} = 3.619$ kg/s

$(m \times Cp)_{hot} = 3.619 \times 1.1 = 3.981$ kW/K

$(m \times Cp)_{cold} = 1.5 \times 4.19 = 6.285$ kW/K

$(m \times Cp)_{min}$ is for the hot gases.

Therefore, from equation (13.13),

$q_{max} = (m \ . \ Cp)_{min} \ (T_{hot,inlet} - T_{cold,inlet})$

$= 3.981 \times (250 - 35) = 855.8$ kW

The actual heat transfer rate is (based on water),

= 1.5 kg/s x 4.19 kJ/kg.K x (130 – 35) K

= 597 kW

From equation (13.11),

$$\epsilon = \frac{q_a}{q_{max}} = \frac{597}{855.8} = 0.7$$

Summary

This chapter described the main types of heat exchangers such as parallel-flow, counterflow and cross flow. Their characteristics and performance evaluation methods were explained and illustrated using examples.

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1. Arora, C P, Heat and Mass Transfer, Khanna Publishers, Delhi, 1986.
2. Bergman, Lavine, Incropera, and DeWitt, Fundamentals of Heat Transfer, 7th edition, John Wiley & Sons, New York, 2011.
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14.0 COMMERCIALLY AVAILABLE HEAT RECOVERY DEVICES

As explained in the earlier chapters of the reference manual, heat recovery devices are required for heat recovery applications. There are many types of standard heat recovery devices available in the market to suit various needs. This chapter will describe the main types of commercially available heat recovery devices, their applications and selection for specific requirements.

Learning Outcomes:

The main learning outcomes from this chapter are:

1. To understand the main types of commercially available heat recovery devices
2. How to select heat recovery devices for specific applications

14.1 Shell and tube heat exchangers

The double pipe type parallel flow and counter flow heat exchangers explained in the earlier chapter (Figures 13.1 and 13.3) are of "single pass" design. Therefore, they are not suitable for many applications with high heat transfer requirements as they would need a large space. An alternative type of heat exchanger used in power plants and industrial facilities is the shell and tube heat exchanger.

Shell and tube heat exchangers consist of a bundle of tubes inside a large shell. One of the fluids to be heated or cooled flows through the tubes while the other fluid which either provides heat or absorbs heat flows over the tubes inside the shell. The fluids exchanging heat can be in liquid form where only sensible heat transfer takes place or can change phase for latent heat transfer as is the case in power plants where steam is condensed to liquid in the shell of the heat exchanger.

Shell and tube heat exchangers can be classified according to the number of "passes" the fluid flows through the shell or tube arrangement. Figures 14.1 to 14.3 show the arrangement of single and two pass heat exchangers.

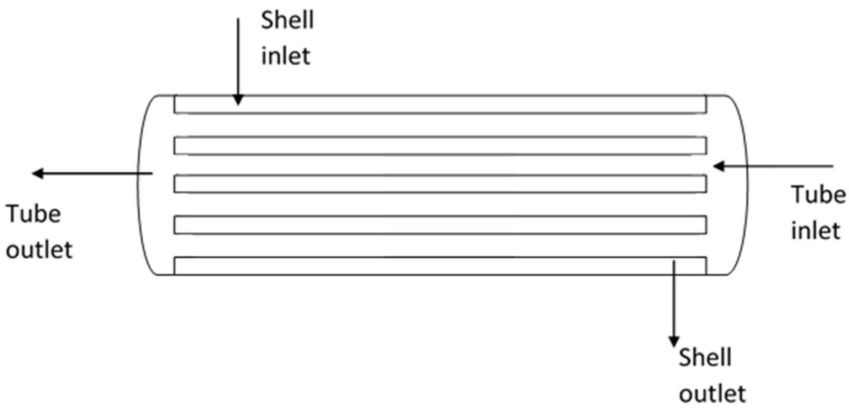


Figure 14.1 One shell pass and one tube pass

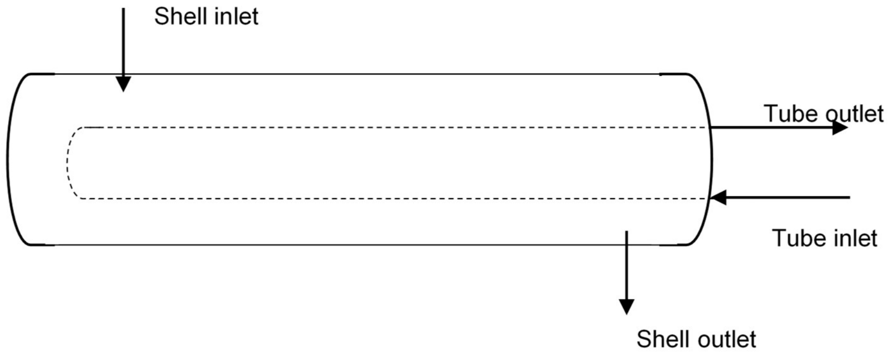


Figure 14.2 One shell pass and two tube passes

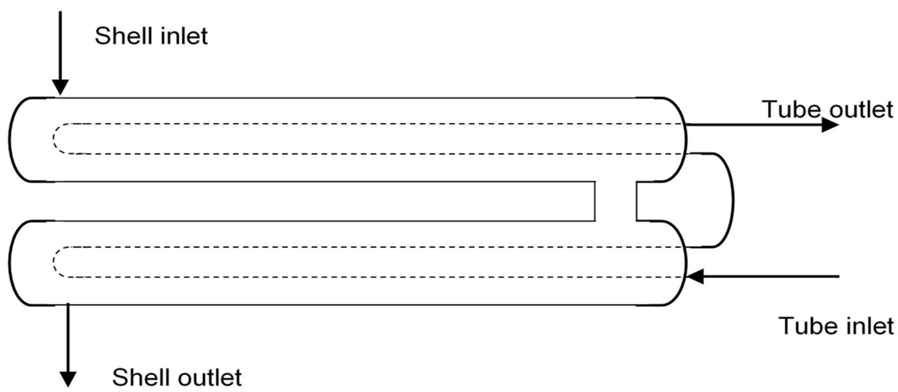


Figure 14.3 Two shell passes and four tube passes

The heat transfer rate can be computed for shell and tube heat exchangers using equation (14.1).

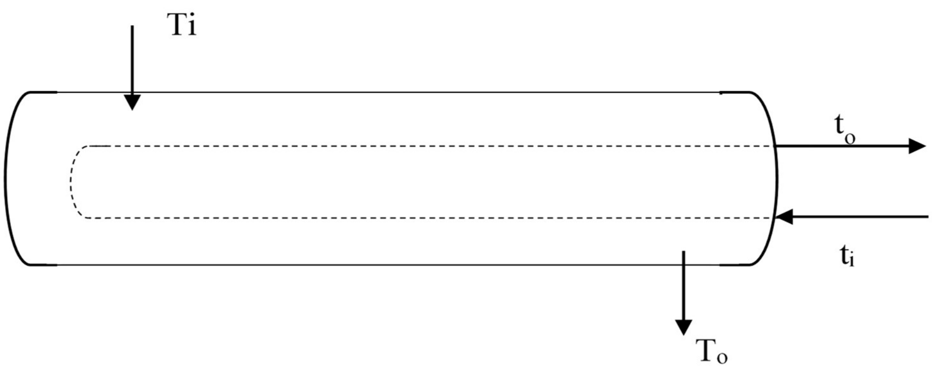
$$Q = U \times A \times \Delta T_m \qquad (14.1)$$

where, F is a correction factor dependent on the number of shell and tube passes

$$\Delta T_m = F \times LMTD$$

$$LMTD = \frac{\Delta T_2 - \Delta T_1}{ln\ \frac{\Delta T_2}{\Delta T_2})}$$

Typical correction factors, "F" are shown in Figures 14.4 and 14.5.



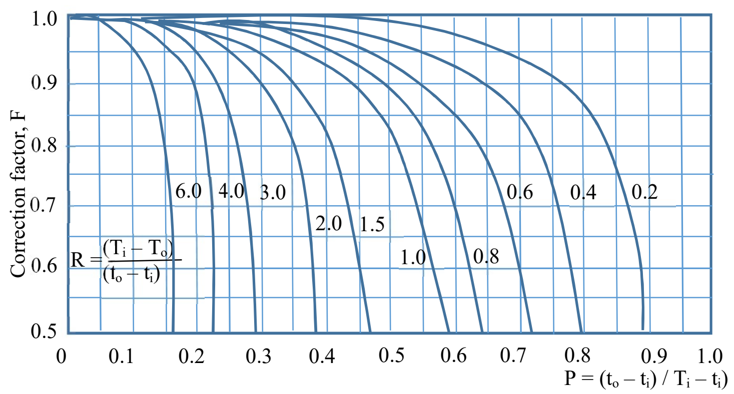


Figure 14.4 Correction factor F for one shell pass and multiple tube passes (multiples of 2)

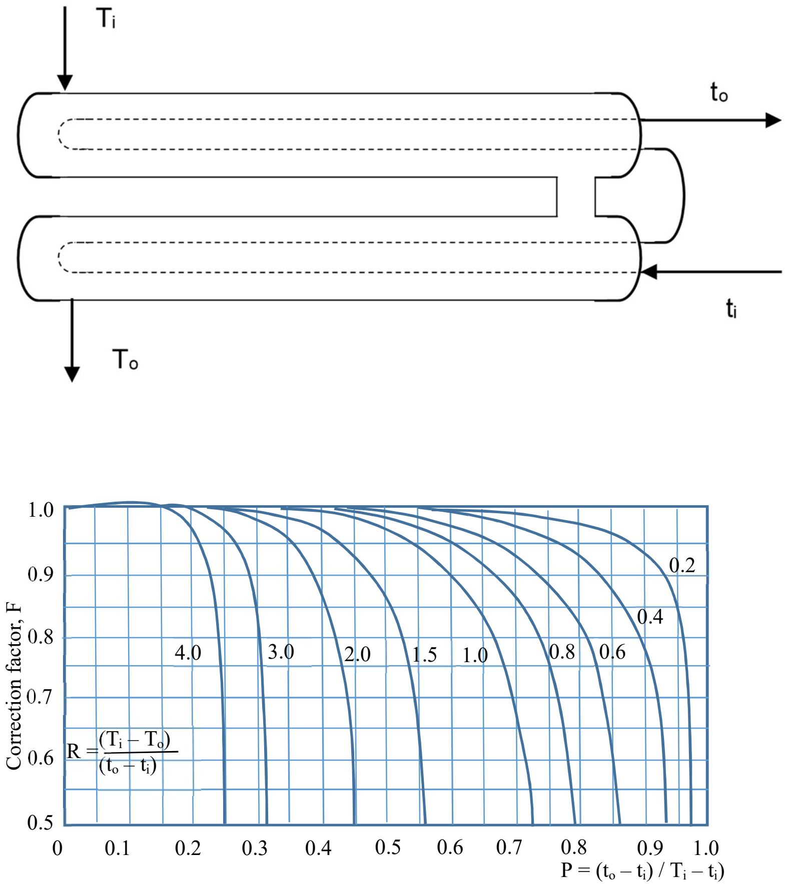


Figure 14.5 Correction factor F for two-shell pass and multiple tube passes (multiples of 4)

Example 14.1

A 2-shell pass and 4-tube pass heat exchanger is used to heat oil from 30°C to 50°C using hot water which flows through 25 mm diameter tubes. The hot water enters the heat exchanger at 70°C and leaves at 40°C. The total length of the tubes in the heat exchanger is 40 m. The convection heat transfer coefficient is 50 W/m^2.K on the oil (shell) side and 200 W/m^2.K on the water (tube) side. Determine the rate of heat transfer in the heat exchanger if the tube thickness is 0.5 mm and the tube material has a thermal conductivity of 400 W/m.K.

Solution

$$LMTD = \frac{\Delta T_2 - \Delta T_1}{ln\ \frac{\Delta T_2}{\Delta T_2})}$$

ΔT_2 = (70 – 50) = 20°C

$\Delta T_1 = (40 - 30) = 10^oC$

$LMTD = (20 - 10) / \ln (20/10) = 14.4^oC$

$LMTD_{shell\ and\ tube} = F \times LMTD_{counter\ flow}$

From Figure 14.5,
$P = (t_o - t_i) / (T_i - t_i) = 30 / 40 = 0.75$

$R = (T_i - T_o) / (t_o - t_i) = 20 / 30 = 0.67$

$F = 0.85$

Tube surface area = $\pi \times 0.025 \times 40 = 3.14\ m^2$

$U = \left[\frac{1}{\frac{1}{h_A}+\frac{x_1}{k_1}+\frac{1}{h_B}}\right]$ (from equation (12.7)

$1/U = (1/50) + (0.0005/400) + (1/200)$

$U = 40\ W/m^2.K$

$Q = U \times A \times LMTD \times F$

$Q = 40 \times 3.14 \times 14.4 \times 0.85 = 1537\ W$

14.2 Plate type heat exchangers

Plate heat exchangers consist of a number of plates which act as the heat transfer surfaces, stacked together on a frame. The arrangement of a typical plate heat exchanger is shown in Figure 14.6. The path for the flow of the two fluids transferring heat is arranged so that the two fluids flow either side of the plates. The plates can be made of different materials depending on the application.

Plate heat exchangers can be gasket type where gaskets are placed between plates to provide a seal. In such heat exchangers, plates can also be easily cleaned by loosening the frame arrangement and moving the plates. This type of heat exchanger is commonly used for HVAC applications and in the food manufacturing industry.

Some plate heat exchangers have welded plates and therefore, cannot be cleaned. Such heat exchangers are used for high temperature and hazardous fluids.

Another type of plate heat exchanger is the semi-welded type where pairs of plates are welded and are sandwiched using gaskets to other pairs of plates. In such heat exchangers, the corrosive fluid flows through the welded plates while the other non-hazardous fluid, which may require regular cleaning of plates to flow through the gasketed plates.

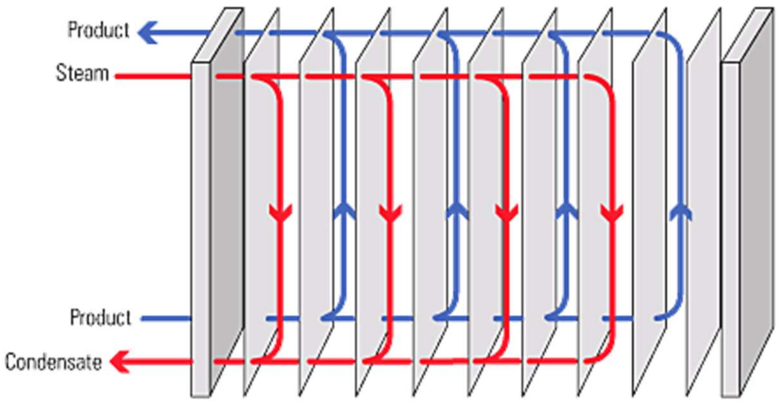


Figure 14.6 Arrangement of a plate heat exchanger (illustration taken from the Spirax Sarco website 'Steam Engineering Tutorials' at http://www.spiraxsarco.com)

The overall heat load of a plate heat exchanger can be computed using the following relationship.

Q = U x A x LMTD (14.2)

where,
U = overall heat transfer coefficient (W/m^2.K)
A = heat transfer area (m^2)
LMTD = log mean temperature difference

$$LMTD = \frac{\Delta T_2 - \Delta T_1}{ln\ \frac{\Delta T_2}{\Delta T_2})} \qquad (14.3)$$

where (based on Figure 14.7),

$\Delta T_2 = T_1 - T_4$

$\Delta T_1 = T_2 - T_3$

T_1 = Temperature inlet – hot side
T_2 = Temperature outlet – hot side
T_3 = Temperature inlet – cold side
T_4 = Temperature outlet – cold side

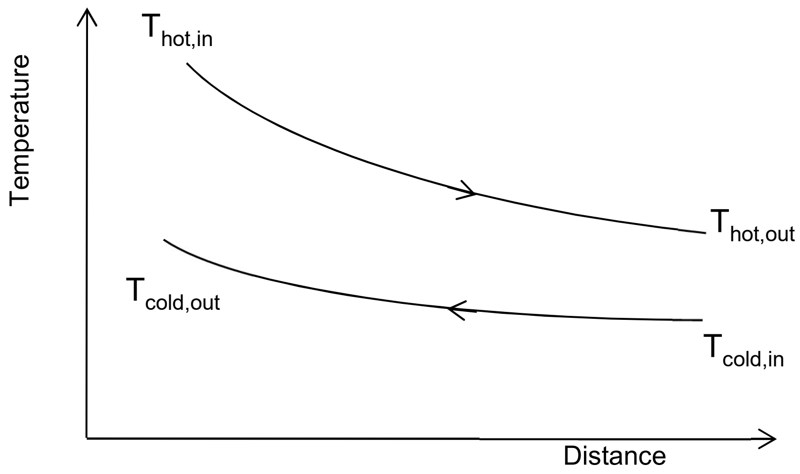


Figure 14.7 Heat exchanger fluid temperatures

The overall heat transfer coefficient U can be expressed as,

$$\frac{1}{U} = \frac{1}{h_c} + \frac{1}{h_h} + \frac{x}{k} \tag{14.4}$$

where,
h_c = heat transfer coefficient on cold side (W/m^2.K)
h_h = heat transfer coefficient on hot side (W/m^2.K)
x = thickness of plate (m)
k = thermal conductivity of plate (W/m.K)

Since the heat transfer plates are relatively thin (0.3 to 0.6 mm) and fluid flow is turbulent, the overall heat transfer coefficient, U can be many times that of a shell and tube heat exchanger.

Example 14.2

A plate heat exchanger is used to heat oil from 35°C to 75°C. The oil stream is heated using water entering the heat exchanger at 90°C and leaving at 60°C. The heat exchanger plate thickness is 0.4 mm and the plates are made with material having thermal conductivity of 350 W/m.K. The heat transfer coefficients on the hot side and cold side are 3000 and 2800 W/m$^{2.}$K, respectively. Compute the total heat transferred if the total heat transfer surface area of the heat exchanger is 10m^2.

Solution

Using equation (14.4)

$$\frac{1}{U} = \frac{1}{2800} + \frac{1}{3000} + \frac{0.0004}{350} = 0.00069$$

Therefore, U = 1446 W/m^2.K

From Figure 14.8,

ΔT_2 = 90 – 75 = 15°C

ΔT_1 = 60 – 35 = 25°C

Using equation (14.3),

$$LMTD = \frac{\Delta T_2 - \Delta T_1}{ln \frac{\Delta T_2}{\Delta T_2})} = \frac{(15-2\ \ )}{ln \frac{15}{25})} = \frac{-10}{-0.51} = 19.6$$

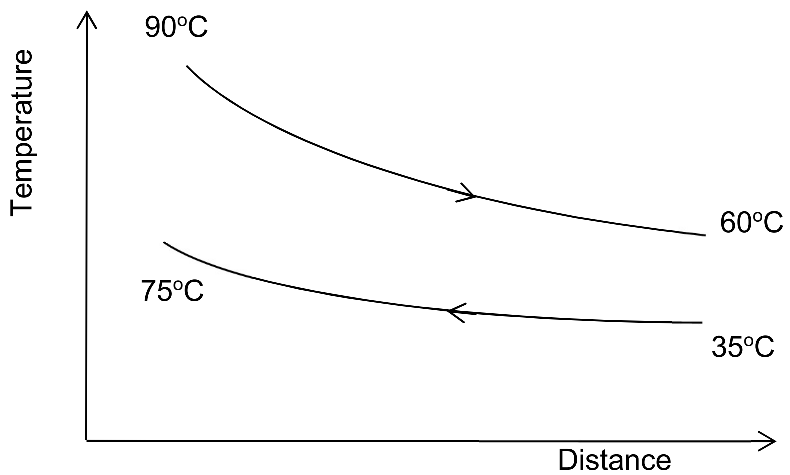


Figure 14.8 Fluid temperatures for example 14.2

Using equation (14.1),

Q = U x A x LMTD

Q = 1446 x 10 x 19.6 = 283,416 W = 283.4 kW

14.3 Rotary heat exchangers

This type of heat exchanger consists of a rotating wheel that absorbs heat from a hot fluid and then transfers it to the colder fluid (Figure 14.9). The working arrangement of a rotary heat exchanger is shown in Figure 14.10. This type of heat exchanger can be used for relatively low temperature applications in HVAC (heating, ventilating and air conditioning) systems as well as in high temperature applications (up to 500^{o}C) to recover heat from the exhaust of furnaces, engines and dryers.

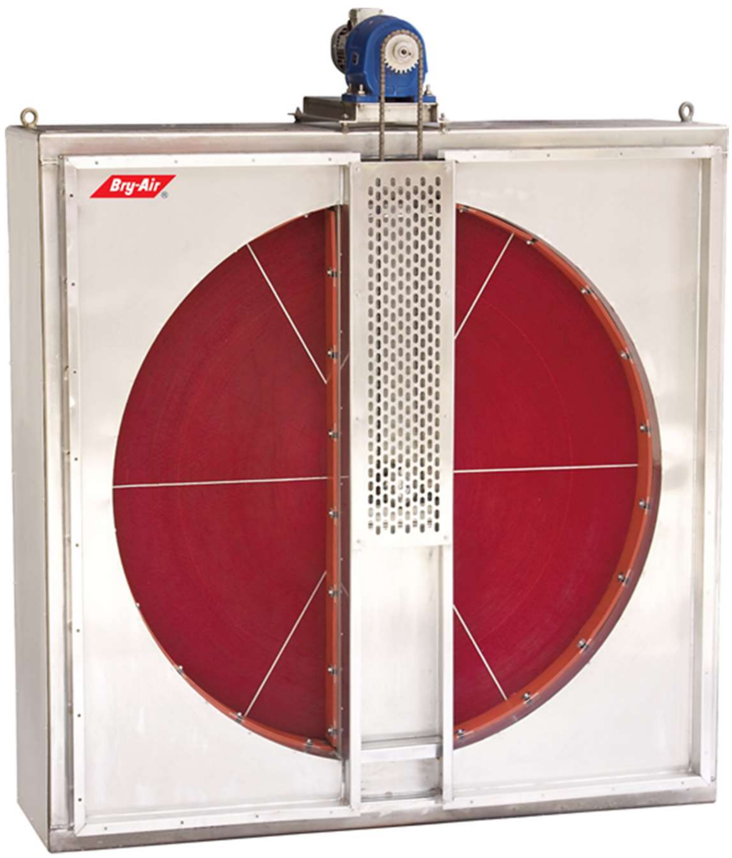


Figure 14.9 Rotary heat exchanger (courtesy of Bry-Air)

The amount of energy recovered depends on the efficiency of the recovery system and can be expressed as follows for systems that exchange only sensible heat.

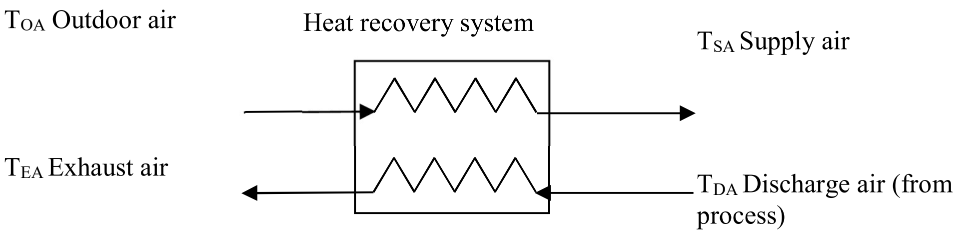


Figure 14.10 Heat exchange in a rotary heat exchanger

Sensible heat transfer,

$$\eta_{Sensible} = \left(\frac{T_{SA} - T_{OA}}{T_{DA} - T_{OA}} \right) x\,100 \tag{14.5}$$

where,

η = efficiency (%)

T = dry-bulb temperature (°C)

OA = outdoor air

SA = supply air

DA = discharge air (exhaust from industrial process)

The amount of energy recovered can be expressed as follows,

$$Q_{sensible} = \rho \text{ x } v \text{ x } Cp \text{ x } (T_{SA} - T_{OA}) \tag{14.6}$$

where,

Q = energy recovered (kW)

ρ = density of air (kg/m^3)

v = air flow rate (m^3/s)

Cp = specific heat capacity of air (kJ/kg.K)

The heat transfer efficiency depends on the individual design and the flow velocity, but typical values for sensible heat exchange is about 75%.

Example 14.3

Consider the case where 1.2 m^3/s of outdoor air is provided to an industrial dryer to make up for an equal amount of air removed by the exhaust system. If the outdoor air temperature is 32°C and the dryer discharge air temperature is 300°C, find the supply air temperature that can be achieved using an energy recovery system that can only transfer sensible heat and has an efficiency of 75%. Also, calculate the total amount of pre-heating done to the outdoor air. Take the density of air to be 0.9 kg/m^3 and the specific heat capacity of air as 1002 J/kg.K.

Solution

Using equation (14.5),

$$T_{SA} = T_{OA} + (T_{DA} - T_{OA}) \times \eta_{Sensible}$$

$$= 32 + (300 - 32) \times 0.75$$

$$= 233^{o}C$$

Sensible heat added,

$$Q_{sensible} = \rho \times v \times Cp \times (T_{SA} - T_{OA})$$

$$= 0.9 \times 1.2 \times 1002 \times (233 - 32)$$

$$= 217.5 \text{ kW}$$

14.4 Heat pipes

A heat pipe is a heat exchange device which transfers heat through the phase change of a substance. The typical arrangement of a heat pipe is shown in Figure 14.11. The heat pipe consists of a hot side and a cold side. The liquid at the hot end absorbs heat from the outer surrounding area through the thermally conductive tube wall. The liquid then vapourises and the vapour travels to the cold side of the tube. When the vapour reaches the cold side, it is cooled by the tube wall, which is exposed to the cold medium, and the vapour condenses. The condensed vapour then flows back to the hot side by gravity or through capillary action to continue the cycle.

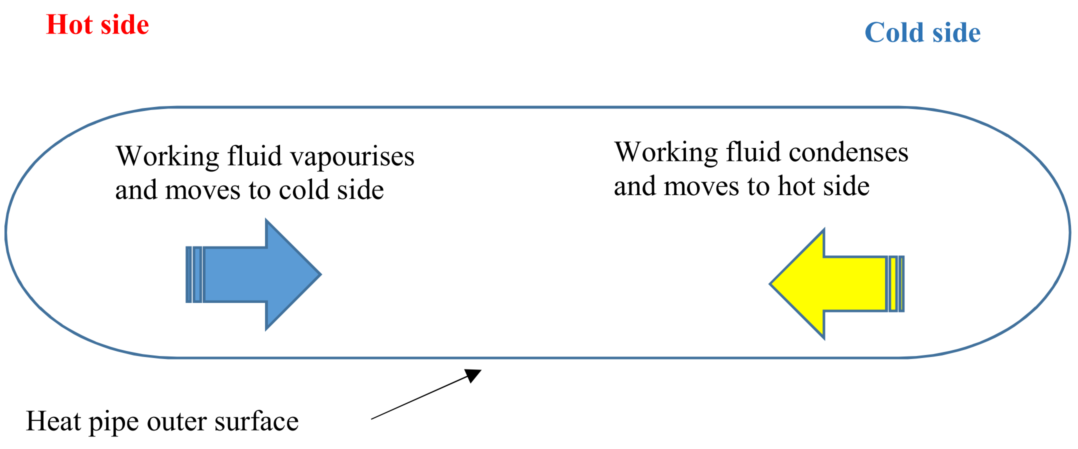


Figure 14.11 Working arrangement of a heat pipe

The heat pipe is made using a sealed metal tube. Air is removed from the tube and a vacuum is created. The tube is then filled with the working fluid. Various working fluids such as water, ammonia and refrigerants are used in heat pipes. The heat pipe tube material is selected to suit the working fluid. Copper is used with water while aluminium is used for ammonia. Heat pipes can be used for various heat recovery applications at different temperatures by selecting appropriate tube materials and working fluids. Copper tubes with water are typically used for temperature ranging from 20 to 300°C.

14.5 Direct contact heat exchangers

A direct contact heat exchanger is a device used to transfer heat between two or more streams without the presence of a separating wall. The mass streams can be parallel-flow, counterflow or cross flow. Each stream can be solid, liquid or gas. The combination of the two streams can be gas-solid, gas-liquid, liquid-liquid, liquid-solid, or solid-solid.

If the two streams are gas-liquid or liquid-liquid, they will mix, unless the streams are immiscible. Hence, stream contamination will occur depending on the degree of miscibility. In direct contact heat exchangers, the two streams also have to be at the same pressure.

One of the main advantages of direct type heat exchangers is the better heat transfer rate for a given size of a heat exchanger. This is primarily because of the effectively larger heat transfer surface area and the ability to transfer heat at much lower temperature differences between the two streams.

Other advantages of direct type heat exchangers include lower pressure drop due to the absence of tubes or plates and lower cost as they can be constructed using vessels with inlet and discharge ports for the two streams.

A common type of direct contact heat exchanger is where a perforated steam coil is submerged in a water tank as shown in Figure 14.12. The steam discharged through the holes in the coil heats the water in the tank.

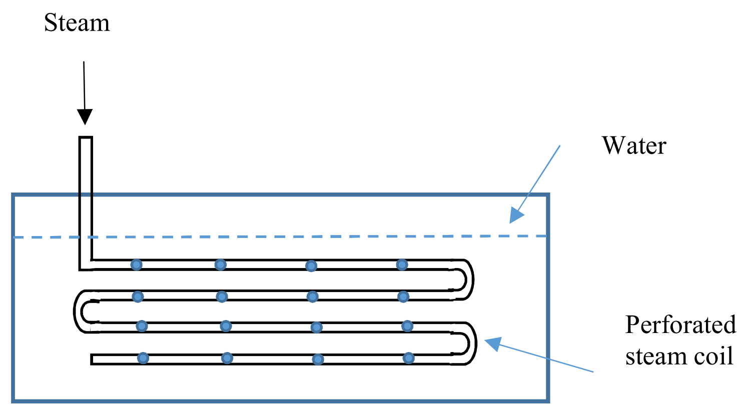


Figure 14.12 Direct contact type heat exchanger used for heating water

There are many other designs of direct contact heat exchangers. Two common arrangements for a turbulent pipe contactor and a mechanically agitated tower are illustrated in Figures 14.13 and 14.14, respectively.

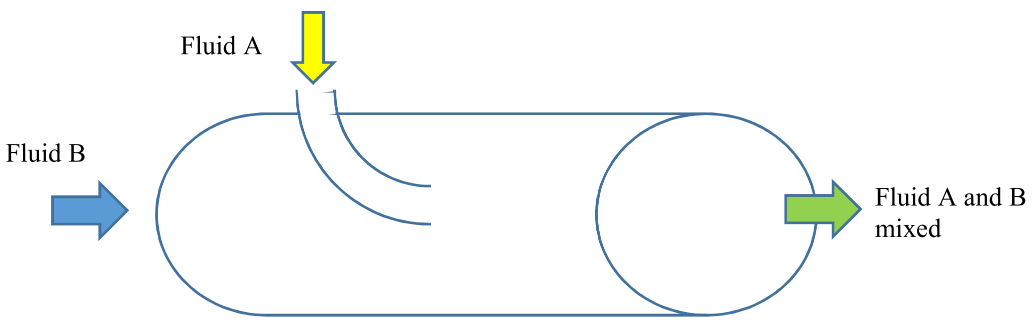


Figure 14.13 Arrangement of a turbulent pipe contactor

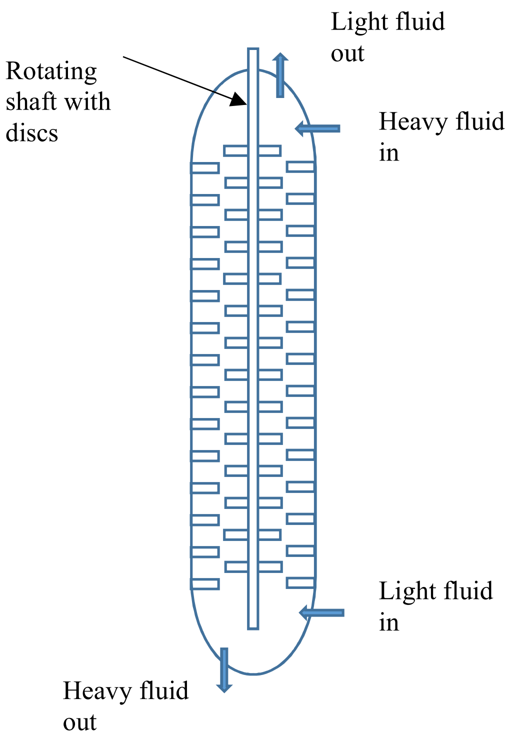


Figure 14.14 Arrangement of a mechanically agitated tower

14.6 Fouling in heat exchangers

Heat exchanger surfaces are often subjected to fouling and scaling due to the depositing of fluid impurities, rust formation or chemical reactions between the fluid and wall material. Fouling or scaling on the heat transfer surface creates an additional layer on the surface resulting in a greater resistance to heat transfer.

Fouling depends on factors such as the type of fluids used, materials used for construction of the heat exchanger and length of time the heat exchanger has been in operation. Often, heat exchangers need to be cleaned periodically to minimise the resistance to heat transfer resulting from fouling.

Therefore, the overall heat transfer coefficient value (U) given in equation (12.7) should be adjusted to include the effect of fouling as shown in equation (14.7) to ensure that the heat exchanger selected for a particular application is able to provide the required capacity.

$$U = \left[\frac{1}{\frac{1}{h_A}+\frac{x_1}{k_1}+\frac{x_2}{k_2}+\frac{1}{h_B}+R_f}\right] \tag{14.7}$$

where, R_f is the fouling factor (zero for clean surfaces)

Some typical fouling factors used when selecting and designing heat exchangers is provided in Table 14.1.

Fluid	Fouling factor R_f ($m^2.K/W$)
Treated water from cooling tower	0.0003
River water	0.0004
Sea water	0.0002
Fuel oil	0.0009
Steam	0.0001
Refrigerating liquids	0.0002
Treated boiler feedwater	0.0002
Boiler blowdown	0.00035
Hydraulic and heat transfer oil	0.00018
Cooling fluids	0.00018

Table 14.1 Typical fouling factors

Example 14.4

Calculate the new heat transfer rate for Example 14.1 taking the fouling factor to be 0.0002 $m^2.K/W$.

Solution

$$U = \left[\frac{1}{\frac{1}{h_A}+\frac{x_1}{k_1}+\frac{1}{h_B}+R_f}\right] \quad \text{(from equation (14.7))}$$

1/U = (1/50) + (0.0005/400) + (1/200) + 0.0004

U = 39.4 $W/m^2.K$

Q = U x A x LMTD x F

Q = 39.4 x 3.14 x 14.4 x 0.85 = 1514 W

14.7 Overall heat transfer coefficient values

Computation of the overall heat transfer coefficient for a particular heat exchanger requires the thermal conductivity value of the heat exchanger material, its thickness, and the heat transfer coefficients at the inner and outer surfaces where heat transfer takes place. Although the first two parameters can be obtained from the design specifications of a heat exchanger, computation of the surface heat transfer coefficient values require in-depth heat transfer analysis as they depend on properties of the fluid, flow velocity and surface conditions.

Therefore, to facilitate ease of computation, typical values for the overall heat transfer coefficient are provided in many reference sources. Some typical values are listed in Table 14.2.

Type of heat exchanger	**Fluids**	**Overall heat transfer coefficient, U ($W/m^2.K$)**
Shell and tube	Water to water	800 - 1500
	Light oil to light oil	100 - 400
	Heavy oil to heavy oil	50 - 300
	Solvent to solvent	100 - 300
Plate heat exchanger	Liquid to liquid	5000 - 7500
	Steam to water	3500 - 4500
Double pipe	Liquid to liquid	150 - 1200
	Liquid inside, steam outside	300 - 1200

Table 14.2 Typical U values

As can be seen from Table 14.2, the U values depend on the respective fluids and the type of heat exchanger. In general, for the three types of heat exchanger compared in the table, double pipe heat exchangers have the lowest U values while plate heat exchangers have the highest U values.

Summary

This chapter described the main types of commercially available heat recovery devices such as shell and tube heat exchangers, plate type heat exchangers, rotary heat exchangers, heat pipes and direct contact type heat exchangers. The application of the various types of heat recovery devices was illustrated using examples.

References

1. Arora, C P, Heat and Mass Transfer, Khanna Publishers, Delhi, 1986.
2. Bergman, Lavine, Incropera, and DeWitt, Fundamentals of Heat Transfer, 7th edition, John Wiley & Sons, New York, 2011.
3. Cornwell, K, The Flow of Heat, ELBS and Van Nostrand Reinhold Company Ltd, London, 1977.
4. Jayamaha, Lal, Energy-Efficient Industrial Systems, Evaluation and Implementation, McGraw-Hill, New York, 2016.

15.0 ENERGY AUDIT OF HEAT RECOVERY SYSTEMS

The main objectives of a heat recovery system audit are to:

- Assess system performance
- Establish the energy baseline
- Identify opportunities for improvement

The audit should cover not only the thermal performance, it should also include the hydraulic performance (pressure losses).

Learning Outcomes:

The main learning outcomes from this chapter are to understand:

1. The parameters to be measured in a heat recovery system audit
2. How to compute key performance indicators
3. How to identify potential improvement measures

15.1 Measurements

Schematic diagram of a typical heat recovery system with the location of sensors used for measurements is shown in Figure 15.1.

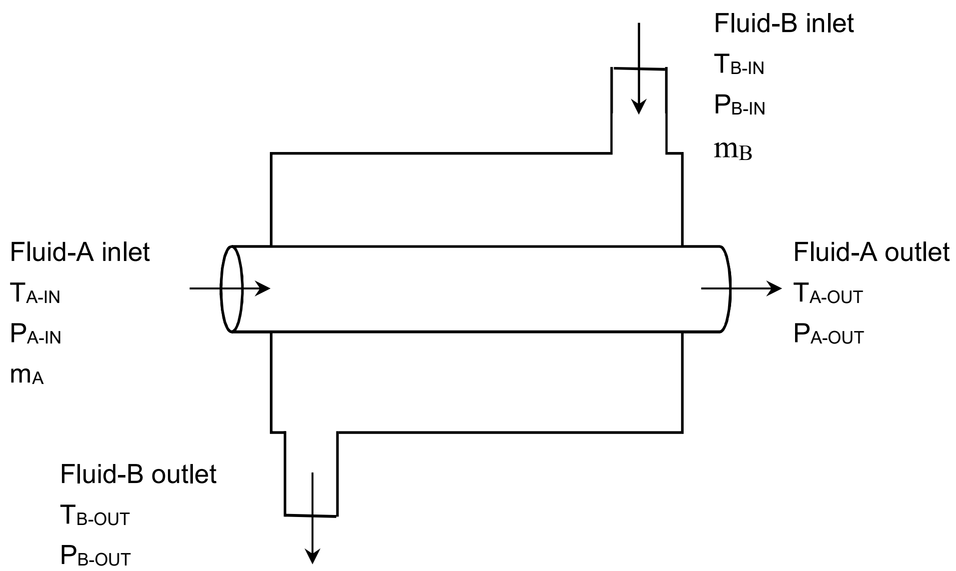


Figure 15.1 Measurement locations for a heat recovery system

The different parameters:

m_A = mass flow rate of fluid A

m_B = mass flow rate of fluid B

P_{A-In} = inlet pressure of fluid A

P_{A-out} = outlet pressure of fluid A

P_{B-In} = inlet pressure of fluid B

P_{B-out} = outlet pressure of fluid B

T_{A-In} = inlet temperature of fluid A

T_{A-out} = outlet temperature of fluid A

T_{B-In} = inlet temperature of fluid B

T_{B-out} = outlet temperature of fluid B

15.2 Calculations

Heat transfer rate

The heat transfer rate Q, can be calculated as follows.

For the hot fluid

$$Q = m_A \times Cp_A \times (T_{A-IN} - T_{A-OUT}) \qquad (15.1)$$

For the cold fluid

$$Q = m_B \times Cp_B \times (T_{B-OUT} - T_{B-IN}) \qquad (15.2)$$

where,

Cp_A = specific heat capacity of fluid A

Cp_B = specific heat capacity of fluid B

Log mean temperature difference (LMTD)

The log mean temperature difference can be calculated as follows.

$$LMTD = \frac{\Delta T_2 - \Delta T_1}{ln\left(\frac{\Delta T_2}{\Delta T_2}\right)} \qquad (15.3)$$

where

ΔT_2 = temperature difference between the fluids at one end of the heat exchanger (eg. $T_{A-IN} - T_{B-OUT}$)

ΔT_1 = temperature difference between the fluids at the other end of the heat exchanger (eg. $T_{B\text{-}IN} - T_{A\text{-}OUT}$)

Overall heat transfer coefficient

The overall heat transfer coefficient U can be calculated as follows.

$$U = Q / (A \times LMTD) \tag{15.4}$$

where

A = heat transfer area of the heat recovery unit

Heat exchanger effectiveness

The heat exchanger effectiveness ε, can be can be computed as follows.

$$\epsilon = \frac{m_{hot}\ Cp_{hot}\ (T_{hot,inlet} - T_{hot,outlet})}{(m\ .\ Cp)_{min}\ (T_{hot,inlet} - T_{cold,inlet})} \tag{15.5}$$

or

$$\epsilon = \frac{m_{cold}\ Cp_{cold}\ (T_{cold,inlet} - T_{cold,outlet})}{(m\ .\ Cp)_{min}\ (T_{hot,inlet} - T_{cold,inlet})} \tag{15.6}$$

where

m_{hot} and m_{cold} = mass flow rates of the hot and cold fluids, respectively

Cp_{hot} and Cp_{cold} = specific heat capacities of the hot and cold fluids, respectively

T = temperature of the cold and hot fluid at the inlet and outlet of the heat recovery unit

m.Cp = the lowest value of the product of the mass flow rate and specific heat capacity of the two fluids

Heat exchanger approach temperature

This term refers to the temperature difference between the leaving process fluid and the entering service fluid. If air is cooled from 60°C to 40°C using 30°C cooling water, the air temperature approaches the water by 10°C (40 – 30 = 10).

As the approach temperature diminishes, the size (and cost) of the heat exchanger will eventually increase exponentially.

Fouling of heat exchangers

The following ratio can be computed to assess the need for cleaning of the heat exchanger due to internal scaling and fouling.

$$\frac{(T_{cold,out} - T_{cold,in})}{(T_{hot,in} - T_{cold,in})} \qquad (15.7)$$

where

$T_{cold\text{-}In}$ = inlet temperature of cold fluid

$T_{cold\text{-}out}$ = outlet temperature of cold fluid

$T_{hot\text{-}In}$ = inlet temperature of fluid being cooled

As the heat exchanger fouls, the cooling fluid temperature in and out will become closer as less heat is transferred from the heat exchanger to the fluid. If the flow rates remain the same, then the denominator of the ratio will remain the same. Thus, a reducing number divided by a static number gives a lower value for the ratio.

Pressure losses

The pressure losses across the heat recovery unit can be computed for both fluid streams using the following relationships.

$$\Delta P_A = P_{A\text{-}IN} - P_{A\text{-}Out} \qquad (15.8)$$

and

$$\Delta P_B = P_{B\text{-}IN} - P_{B\text{-}Out} \qquad (15.9)$$

where

$P_{A\text{-}In}$ = inlet pressure of fluid A

$P_{A\text{-}out}$ = outlet pressure of fluid A

$P_{B\text{-}In}$ = inlet pressure of fluid B

$P_{B\text{-}out}$ = outlet pressure of fluid B

Surface heat loss

Heat can be lost to the environment from the surface of the heat recovery system. For heating applications, this can result in energy wastage. The surface temperature of the heat recovery unit can be estimated using an infrared temperature sensor. The measurements can be used to assess whether it is necessary to improve the insulation.

15.3 Case study of a boiler heat recovery energy audit

Information on the system

In a manufacturing plant, one steam boilers is usually in operation. Diesel is used as the primary fuel for the boiler. Part of the condensate is recovered and to return back to the feedwater tank. The feedwater temperature is usually maintained at around 60°C. The flue gas temperature is about 200°C.

Data collection and analysis

An audit was carried out to study the potential for recovering heat from the boiler flue gas to raise the temperature of feedwater.

The boiler feedwater flow rate is shown in Figure 15.2.

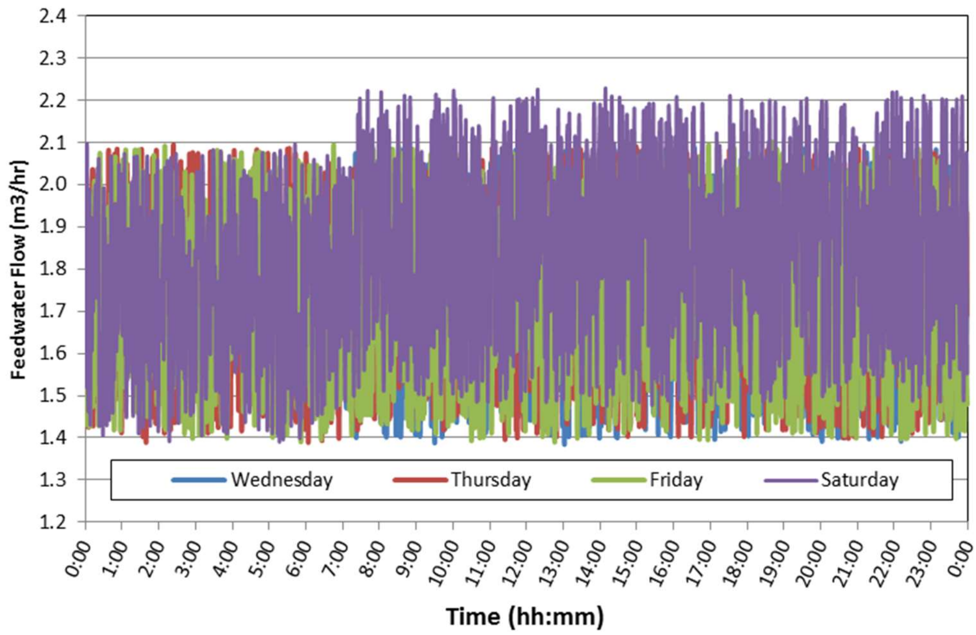


Figure 15.2 Feedwater flow rate to the boiler

System operating performance is summarised below:

Descriptions	Measured
Average feedwater flow rate (kg/hr)	1750
Average Feedwater Temperature (°C)	60
Flue gas temperature (°C)	200
Boiler operating efficiency (%)	75

Table 15.1 Summary of important performance parameters

Main findings:

- Flue gas temperature of 200°C was higher than the minimum exit stack temperature of 135°C required for diesel to prevent condensation
- The feedwater temperature is relatively low at 60°C.

Main recommendations

The main recommendation was to install an economiser to recover heat from the exhaust flue gas and preheat the boiler feedwater to about 80°C.

Estimated savings

The approximate energy and cost savings achievable for the recommended measure was as follows:

Energy savings = 400,000 kWh/year

Energy cost savings = $ 22,000/year

* Diesel cost is taken as $0.6/litre

Summary

This chapter described the parameters to be measured during an audit of a heat recovery system followed by how to compute the various key performance criteria that provide an indication of the performance of the system.

References

1. Arora, C P, Heat and Mass Transfer, Khanna Publishers, Delhi, 1986.
2. Bergman, Lavine, Incropera, and DeWitt, Fundamentals of Heat Transfer, 7th edition, John Wiley & Sons, New York, 2011.
3. Cornwell, K, The Flow of Heat, ELBS and Van Nostrand Reinhold Company Ltd, London, 1977.
4. Jayamaha, Lal, Energy-Efficient Industrial Systems, Evaluation and Implementation, McGraw-Hill, New York, 2016.

STEAM AND COMPRESSED AIR SYSTEMS ABRIDGED SYLLABUS

Compressed Air Systems

General introduction

- Components of compressed air system
- Definitions parameters such as standard pressure and temperature, standard flow rate, dew point, free air delivery, utilisation factor, specific power.
- Types of compressors (positive displacement, dynamic)
- Selection of compressors for different industries
- Basic theory of compression (isothermal, adiabatic, polytropic)

Design considerations

- Power consumption calculation
- Comparison of specific power
- Efficiency (isothermal, volumetric)
- Single stage, multi-stage, intercooling
- Heat recovery potential
- Different types of dryers, requirements, comparison of energy consumption

Energy saving measures

- Receiver sizing
- System pressure losses
- Compressed air leaks, quantification
- Compressor intake temperature
- Compressor controls strategies

Steam Systems

General introduction

- Steam generation processes
- Types of steam
- Properties of steam
- Steam tables
- Calculation of steam properties

Boilers

- Introduction to boilers
- Basic boiler operation
- Fuels
- Boiler blow down & draught
- Air fuel ratio, effect of excess air
- Combustion efficiency
- Calculation of boiler overall efficiency
- Quick estimation of boiler efficiency

Optimising steam systems

- Steam pressure
- Fuel switching
- Boiler blow down
- Auxiliary equipment

- Heat recovery from flue gas
- Condensate recovery
- Steam leaks
- Steam traps
- Other measures

Waste Heat Recovery System

General introduction

- What is waste heat?
- Benefits of waste heat recovery
- Sources and use of waste heat
- Calculation involving waste heat recovery
- Parameters controlling waste heat recovery

Modes of heat transfer

- Conduction, Convection and Radiation
- Heat transfer through composite materials and radial heat transfer

Heat exchangers

- Parallel flow
- Counter flow
- Cross flow
- Performance of heat exchangers
- Log mean temperature difference
- Correction factors for cross flow heat exchangers
- Shell and tube heat exchangers
- Correction factors for shell and tube heat exchangers
- Plate type heat exchangers
- Extended surfaces (fins)
- Fouling of heat exchangers
- Overall heat transfer coefficient of heat exchangers
- Typical values of overall heat transfer coefficient

Commercially available waste heat recovery devices

- Shell & Tube Heat Exchanger
- Heat Pipes
- Heat Wheels

REVIEW QUESTIONS

1. What is the approximate percentage of input energy to a compressor that is converted to heat of compression?
   (a) 10%
   (b) 20%
   (c) 40%
   (d) 80%

2. Dew point temperature in compressed air systems refers to:
   (a) the temperature of air at the inlet to the compressor
   (b) the temperature of air after the dryer
   (c) a measure of moisture in compressed air
   (d) the temperature of air after the air receiver tank

3. Specific power for a compressor is defined as:
   (a) power input / free air delivery
   (b) amount of compressed air produced / power input
   (c) power input / power output
   (d) none of the above

4. Which out of the following is the best control strategy for rotary screw compressors to minimise energy consumption when operating at part load conditions:
   (a) inlet modulation
   (b) variable displacement
   (c) venting at discharge
   (d) load / unload

5. Thermodynamically the best efficiency can be achieved for which type of compression process:
   (a) polytropic
   (b) adiabatic
   (c) isothermal
   (d) isobaric

6. Compressed air demand in a plant can be reduced by:
   (a) increasing system pressure
   (b) reducing leaks
   (c) reducing intake temperature to the compressor
   (d) all of the above

7. Which of the following increases compressor motor energy consumption:
   (a) higher compressor discharge pressure
   (b) lower compression ratio
   (c) lower intake air temperature
   (d) lower air demand

8. Advantage/s of using steam as a source of heat for industrial application is:
   (a) high heat capacity
   (b) ease of transportability
   (c) low toxicity
   (d) all of the above

9. Once water reaches the boiling temperature, further addition of heat will cause:
   (a) increase of water temperature
   (b) phase change from liquid to vapour at constant temperature
   (c) increase of water temperature and phase change from liquid to vapour
   (d) decrease of water temperature and phase change from liquid to vapour

10. Quality of steam or dryness fraction of steam is defined as:
   (a) mass of dry steam / (mass of dry steam + mass of water in suspension)
   (b) mass of water in suspension / (mass of dry steam + mass of water in suspension)
   (c) mass of dry steam / mass of water in suspension
   (d) mass of water in suspension / mass of dry steam

11. Which of the following can be considered as a property of a good fuel for boiler application:
   (a) high ignition point
   (b) low calorific value
   (c) can undergo complete combustion with minimum air
   (d) all of the above

12. Main purpose of intermittent bottom blow down of boiler is:
   (a) to remove suspended solids in boiler water
   (b) to remove dissolve solids in boiler water
   (c) to remove both suspended and dissolve solids in boiler water
   (d) to maintain boiler design pressure

13. Purpose of boiler draught is to:
   (a) supply required quantity of fuel to the combustion chamber
   (b) to discharge ash from the combustion chamber to the atmosphere
   (c) to discharge the combustion gases to the atmosphere through the chimney
   (d) all of the above

14. Which of the following statements is true for an induced draught of boiler:
   (a) pressure inside the furnace is above the atmospheric pressure
   (b) pressure inside the furnace is below the atmospheric pressure
   (c) pressure inside the furnace is closer to the atmospheric pressure
   (d) none of the above

15. If too much excess air is supplied to the combustion chamber of a boiler:
   (a) the increased excess air will help for complete combustion of fuel
   (b) the increased excess air will help to develop balanced draught
   (c) the increased excess air will carry away heat resulting in a drop in boiler efficiency
   (d) the increased excess air will help to flow the flue gas with reduced resistance

16. If too much excess air is supplied to the combustion chamber of a boiler, which of the following statement on concentration of O_2 and CO_2 in flue gas will be true:
   (a) concentration of both O_2 and CO_2 will increase
   (b) concentration of O_2 will increase and concentration of CO_2 will decrease
   (c) concentration of O_2 will decrease and concentration of CO_2 will increase
   (d) concentration of both O_2 and CO_2 will decrease

17. For effective heat transfer from a liquid to a gaseous fluid, fins should be added:
    (a) on heat exchanger surface over which gaseous fluid flows as heat transfer coefficient of gaseous fluid is relatively low
    (b) on heat exchanger surface over which gaseous fluid flows as heat transfer coefficient of gaseous fluid is relatively high
    (c) on heat exchanger surface over which liquid fluid flows as heat transfer coefficient of gaseous fluid is relatively low
    (d) on heat exchanger surface over which liquid fluid flows as heat transfer coefficient of gaseous fluid is relatively high

18. For the same heat flow rate (under same inlet and outlet temperatures), required heat transfer surface area for:
    (a) counter flow heat exchanger is less than parallel flow
    (b) counter flow heat exchanger is more than parallel flow
    (c) both counter and parallel flow heat exchangers are equal
    (d) none of the above

19. FAD in compressed air systems refer to conditions at:
    (a) compressor intake
    (b) compressor discharge
    (c) at STP conditions
    (d) none of the above

20. Which of the following parameters is not required for computing the volumetric efficiency:
    (a) free air delivery (FAD)
    (b) power
    (c) stroke length
    (d) cylinder bore diameter