Abstract

This report provides a guide to the principles of combustion-based steam cycle plants and combined (gas and steam) cycle plants fuelled by coal. The main types of power generation cycle are introduced, followed by background on the thermodynamics of heat engines and gas and steam cycles. The chapter on steam plant contains sections on PCC boilers, main features of turbines and the cycles themselves. The descriptions cover the influence of appropriate pressures and temperatures, designs of feed heating trains, use of reheat, export of heat, subcritical and supercritical cycles and other aspects, such as materials developments. The chapter on combined cycles concentrates on the two main types (pressurised fluidised bed combustion and integrated gasification combined cycles), describing configurations and steam cycles for these technologies. There are discussions of design aspects and of the influence of parameters including technology type, carbon utilisation, gasification efficiency, gas turbine and gas clean-up. Materials issues for combined cycle plants are discussed. Future power cycles based on coal will probably involve new configurations to accommodate carbon dioxide (CO₂) capture and storage. Examples are given to illustrate how these will impact on the energy flows.
## Acronyms and abbreviations

<table>
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<tr>
<th>Acronym</th>
<th>Description</th>
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<tbody>
<tr>
<td>A-PFBC</td>
<td>Advanced pressurised fluidised bed combustion (Japan)</td>
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<tr>
<td>BFP</td>
<td>boiler feed pump</td>
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<td>BGL</td>
<td>British Gas/Lurgi (gasifier)</td>
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<td>CFBC</td>
<td>circulating fluidised bed combustion</td>
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<td>CHP</td>
<td>combined heat and power</td>
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<td>CO₂</td>
<td>carbon dioxide</td>
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<td>COS</td>
<td>carbonyl sulphide</td>
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<td>DTI</td>
<td>Department of Trade and Industry (UK)</td>
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<td>EAGLE</td>
<td>Coal Energy Application for Gas, Liquid and Electricity (Japan)</td>
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<td>EU</td>
<td>European Union</td>
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<td>FGD</td>
<td>flue gas desulphurisation</td>
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<td>GE</td>
<td>General Electric Company (USA)</td>
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<td>HCl</td>
<td>hydrogen chloride</td>
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<td>H₂S</td>
<td>hydrogen sulphide</td>
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<td>HHV</td>
<td>higher heating value</td>
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<td>HP</td>
<td>high pressure</td>
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<td>IEA</td>
<td>International Energy Agency</td>
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<td>IGCC</td>
<td>integrated gasification combined cycle</td>
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<td>IGFC</td>
<td>integrated gasification fuel cell</td>
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<td>IP</td>
<td>intermediate pressure</td>
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<td>ITM</td>
<td>ion transport membrane</td>
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<td>kJ</td>
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<td>kW</td>
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<td>LHV</td>
<td>lower heating value</td>
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<td>MCFC</td>
<td>molten carbonate fuel cell</td>
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<td>MCR</td>
<td>maximum continuous rating</td>
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<td>MDEA</td>
<td>methyldiethanolamine</td>
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<td>MEA</td>
<td>monoethanolamine</td>
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<td>MHI</td>
<td>Mitsubishi Heavy Industries</td>
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<td>MPa</td>
<td>megapascal</td>
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<td>MWe</td>
<td>megawatt electrical</td>
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<td>NETL</td>
<td>National Energy Technology Laboratory (US DOE)</td>
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<td>NOx</td>
<td>oxides of nitrogen</td>
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<td>O₂</td>
<td>oxygen</td>
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<td>PC</td>
<td>pulverised coal</td>
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<td>PCC</td>
<td>pulverised coal combustion</td>
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<td>PFBC</td>
<td>pressurised fluidised bed combustion</td>
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<td>p-v</td>
<td>pressure-volume (diagram)</td>
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<td>SCR</td>
<td>selective catalytic reduction</td>
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<td>SO₂</td>
<td>sulphur dioxide</td>
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<td>SOFC</td>
<td>solid oxide fuel cell</td>
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<td>US DOE</td>
<td>US Department of Energy</td>
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Coal-fired power generation plants are most commonly based on pulverised coal combustion (PCC) systems, in which heat from combustion of the coal is used to raise high pressure superheated steam that drives a steam turbine generator. Steam turbine plants have been in use for over a hundred years, and have reached supercritical conditions with percentage efficiencies in the upper 40s, LHV (lower heating value) basis, at favourable locations. Coal-fired power plants can also be based on combined gas and steam cycles, which use gas turbines as well as steam turbines. This report provides an introduction to the principles of both types of plant.

The decision on whether to go for combined cycle or on what type of combined cycle to opt for is connected with issues such as the attitude to risk of the utility, the environmental requirements imposed by the regulating authority, the type of duty cycle envisaged, and various other possibilities. Such choices depend on the owner’s priorities, such as maximising full load efficiency or ensuring operational flexibility, minimising capital or running costs, and so on. For example, it is unlikely that a new plant based on entrained gasification in an integrated gasification combined cycle (IGCC) will have been selected if the electricity demand cannot be relied upon to provide base loading for most of the time (because of the probably higher investment cost and lower degree of flexibility than for combustion-based simple cycles), or if a highly refractory high ash coal is the design fuel; a combustion-based combined cycle will suit a lower quality fuel or smaller plant, and so on.

The report deals with the heat utilisation aspects of power plants, rather than fuel handling or combustion/gasification reactions. It includes background on the thermodynamics of heat engines to support the discussion on cycles. Materials issues are included in the considerations as they have an important bearing on the performance that is now and will in future be attainable.

Future power cycles based on coal will probably involve new configurations to accommodate carbon dioxide (CO₂) capture for storage. Whatever the means to be adopted, they will all involve changes to the energy flows within the plants to some degree. Integration aspects will be important. Such cycles are also introduced in this report.

The report is organised as follows. In Chapter 2, cycles based on steam turbines, gas turbines and combined cycles are introduced. Chapter 3 outlines the thermodynamics of heat engines and steam cycles. Chapter 4 considers steam plant cycle designs. Chapter 5 looks at combined cycle plants. The principal parameters determining performance are described in both of these chapters. Chapter 6 discusses future developments in power cycles, particularly incorporating CO₂ capture and storage. Main conclusions are in Chapter 7.
2 Types of power generation cycles

Figure 1 shows the basic principle of combustion-based power generation using a steam turbine. Combustion of the fuel generates heat that is used to convert water pumped to high pressure into high pressure superheated steam, which is then used to drive a turbine. The turbine drives a generator. Steam emerging from the final stages of the steam turbine is recondensed to water which is then pumped back to the boiler. In the steam turbine itself, the steam is expanded and its pressure reduced through a sequence of stages of nozzles (fixed blades) and their associated rotors (moving blades) while its energy is converted into rotational energy of the rotor shaft. The steam leaving the high pressure turbine at intermediate pressure is usually reheated before further expansion, to raise efficiency.

Other systems that make up a PCC plant are integrated within the thermodynamic cycle, for example, the air heater that recycles low grade heat to the boiler. Flue gas reheat after flue gas desulphurisation (FGD) systems can take heat from the thermodynamic cycle if they use steam heating, as commonly applied in the USA, although newer FGD have feed/effluent gas/gas heat exchangers that avoid losses from this source. Where a significant accessible market exists, heat may be exported from the plant in addition to electricity, increasing the overall thermal efficiency while reducing the electrical efficiency. For example, district heating schemes can be supplied with heat from the cooler parts of the water/steam cycle.

Combustion (gas) turbines (Figure 2) convert heat into shaft power by using air as the working fluid rather than steam. The air is first compressed, usually in a multi-stage axial compressor, to around 1.5 MPa (higher pressures up to 3 MPa are used in aero-engines and their land-based derivatives). The air is then heated by combustion of a fuel in it, and the added energy is exploited by expansion of the hot product gases in the turbine section. As in a steam turbine, the fluid is expanded and its pressure reduced as it passes through a sequence of stages of stators and rotors while its energy is converted into rotational energy. The turbine directly drives the compressor, and the balance of motive power drives the generator. Gas turbines are most easily designed for fuelling on natural gas and distillate oils, but coal-derived gas can be used, as discussed later.

Key properties of a gas turbine are the inlet air flow rate, the turbine inlet temperature and the pressure ratio (the compressor delivery pressure divided by the turbine outlet pressure). The air flow rate sets the output, while the other two parameters determine the efficiency. As gas turbine technology has evolved, entry temperatures have been increased, raising efficiency. The most advanced gas turbines used in power generation have turbine inlet temperatures approaching 1400°C. For example, the firing temperature of GE’s Frame 9FB machine is over 1370°C (Modern Power Systems, 2002). This has been achieved through the use of improved alloys and blade cooling systems using air channels aspirating through holes in the blades. Increasing the pressure ratio raises efficiency up to a maximum, before it falls off as the compressor power requirement rises faster than the power developed by the turbine. The optimum point lies at higher pressures as the turbine inlet temperature is increased. As most gas turbines exit to atmosphere, ambient conditions generally affect output. The lower the turbine outlet pressure, the greater the delivered motive power obtained by expansion of the gases.

Exhaust gases leaving gas turbines are typically at a temperature of 550–600°C, sometimes higher, and this is sufficiently high for the production in a waste heat boiler of high pressure superheated steam for expansion through a steam turbine to form a combined cycle (see Figure 3). Combined cycles enable more energy to be obtained, because they allow a higher upper temperature in the heat engine cycle (gas turbine inlet temperature), while keeping the same lower temperature of final heat rejection (wet steam outlet from the final low pressure steam turbine stage). Reheat is not always used in the steam cycles of combined cycles, depending on cost-effectiveness.

The use of coal as a fuel for plants employing gas turbines necessitates measures to limit the amount of entrained particulates and liquid droplets (slag) passing through the turbine to avoid damage and so achieve practical systems. Two principal approaches have been adopted. The first uses pressurised fluidised bed combustion (PFBC) in a bubbling bed at around 1.0–1.5 MPa and 850°C, followed by particulates removal using cyclones (or sometimes a ceramic
filtration system), before expansion of the cleaned pressurised hot flue gas through the turbine expander (see Figure 4). The pressurised air feed to the combustor is driven by the expander, so the system resembles a normal gas turbine, but with the combustor located remote from the rest of the turbine. The combustion temperature is limited to around 850ºC to avoid sintering of the coal ash in the fluidised bed, which can lead to defluidisation.

The other approach is first to convert the feed coal into a fuel gas, using a pressurised gasification system, then to clean the gas before firing it in a more conventional gas turbine combustor (though designed to accommodate a medium calorific value gas rather than natural gas). This permits higher turbine inlet temperatures to be achieved than in PFBC and also allows much greater variation in cycle configurations to suit different gasifier types. Figure 5 shows the principle of these integrated gasification combined cycle (IGCC) systems. Water flows are not shown, to simplify. The majority of types of gasifier that have been used for IGCC are of an oxygen-blown entrained design, either using a liquid coal slurry feed or a fine dry coal feed. However, other gasifier types can be used. IGCC oxygen supply arrangements range from complete integration, in which all of the air for the air separation plant is provided by the gas turbine compressor, to zero integration in which the air separation plant has its own air supply compressors. The optimum choice depends on flexibility required and the type of gas turbine to be used.

Neither PFBC nor IGCC is yet widely established but IGCC in particular is promising for the future, with its low
emissions, high potential efficiency and scope for adaptation to near-zero emissions and co-production of hydrogen (Henderson, 2003a,b). There are various sources of heat from the gas cycle in IGCC cycles apart from that in the turbine exhaust gases, and these heat flows have to be integrated to the maximum extent with the steam cycle for satisfactory practical systems. Some heat or steam consumption may be internal to the plant, for example steam may need to be sent to a coal gasifier.

Gas turbine cycles do not have to be heated by burning fuel within the working fluid. External (indirect) heating can in principle be used, thus enabling fuels such as coal to be used without gas clean-up but, because high temperatures are needed for useful efficiencies from gas turbines, special materials are required and such air heater systems are still developmental.

Figure 5  Principle of integrated gasification combined cycle power generation
This chapter introduces some information on the thermodynamics of heat engines and steam cycles to facilitate reading in context the information in later chapters. It is necessarily kept brief.

A heat engine is a device for extracting useful work from heat by utilising the expansion of a gas. The Second Law tells us that it is not possible to convert all of the heat energy supplied to an engine into work, but that some energy will inevitably be unused and rejected at a lower temperature. We cannot simply take out all the energy fed in as heat in the form of the equivalent amount of work. However, the First Law of Thermodynamics also holds – energy must be conserved: the total heat energy input equals the total energy obtained from the engine as work done plus rejected heat at a lower temperature.

The operation of an ideal engine between temperatures $T_1$ and $T_2$ is represented by the Carnot cycle (Figure 6). This shows, in the form of a pressure-volume ($p$-$v$) diagram, a cyclical sequence of reversible processes performed on a gas, involving isothermal (constant temperature) expansion ($A \rightarrow B$) at temperature $T_1$, adiabatic (without heat addition or extraction) expansion ($B \rightarrow C$), isothermal compression ($C \rightarrow D$) at temperature $T_2$, and adiabatic compression ($D \rightarrow A$). The engine is drawing heat from the heat source at $T_1$, performing work by expansion, then returning a smaller quantity of heat to a sink at $T_2$. A reversible process is a process that occurs at equilibrium throughout: it is unattainable in practice, but may be approximated where changes occur slowly. Irreversibility will result in lost opportunities for producing work. An example is friction. Inefficiencies emerge ultimately as heat flows from the system over and above that of the ideal engine. The work done by the ideal engine around the Carnot cycle is the sum of all the $p$-$v$ work and so is given by the area enclosed by the $p$-$v$ diagram.

![Figure 6 p-V diagram of Carnot cycle](image)

The proportion of the heat supplied that is converted into mechanical work is referred to as the efficiency of the cycle. For the ideal engine, the efficiency is related to the initial and final temperatures of the system as follows:

$$\eta = \frac{T_1 - T_2}{T_1}$$

where $\eta$ is the efficiency and $T_1$ and $T_2$ are the upper and lower (initial and final) temperatures between which the engine operates, expressed in degrees Kelvin. Because thermal energy exists in the form of molecular translational, rotational and vibrational motions, and such motions cease at 0 K (−273.15°C), the only circumstance in which no heat would be rejected would be for an impossible engine that would operate with its temperature of heat rejection at 0 K, when the above equation would give an efficiency of conversion to mechanical work of 1, or 100%.

The lowest temperature in the steam turbine cycle depends on the temperature of the coolant water that cools the steam and condensate leaving the low pressure outlet of the turbine system. Sites with access to cold sea water can reach lower bottom end temperatures in the cycle and so higher efficiencies. There is more scope to increase the upper operating temperatures of steam turbines (subject to materials availability), as has occurred over the past 100 years or so for this reason. For an upper temperature of 600°C (typical of a state-of-the-art supercritical turbine) and lower temperature of 20°C (typical for sea water cooling at a North European coastal location), the theoretical maximum efficiency given by equation (1) is 66%.

In practice steam cycle efficiencies are lower. For example, a calculation based on the gross efficiency of the Boxberg lignite-fired plant in Germany, given as 48.65% by Smith (2001), indicates a steam cycle efficiency of around 52–53%, assuming a boiler efficiency of 92%, LHV basis. Welford and others (2002) have calculated a gross efficiency of 48.5% and net plant efficiency of 45.1%, LHV basis, for a notional 30 MPa/600°C/620°C plant at a UK inland location, implying a similar steam turbine cycle efficiency. Materials development programmes are in progress to allow higher temperature and pressure conditions. These and cycle design advances could ultimately realise steam cycle efficiencies of around 60% (estimated from an overall efficiency of 52%, LHV implying a gross efficiency of 56%, LHV basis).

An important property, entropy ($S$), can be defined by the equation:

$$\Delta S = \frac{\delta Q_{rev}}{T}$$

Here, $\delta Q_{rev}$ represents the heat absorbed during a reversible change at temperature $T$ (Kelvin). The entropy change, $\Delta S$, effectively measures the energy dispersion in a system as a function of temperature. An increase in entropy represents heat energy that is unavailable for doing work. Operation of an ideal engine around a complete cycle will therefore occur
with no overall change in entropy and any work accompanying such a process will be accompanied by an equivalent decrease in enthalpy. Entropy can be expressed per unit mass as specific entropy. Temperature-entropy ($T$-$S$) diagrams can be plotted for heat engine cycles and are particularly useful in consideration of steam cycles.

Steam turbine cycles involve changes of state and the Rankine cycle was proposed independently by Rankine and Clausius to accommodate this. In these systems, all of the compression is performed on the working fluid while it is liquid, so less work is required for that stage than in the Carnot cycle. Figure 7 shows a basic Rankine cycle in outline, together with its $p$-$v$ and $T$-$S$ diagrams.

The heat from the boiler increases the enthalpy (known sometimes as heat content) of the water as its state, temperature, volume and pressure change. Enthalpy ($h$) is defined as:

$$h = u + pv$$  \hspace{1cm} (3)$$

where $u$ is the internal energy, $p$ is the pressure, and $v$ is the volume of a system. The internal energy of a system includes the sum of potential and kinetic energy as well as that from energy transferred as heat, which is associated with the motional energies of the constituent molecules. The latter can for practical purposes be thought of as accounting for all of the internal energy in the context of heat engines. The enthalpy of a fluid can be seen from the above to be the total of the energy arising from its pressure and volume and from its temperature (internal energy). For both the $p$-$v$ and $T$-$S$ diagrams the work done by the engine is given by the areas enclosed. In practice, irreversibilities and application of features such as steam reheat alter the shape of the diagrams somewhat.

It is convenient in consideration of heat engines to look at changes in properties such as enthalpy, entropy and internal energy, since, although they are state functions (that is, changes depend only on initial and final states, regardless of the pathway between them), their absolute values cannot be known. If heat is absorbed at constant volume, the increase in energy is the change in internal energy. If volume changes during the absorption of heat, so that work is done, the heat absorbed is the change in enthalpy. Here, for the Rankine cycle, enthalpy is the appropriate measure of the heat absorbed since it applies to energy absorbed that changes the internal energy and the product $pv$. Tabulated specific enthalpies (that is, per unit mass of water/steam), for instance, are referenced to a standard condition, for example they may be defined as zero at 0°C and 100 kPa in a gaseous or vapour phase.

Referencing to the specific enthalpies around the major process areas of the cycle (Figure 7):

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**Figure 7  Simple Rankine cycle with $p$-$V$ and $T$-$S$ diagrams** (adapted from Goodall, 1981)
where \( Q_b \) is the heat absorbed from the boiler (kJ), \( m \) is the mass flow of water/steam (kg/s), and \( h_1 \) and \( h_2 \) are the specific enthalpies of the steam or water entering and exiting the boiler, respectively (kJ/kg).

The increased enthalpy of the steam is exploited in the turbine, which if ideal will convert all of the enthalpy drop across it into work. Referring to Figure 7:

\[
W_t = m(h_2 - h_3)
\]

where the work done by the ideal turbine is \( W_t \) (kW). In practical turbines, the work actually done by the turbine, \( W_{act} \), is reduced by irreversibilities, and these are reflected in the isentropic efficiency of the turbine, \( \eta_i \) (ideal processes have no change in entropy associated with them), which may be of the order of 80–95%.

\[
W_{act} = m(h_2 - h_3)\eta_i
\]

Although saturated steam can be used, and expansion of wet steam occurs in the low pressure stages of power plant turbines, raising the enthalpy of the main steam through superheating is needed to achieve sufficiently high efficiencies and to ensure that the dryness fraction does not fall below about 90%, to minimise water droplet-induced erosion in the low pressure cylinders. The higher energy content in superheated steam also enables the specific steam consumption to be reduced. Further improvements in steam consumption, efficiency and steam dryness are also achieved by employing reheat of the intermediate pressure steam exiting the high pressure turbine (usually to a temperature comparable with that of the superheated steam) before expanding it through the remaining turbine stages. Reheat raises the average upper temperature of the thermodynamic cycle.

The equations are used to calculate the required distribution of flows within the steam cycle once the temperature and pressure conditions (and hence specific enthalpy) at each point have been set.

Turbine cycles are often characterised by their heat rate, which is a measure of the heat supplied compared with the gross electrical output in kWe from the turbine. It therefore embraces generator losses also. Commonly used units for heat rate are kJ/kWh:

\[
\text{Heat rate (kJ/kWh)} = 3600\frac{Q_b}{W_e}
\]

The heat available to the steam cycle in a power station will be available over different temperature ranges. Practical systems make optimum use of the different grades of heat in order to realise maximise efficiency. Viewing the design of energy utilisation and transformation systems from the point of view of ensuring that higher grades of heat are utilised only when all reasonable use of lower grade sources has been made has led to the concept of \textit{exergy}, or available energy. Exergetic analysis is now a firmly established sphere of applied thermodynamics, of value in ensuring that these important aspects are considered (in, for example in another area, questioning whether it is appropriate to use the high temperature heat from a flame purely for heating water at low pressure to less than 100ºC for space heating systems). A loss of exergy represents the destruction of work potential (De and others, 2003). In practice, some form of trade-off will be necessary in deciding details of design to achieve optimum generation cost at some sacrifice of efficiency.

Increases in main steam pressure lead to an increase in evaporation temperature and reduced heat of vaporisation, enabling greater temperature differences across the cycle to be achieved, and may make more room for an additional reheat stage. All these factors favour higher efficiency cycles. Other important factors in practice include increasing boiler feedwater temperatures (this effectively raises the upper temperature of the thermodynamic cycle) and choice of turbine pressure stages.

The pressure-volume diagram for a gas turbine cycle, known as the Brayton cycle, is shown in Figure 8. The curve 1–2 represents the reversible adiabatic compression in the compressor, 2–3 is the line of heating at constant pressure, and 3–4 represents reversible expansion in the turbine. The line 4–1 represents the process of heat rejection by the exhaust gases. The work used by the compressor is represented by the area a21b and the work performed by the turbine by a34b. The area of the diagram 1234 is the net work output (in practice, irreversibilities alter the shape of the diagram slightly).

The efficiency of a gas turbine is typically around 30% because of the high exhaust gas temperature but the latter is turned to advantage in associating the gas turbine with a steam turbine to form a combined cycle. This uses the high temperature gas turbine exhaust heat as the source of energy for a Rankine cycle. State-of-the-art natural gas fired combined cycles have thermal efficiencies typically of 58%, LHV basis. Coal-fired combined cycles were described in Chapter 2 and are looked at in more detail in Chapter 5.

There are ways to avoid the inherent energy rejection from

\[p-V\text{ diagram for Brayton cycle}\]
the Second Law. These utilise other systems for converting the chemical energy in the fuel into work. An example is the fuel cell, which uses electrochemical processes to convert the fuel’s chemical energy into electricity. However, adequate fuel utilisation has to be achieved through recycling the fuel gas to the fuel cells and/or combustion of the off-gas. In large scale power generation applications from syngas, fuel cells will have to be integrated with heat engine cycles for achieving highest electrical efficiencies because of the high thermal energy content of the exit gases. The use of fuel cells is discussed in Chapter 6. It is also possible to convert heat directly to electricity using the thermoelectric effect but technologies for this are currently applied only to very specialised, small-scale applications such as for space probes.
The vast majority of combustion-based single cycle steam plants fuelled by coal utilise pulverised coal combustion (PCC). In a PCC power station unit, heat from combustion of coal is used to raise high pressure superheated steam which is used to drive a turbine to generate power. This chapter is mainly concerned with the steam turbine cycle but an initial section discusses the steam generator (boiler). Firing systems are not discussed.

### 4.1 PCC boilers

A drum-type subcritical PCC boiler takes the pressurised preheated boiler feedwater from around 250–260°C to evaporation point then superheats it to 540°C or above for sending to the HP turbine. A supercritical boiler heats the pressurised preheated (260–300°C) water beyond the critical point (22.1 MPa, 374°C), above which water and steam are indistinguishable, to superheat temperatures. As it is heated, the water, already at supercritical pressure, changes smoothly into vapour, when the critical temperature is passed, without a liquid/vapour boundary becoming discernible. In practice, turbulence (pseudo-boiling) can be observed through its effect on heat transfer rates. Beyond the critical temperature, vapour and liquid are not distinguishable and two phases do not exist. In both supercritical and subcritical boilers, intermediate pressure steam returning from the high pressure turbine is normally reheated to main steam temperature then returned to the intermediate pressure turbine.

Figure 9 shows in simplified diagrammatic form a drum-type subcritical boiler. The economiser forming the last stage of the boiler’s convective section takes the incoming feedwater’s temperature to about 60°C below evaporation temperature for sending on to a steam drum/evaporator recirculation loop. The economiser takes the flue gases down to around 350°C. Lower temperature heat down to about 100–150°C is recovered by transferring the heat to the incoming combustion air using an air heater. Evaporation takes place using mainly radiant heat, which is transferred to water carrying tubes forming the wall of the furnace zone of the boiler. The ensuing water/steam mixture passes to a large steam drum, which allows the steam to separate, and the water returns to the evaporator. Either forced or natural circulation is used to send the water round the loop. Steam leaving the drum is then sent to the primary superheater.

In both subcritical and supercritical boilers, superheat and reheate heat transfer surfaces are mounted above the furnace and in the convection section of the boiler. Superheaters and reheaters are generally of pendant and/or horizontal type, each with at least two stages (to simplify, only two for superheat and one for reheate are shown on Figure 9). To control steam temperatures, facilities are usually provided for

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**Figure 9  Drum-type boiler (diagrammatic)**

*Understanding coal-fired power plant cycles*
The attemperation of superheated and reheated steam by injection of boiler feedwater sprays between the stages of heating (attemperation features are not shown on the simplified figure). Attemperation water may be taken at final feedwater temperature and pressure or from the main feed pump outlet for superheat steam attemperation or from a tapping within the main feed pump for reheat steam attemperation. However, since there is a significant impact on cycle efficiency, use of these, especially the latter, is minimised so alternative methods to control steam temperature, involving adjusting heat transfer rates, by tilting burners downwards, use of flue gas recirculation or even adjustment of excess air levels may be provided. Once-through boilers sometimes omit the facility for attemperation, using instead close matching of firing rate to feedwater flow for steam temperature control.

The familiar two pass design may be substituted by a tower type that can offer some advantages, including less potential for tube bend erosion and allowing the boiler to be more completely drainable (Stultz and Kitto, 1992). The economiser may be mounted either above the superheater and reheater sections or above the air heater (Scheffknecht and others, 2003).

The configuration of a typical once-through supercritical boiler is shown in Figure 10 in simplified form. Supercritical boilers have smaller economisers than equivalent subcritical boilers as the bleed steam feedwater heating system delivers the water at a higher temperature. Once-through boilers do not have a steam drum apart from small water/steam separators for use during start-up (Lorey, 2003; Siemens, 2001). During normal operation, the separators (not included in the figure for clarity) are unnecessary and are bypassed.

Once-through boilers require a different tube arrangement from drum boilers within the furnace area. In the absence of recirculation there is inevitably a lower liquid mass flow rate within the furnace wall tubes, so spiral wound furnace tube designs in the zone near to the pulverised coal (PC) burners have commonly been adopted to give a longer path and higher water mass flow in the tubes for greater cooling, to prevent tube damage arising from overheating (see Figure 11). As an alternative, internally rifled tube designs are available, offering improved heat transfer and lower wall temperatures at much lower mass flows through induced swirl (Figure 12). This allows vertical tubing to be used throughout the furnace, giving greater operating flexibility (Scheffknecht and others, 2003). Overall, the pressure drop is reduced, as less pipe length is needed, so pumping power is lower than for spiral wound designs (Luby and Susta, 2002).

As supercritical cycles have evolved, improved steels have been developed to cater for those parts of the boiler that are under particular stress. The use of alloys for the critical pressure part components such as waterwalls, final superheat and reheat sections, and main steam headers and piping has been summarised by Scheffknecht and others (2003). The extension to future, more severe conditions is discussed in Section 4.3.4 of this report.

Other types of coal-fired boilers that are used for steam turbine generation units are based on atmospheric fluidised
bed combustion, of which the most widely deployed are of the circulating type (CFBC). The application of the heat from such systems is very similar to that for PCC (if required, descriptions of CFBC technology can be found in other recent publications by the IEA Clean Coal Centre by Rousaki and Couch (2000) and Henderson (2003a)).

The next section looks at steam turbines and the steam cycles in which they are used, reviews the influence of parameters on performance and looks at the use of new materials for future improvements.

4.2 Steam turbines

4.2.1 General arrangement

In a modern large steam turbine, superheated steam at a high temperature and high pressure is expanded while its energy is converted into mechanical work. A turbine (Figure 13) contains a sequence of stages of nozzles (stators, or fixed blades) and their associated rotors (moving blades). Steam is admitted via a stop valve, which can trip to shut off the steam flow in emergency situations, then flows through control valves that allow the correct flow rate for the power output required. Pressure variation as an alternative means of setting output is also increasingly used.

In impulse turbines, energy transfer comes from the action of the steam hitting the blades. As the steam passes through each stage of nozzles, its velocity increases, its pressure drops, and the emerging high velocity stream transfers its kinetic energy to the rotor, while the steam loses velocity. The next nozzle stage then further drops the steam pressure, raising the velocity for its associated rotor. Thus, the whole of the pressure drop and attendant velocity increase are arranged to take place within the stationary nozzles. In the other main type of turbine (reaction), the pressure drop takes place over the rotor blades themselves. The rotor blades are shaped to be wider apart at the inlet side than at the outlet so that transfer of energy to the rotor comes from the recoil as the steam escapes. Conventionally, turbines have been largely impulse or about 50/50% impulse/reaction in design, but some modern turbines are designed with the reaction of each stage individually set.

At the scale of power station turbines, it is not practical or economic to design a single casing for taking the steam down from main stop valve pressure to condenser pressure, as such large degrees of volume increase and pressure drop are involved, so separate turbines are mounted on the same axle, tandem compounded into one machine. The high pressure (HP) turbine exhausts at about 25% of the pressure at its inlet, and this steam, net of that extracted for feedwater heating, enters, usually after reheat, the intermediate pressure (IP) turbine, normally a double flow design, which drops the pressure to around 0.5–1 MPa, before sending it to the low pressure (LP) turbines that exhaust at sub-atmospheric pressure to maximise expansion and work done. On large machines, two or three double-ended LP cylinders are needed to accommodate the exhaust volume flow rate. Extraction
points are located in the cylinders (casings) of the IP and LP turbines for obtaining steam at various pressures for feedwater heating and other possible uses, for example export as steam or to provide heat for export after heat exchange, for example for a district heating scheme. Where a significant medium or low pressure export steam requirement exists, back-pressure turbines may be selected. These exhaust at the required pressure for the export steam and condensing stages are not used. Such systems are not normally encountered in utility scale power stations.

The wet steam emerging from the LP cylinders is completely condensed in the condensers mounted underneath the turbines. The condensers are kept at a pressure of typically 0.003–0.005 MPa, depending on the temperature of the available cooling water. This in turn depends on the cooling system adopted and local conditions. Sea water cooling gives the lowest temperatures in temperate coastal sites, while inland sites often use cooling tower systems with natural draught cooling of recirculating water (with make-up to compensate for evaporation and blowdown). Forced draught heat exchangers may be used where smaller, lower structures are required. In the condenser, the cooling water is pumped through tubes over which the steam passes. Minimising the leakage of air into the condenser is important for maintaining adequate vacuum. Deterioration in vacuum reduces turbine output and efficiency and air in-leakage can increase the oxygen concentration in the condensate, leading to corrosion of feedwater heaters, boiler and turbine. It is also necessary maintain the temperature rise across a condenser within rated values to keep to the required vacuum at the turbine exit. Increases of 2% in heat rate due to poor heat transfer from fouling are not uncommon (Putman and Walker, 2000).

**4.2.2 Sources of losses in steam turbines**

Sources of losses in turbines include aerodynamic losses, leakage of steam and leaving loss. Computer aided design and three-dimensional packages for the optimisation of blade aerodynamic designs for maximising stage efficiencies have enabled turbine efficiencies to be improved considerably over the last 10 years through new three-dimensional blade designs. A recent development, the so-called 3DV system developed by Siemens (3D blading with variable stage reaction) offers freedom in design philosophy and so greater efficiency, avoiding the need for the same degree of reaction to be applied to all stages (Luby and Susta, 2002; World Coal, 2004).

Leakage losses occur when steam bypasses a stage by leaking past the tips of stationary and moving blades. Designs to minimise this depend on where most of the pressure drop occurs, and so on the degree of reaction. With impulse blades, most of the potential loss area is around the nozzle blades, where most effort is concentrated in sealing, but some sealing at the tip of the moving blades is usually also applied. With 50% reaction blades, more elaborate sealing systems need to be applied to the rotor blades. Water droplets also cause a drop in efficiency in wet stages from a braking effect on the rotor by impingement with the (faster moving) blades. Another source of losses is associated with the kinetic energy of the steam leaving the final LP stage. This ‘leaving loss’ is minimised by maximising the last stage exit area, but a trade-off between size and cost exists. It is generally more economic to increase the LP turbine annulus area by maximising the last stage blade length rather than to multiply the number of cylinders (Luby, 2003). Although it
may not be considered a loss as such, steam is also needed to seal the glands at the end of turbine shafts and so the enthalpy in this does not provide motive power.

Other important aspects associated with losses are the minimising of blade erosion from impingement of water droplets, by hardening of the leading edge of the blades of wet stages, and understanding and eliminating blade vibration. In supercritical units, at the high temperatures encountered, solid particle erosion of the first and second high pressure and intermediate pressure stages can lead to efficiency loss. This has required the development of suitably robust control stage nozzles (Logan and Nah, 2002).

Larger turbines tend to exhibit higher efficiencies than smaller ones. This is not only due to it being generally more cost effective to use more sophisticated features, but also because the rotor shaft cross sectional area relative to that of the steam flow path becomes smaller. As a consequence, the power-to-weight ratio of larger turbines is greater and frictional and leakage losses are relatively smaller.

4.2.3 Steam admission and control

Steam enters the HP turbine typically through four nozzle chests, each of which is controlled by its own steam valve. Where steam flow is used to set output at part load, there are three potential means of control to set the flow rate to adjust the power output as required. These are throttle, nozzle and bypass governing. In throttle governed systems, all of the valves are operated in parallel. As the load is reduced, the turbine itself remains fairly constant in efficiency but there is a loss due to throttling by the valves. A graph of load versus steam flow (the so-called Willans line, see Figure 14) is linear above an initial steam flow needed to overcome losses, friction and back-pressure (Quayle, 1984). The plot of overall turbine heat rate shows a curve (the upper line in Figure 15).

Nozzle control governing involves changing the number of steam introduction nozzles that are open as the load changes. Only two are used up to a certain load. When these are fully open, a third nozzle is used to supply the additional flow as demand for output rises. Finally, the fourth nozzle is opened at loads above that obtaining when the first three are fully open. There is therefore less throttling loss at reduced outputs and turbine heat rate is better. The Willans line is not quite straight and a graph of heat rate versus output is as shown by the lower line on Figure 15.

In the less commonly encountered bypass governing, the extra steam flow above a certain output is taken from before the HP turbine and introduced further along the turbine train.

A method of regulating output that is gaining in popularity is sliding pressure operation. This is discussed below.

4.2.4 Sliding pressure units

Boiler-turbine units are increasingly being designed for sliding pressure operation to achieve better cycling performance. In this case, boilers are generally once-through supercritical designs, as more rapid output changes can thereby be achieved because of their lower thermal inertia. At the reduced pressure used for part load operation, when the system is operating subcritically, evaporation occurs within a region of tubing that shifts with changes in load while the enthalpy increases in the boiler associated with preheating, evaporation and superheating change with pressure. Once through boilers can increase output by 5% per minute, compared with 3% per minute for drum boilers (Luby, 2003). Such flexible systems reduce the drop in efficiency at reduced load, as better control of turbine temperatures and wetness is possible than when turbine governor valves are throttled. In addition, boiler feed pump power consumption is reduced and there are lower stresses on components such as valves. In practice, a certain degree of throttling is usually also used to maintain some reserve steam to give the capability for meeting sudden increases in power demand. Sliding pressure has been used for over ten years in units in Denmark (Kjær, 1990) and is the recommended method for new supercritical units (Fernando and others, 2000). Sliding pressure once-through boilers are provided with small steam...
drums for water/steam separation at the (subcritical) conditions of reduced load. Small separator vessels and a recirculating pump are in any case normally provided for start-up (Siemens, 2001).

### 4.3 Steam cycles

#### 4.3.1 Principles

Figure 16 (Welford and others, 2002), which is a design for a supercritical cycle with state-of-the-art 30 MPa/600ºC/620ºC steam conditions, demonstrates the principles of the modern reheat steam cycles used in power generation. Water from the condenser is pumped through feedwater heating stages then fed to the boiler for conversion into high pressure superheated or supercritical steam. This is fed to the steam turbine, expanded to around 0.003–0.005 MPa and the emerging steam condensed in the condenser, completing the circuit. Steam is taken off for feedwater heating, and high pressure water (around 2–5% of main flow) may be diverted for attemperation of the superheated and reheated steam. Steam from reheater or other superheater headers is commonly used for sootblowing to clear the boiler surfaces of deposits but air or water can also be used.

Most of the heat available to the steam cycle in a PCC power station is at a high temperature, and there is no difficulty in raising high pressure steam and heating it to superheat and reheat temperatures (typically 540–600ºC). Heat at lower temperatures is taken out via the economiser, which brings the boiler feedwater to the required temperature for introduction to the steam drum, and via the air heater, which recycles heat to the boiler via the incoming combustion air, bringing the boiler flue gas temperature down sufficiently to achieve satisfactory boiler efficiency (90–94%, LHV basis).

A broad order-of-magnitude split of heat duties within a single reheate subcritical PCC power plant unit would be around 8% in the economiser, 35–45% evaporator, 30–40% in the superheater elements and 15% in the reheater elements. Such a division of heats is less meaningful for a supercritical boiler as evaporation is not a distinct process: there is no step by step from the steam drum to the reheater drums for water/steam separation at the (subcritical) conditions of reduced load. Small separator vessels and a recirculating pump are in any case normally provided for start-up (Siemens, 2001).

A typical final feedwater temperature for a subcritical plant with 18 MPa main steam pressure would be 260ºC. Supercritical plants can have final feedwater temperatures of up to 300ºC, for example at Nordjyllandsværket Unit 3 it is 298ºC (Jensen, 2000). Feedwater heating is achieved by use of steam extracted from the turbine at appropriate points to feed indirect heat exchangers. A deaerator to keep the oxygen content of the boiler feedwater low to avoid corrosion also heats the water by direct contact. The supercritical steam cycle design in Figure 16 uses four surface type (indirect) low pressure heaters, three surface type high pressure heaters and a deaerator (Welford and others, 2002). Subcritical cycles will have fewer feedwater heaters. Pumping is effected in two stages: a low pressure extraction pump pumps the condensate at around 0.5–1 MPa from the condenser through the LP heaters and deaerator, then the main boiler feed pump raises the pressure to boiler pressure and sends the feedwater through the HP feedwater heaters to the input of the boiler economiser. A third pump is sometimes used after the deaerator to raise the water temperature to an intermediate level for passage through the first two high pressure feedwater heaters, with the main feed pump pressurising the water in advance of the last high pressure heater. Thus, only one of these heaters then needs to be designed to withstand the highest pressures.

Reliability and cost can be issues with feedwater heaters and the balance of advantages and disadvantages normally lies with keeping to no more than eight heaters plus the deaerator for large steam turbine units. Depending on the pressures used in the cycle, the final high pressure feedwater heater may take for heating duty steam extracted from or after the high pressure turbine. The heaters are cascaded such that the hotter drains are added to the steam side of the next heater. Depending on the adopted design conditions, uncondensed steam may sometimes be taken from one or two of the highest pressure surface type heaters to feed other HP heaters, thereby reducing the number of HP turbine steam extraction points required. For feedwater heaters, there is a need to achieve minimum terminal temperature differences (pinches) with adequate heat transfer rates to minimise the size and cost of the heaters. Feedwater outlet temperatures are usually close to the bleed steam saturation temperature, especially for HP heaters. In some designs, to achieve a high final feedwater temperature, the feedwater in the last HP heater may be heated to above the bleed steam saturation temperature, but such heaters are more expensive. An alternative is to use higher pressure bleed steam. Drains from the condensed steam may typically be designed to be around 5ºC warmer than the inlet feedwater’s temperature.
Figure 16  Supercritical cycle showing feedwater heating arrangements (Welford and others, 2002)
Feedwater heating effectively reduces the heat addition to the water in the boiler, raising the efficiency, but reduces the power output per kg of main steam production, so increasing the specific steam consumption. Monitoring and keeping to design conditions at inlets and outlets of the heaters and to feedwater inlet and outlet conditions across the deaerator is important for maintaining highest efficiency.

Surface type feedwater heaters are usually of shell and tube construction. Designs in which the feedwater tubes are welded to tubular inlet and outlet feedwater headers inside the steam shell offer advantages over those employing flat tubesheds that have to be much thicker for strength. These include reduced thermal stress-induced failures from load cycling, reduced maintenance requirements and better reliability. The carbon steel tubes used in header heaters do have to be very carefully welded because of their complex shape (Armor and others, 2003).

**Combined heat and power production**

Where heat or steam is exported from the plant in addition to electricity, significant changes occur in the feedwater heating system flows. In district heating systems taking hot water, part of the bleed steam goes to the district heating scheme heaters, with the condensate from these re-entering the low pressure feedwater stream at different points along the LP feedwater heating train according to their temperature. The principle is illustrated in Figure 17.

If steam is exported, there may be no return of condensate, depending on whether the steam is intended for process use or for supplying heat exchangers. Boiler feedwater consumption is then increased. Depending on the required steam pressure, significant steam export may be best achieved in back-pressure turbine systems, which exhaust above ambient pressures.

### 4.3.3 Influencing parameters

This section looks at the influences of some of the more important parameters on steam cycle performance. Station thermal efficiency directly affects fuel costs and has a major influence on emissions. It is therefore probably the most important design target after capital cost and always forms part of the guaranteed performance specification for a new unit. Unit efficiency is determined by factors that can be grouped into those appertaining to the efficiency of the heat engine cycle, those related to the extent of boiler heat recovery for that cycle, and other factors such as auxiliary power consumption and transformer losses. Auxiliary power consumption can actually have a bearing on the thermodynamic cycle design, although indirectly (thus, the main boiler feedwater pump may be driven by a steam turbine or use an electric drive). The text below focuses mostly on the effects of heat engine cycle parameters.

**Main steam conditions**

Chapter 3 showed that the upper (main) steam turbine temperature is among the most significant determining factors for steam cycle efficiency, and therefore among the first parameters to be decided upon in designing a boiler/turbine generating unit. Figure 18 shows steam cycle heat rate improvements as main steam pressure and main steam and reheat steam temperatures are increased from subcritical to supercritical conditions (Logan and Nah, 2002). The adjacent curves show the influences of individual changes in the main steam and reheat steam temperatures. These data are for single reheat cycles. It can be seen that the effect of changing reheat temperature is virtually linear, at 0.03% heat rate improvement per degree Celsius, and the effect is perhaps not surprisingly almost independent of main steam pressure. The gain from increasing the main steam temperature decreases at higher temperatures.

![Figure 17 Integration of hot water production for district heating scheme into LP feedwater heating train](image-url)
Heat rate improvement, %
Increase of net efficiency, %
1
3
4
6
7
8
15
20 25 30 35
Rated main steam pressure, MPa
(figures on curve are main and reheat steam temperatures °C)

Figure 18 Heat rate improvement of single reheat steam turbine with main steam pressure (Logan and Nah, 2002)

Figure 19 Effect of steam conditions on plant efficiency (Siemens, 2004)

The difference in net plant efficiency between a 25 MPa Benson (once-through) boiler based system and a drum boiler at 16.7 MPa, both using 538°C main and reheat steam temperature, is shown as over three percentage points in Figure 19 (Siemens, 2004). Since increasing the pressure also increases the power requirement of the main boiler feed pump, the gain is remarkable. The effects of main and reheat steam temperatures at fixed pressure are also shown on the figure up to 580°C/600°C.

There is an accompanying gain in that higher temperatures allow higher pressure ratios before the degree of wetness at the turbine outlet becomes too high (>10%). There are practical limits to achievable temperatures, with long-term creep-resistance under the stresses of normal operation and resistance to thermal strain during start-up and shut-down being the important determinants. Ferritic/martensitic alloys in state-of-the-art boiler and turbine components currently allow temperatures of 600°C to be used. Table 1 shows the steam conditions and overall efficiencies of some recent PCC units (Bugge and others, 2003; Kjær, 2003; Luby and Susta, 2002). There are some discrepancies between some of the data from the different sources, and ranges are given where the differences are quite large.

Increases in main steam pressure lead to an increase in evaporation temperature and reduced heat of vaporisation. This leads to better utilisation of the high temperature heat and a consequent increase in cycle efficiency. Figure 18 (Logan and Nah, 2002) shows that the gain in efficiency is independent of the adopted temperatures but is less at higher pressures.

Achieving a satisfactorily low exhaust wetness will in practice limit the upper pressure, although higher main steam pressures can allow scope for two stages of reheat to get around this while maximising the advantage of low temperature cooling water availability, as at Skærbækverket and Nordjyllandsværket in Denmark (Luby and Susta, 2002). The additional stage of reheat has been quantified as contributing a two percentage point efficiency improvement (Scheffknecht and others, 2003) but there is an associated capital cost penalty. Double reheat reduces the specific steam consumption by around 15% (Kjær, 1993).

The effect of cold reheat pressure on turbine cycle heat rate is shown in Figure 20 (Welford and others, 2002).

Cooling water
Design cooling water temperature has a strong effect, as it determines the vacuum achievable at the turbine outlet and so strongly affects the degree of expansion and final volumetric flow of the steam. For example, a halving of turbine exhaust pressure approximately doubles the volumetric flow.

Changing the design condenser pressure from 0.005 MPa to 0.003 MPa increases the efficiency of a PCC power plant by around 1 percentage point. For an approach temperature in the condenser of 12°C, a change in condenser pressure from 0.005 MPa to 0.003 MPa corresponds to a change in cooling water inlet temperature from 21°C to 12°C. Since location is the prime determinant of achievable condenser conditions, a trade-off may exist between marginal costs of coal transportation versus marginal fuel costs in determining where to site a plant. Because the size of the last LP stage is strongly affected by exhaust pressure, choice of cooling water design temperature can have major implications for the cost of the LP turbine, so a trade-off exists also between capital cost and achievable efficiency.

A reduction in cooling water temperature from 20°C to 15°C at an existing plant (the Maasvlakte 540 MW coal-fired plant in the Netherlands) reduced heat rate by almost 1% (Kromhaut and others, 2001). Between 15°C and 10°C, it decreased by 0.5%. The capability of existing plants to take advantage of reduced cooling water temperatures may of course be limited by the volumetric flow rate capability of the installed LP turbine.

In a given plant, cooling water flow rate will also influence efficiency by reducing the cooling effect in the condenser, so reducing condenser vacuum and power output. However, this
may be compensated to an extent by the reduced pumping power so that an optimum cooling water flow rate may in practice exist. At the Maasvlakte plant in the Netherlands, thermodynamic modelling showed that there was a different optimum flow rate depending on the temperature of the cooling water (which varies at the plant’s location from 1°C to 22°C).
to 20°C). At 4°C, the optimum flow rate was 12.2 m/s, while at 20°C it was 18.8 m/s. Results from the modelling were confirmed by actual plant data (Kromhaut and others, 2001).

**Final feedwater temperature and feedwater heating circuit**

Increasing boiler feedwater temperature effectively increases the upper temperature of the thermodynamic cycle, raising the efficiency. Welford and others (2002) cite 0.15 percentage points efficiency rise for the whole unit through an increase of 20°C in final feedwater temperature. Feedwater heating was discussed in detail in Section 4.3.2.

**Other steam cycle influences**

The flow of steam lost to sootblowing needs to be minimised both to keep tube erosion under control and to reduce generation and efficiency loss from taking the steam out of the steam turbine pathway. The steam used (up to 0.5% of main steam flow) is normally from intermediate superheater or reheater headers but it is taken only intermittently. Boiler blowdown on drum boilers to control water quality will also result in some loss of heat from the cycle. Once-through boiler systems do not have an inventory of recirculating water, so blowdown is not possible, and a condensate polishing system of high reliability is essential. This does however reduce energy losses, total make-up water requirements and waste water quantities (Luby, 2003).

**Boiler heat recovery**

The heat for the steam cycle has to be extracted from the chemical energy in the coal first. Large state-of-the-art PCC boilers have percentage efficiencies in the low-mid 90s, depending on the air heater exit temperature on the flue gas side. For example, the design boiler efficiency at Niederaussem – K is 94.4% (Luby and Susta, 2002) on an LHV basis of heat input. There is not usually any need to debate how far to go in heat recovery as it is almost always cost effective to maximise it. Combustion efficiency may be typically 99.5% on design fuel so losses from this quarter are small. The remaining lost energy is in the form of heat, partly through radiation and air leakage across the air heater, but mainly as sensible and latent heat in the flue gases. It is comparatively easy to achieve efficiencies over 90% using rotary air heaters with flue gas temperatures at their exit of about 100–120°C. A reduction in air heater exit temperature by 20°C increases thermal efficiency by around 0.5 percentage points but with most coals, to avoid acid dew point corrosion, the gases cannot be allowed to cool below 120°C before desulphurisation. After flue gas desulphurisation (FGD), stack gases at temperatures of about 80°C are normal, and the latter is around the minimum required for plume buoyancy to ensure dispersal (most efficiently achieved using FGD inlet/outlet gases heat exchange). It appears unlikely that further large gains in the efficiency of heat recovery from state-of-the-art PCC boilers can occur, although throughputs are so large that on-going small improvements will continue to give worthwhile fuel cost savings.

An energy flow (Sankey) diagram for a PCC plant with an efficiency of 40% is shown in Figure 21. The basis of the efficiency (whether the fuel input energy is expressed as higher or lower heating value) is not stated in the source, but the diagram illustrates well the pattern of heat flows. Around half of the input energy is discharged at the condenser.

**Figure 21 Energy flow diagram for a PCC plant**

(Tsuji, 2004)

**Flexibility**

In many OECD countries, where utilities have been deregulated, flexibility to generate efficiently during periods of rapid load change has become more frequently specified, even for large units. An example of large, flexible PCC units is at Schwarze Pumpe in Germany, where two 800 MWe supercritical lignite fired PCC units have been designed to operate at sliding pressure to down to 40% output, with high rates of change, and short hot start-up times. Achieving dynamic control covers the whole plant, including firing and emissions control as well as the steam cycle. A 6% load change per minute is achievable at Schwarze Pumpe with only minor effects on the main steam and hot reheat steam temperatures without turbine throttling and without using the storage capacity of the boiler. Very rapid ramping (for example, by 50 MWe in seconds) requires use of a degree of steam reserve and temporary changes in feedwater heating (temporarily diverting steam away from feedwater heating so that more passes through later turbine stages will have a very short term negative effect on efficiency, but this matters less than meeting demand over these transient periods). If the feed pump turbine trips, a back-up auxiliary feed pump is started and firing capacity is rapidly reduced to prevent the furnace tubing temperature from rising excessively (Kirmse and others, 2000).

**Other influences**

Auxiliary power demand is an important determinant of station overall efficiency though mainly not through thermodynamic effects. Most equipment areas on a PCC plant consume some power. They include the boiler feed pump (when electrically driven), condensate and other feedwater pumps, fans, cooling water pumps, flue gas cleaning systems, coal grinding, coal reception and crushing, ash handling and limestone and gypsum handling systems. A modern unit with an electrically driven feed pump has a total
auxiliary power demand of around 6% of gross generation, excluding the power consumption of an FGD plant. The power demand of draught plant and pulverisers can be as much as 2% of gross generation (Juniper and Pohl, 1996), the boiler feed pump consumes around 15 MWe on a 600 MWe plant, or around 2.5% of gross power (Goodall, 1981), while balance of auxiliaries account for the remainder. An FGD plant uses around 1.5% of gross power.

Where applicable, variable speed drives, preferably based on variable speed electric motors, are used to reduce losses and minimise power consumption, for example on feed pump drives that otherwise can take excessive power if throttled at part load. Alternatively, the main boiler feed pump may be provided with a turbine for its motive power, as at the Schwarze Pumpe plant in Germany (Luby and Susta, 2002). The pump can therefore be operated at variable speed using this system also, giving an alternative way to achieve economy. With feed pump turbines, the auxiliary power demand is in any case considerably reduced, but gross generation is lower since steam is taken of from the main turbine to drive the boiler feed pump turbine.

4.3.4 Materials

Because the steam turbine efficiency is strongly influenced by temperature and pressure, the drive to achieve higher efficiencies is closely connected to the development of improved materials to withstand ever more difficult conditions, particularly materials for PCC boiler components such as main steam headers and main steam and reheat steam pipework. The upper temperature limit for ferritic and martensitic steels can be stretched to above 600ºC – the COST 522 programme in the EU is looking to the development of martensitic steels for steam temperatures above 620ºC and Japanese researchers envisage temperatures up to 650ºC using ferritic steels (DTI, 2002b) – but, to progress much further, nickel based superalloys will be needed. These are already used in the manufacture of gas turbines, but, as the environment in which the materials will need to operate differs from that of current commercial applications, R&D is in progress to achieve alloys that are sufficiently resistant to gas-side high temperature corrosion and steam-side oxidation and capable of fabrication in the large component sizes required for ultrasupercritical PCC units. One of the barriers limiting the application of new materials for water-walls and thick-walled components such as high pressure headers and main steam piping is their fabrication (Chen and Scheffknecht, 2003).

Programmes are in progress in Europe and the USA to achieve these aims (Henderson, 2003a,b). Superalloys are much more expensive than the current state-of-the-art materials, but only parts of the plant need to accommodate the most extreme temperatures and pressures. The EU is supporting a major THERMIE project under the Fourth and Fifth Framework RTD Programmes aimed at reaching live steam parameters of 700ºC and 37.5 MPa (DTI, 2002a). Important within this AD700 project is the materials development programme, including components fabrication. It is expected that net thermal efficiencies of 50–52% will be realised with single reheat and cooling tower cooling (3.5 kPa condenser pressure) and 54–55% with double reheat and sea water cooling in northern Europe incorporating reheat at 7.5 MPa and 720ºC (Bugge and others, 2003; Kjær, 2003). Figure 22 shows how the use of superalloys is expected to move plant thermal efficiencies forward up to 2020 in Denmark. The superalloys for long-term operation at 700–720ºC will be developed for use in thin-walled superheater and reheater tubes, thick-walled outlet headers and steam piping. New austenitic materials will also be developed for boiler tubes operating in the 600–700ºC temperature range. The first commercial plants using the technology are expected during the second decade of this century. Longer term targets are for net efficiencies above 55% after 2020, using upper steam temperatures of 800ºC (Bugge and others, 2003). A design for a horizontal supercritical boiler of reduced height has also been produced to enable the quantity of expensive high temperature pipework to be reduced (Smith, 2000; Scott, 2001; Welford and others, 2002).

In the USA, materials technologies for ultrasupercritical conditions of 760ºC at 37.9 MPa are being developed and materials for steam temperatures up to 870ºC are even being considered. Nickel alloys will be considered also for superheater and reheater tubing. Efficiencies of 46% and 48% (presumed HHV basis) have been cited for cycles based on single reheat and double reheat for upper temperatures of 760ºC (Schimmoller, 2003). Further information on developments in materials for ultrasupercritical boilers are in a recent report by the Clean Coal Centre (Moreea-Taha, 2002) and the materials have also been reviewed by Viswanathan and others (2002). There are also descriptions of the materials envisaged for the various boiler components and pipework by Bugge and others (2003) and Chen and Scheffknecht (2003).

Advanced materials are also being developed for turbines. In fact, it is arguable that available turbines rather than boiler materials are limiting conditions (Welford and others, 2002). The design of high temperature turbines is strongly
influenced by the developments of improved materials and by the use of more effective cooling steam arrangements for critical components. Table 2 shows materials used by MHI for high pressure parts in steam turbines for state-of-the-art supercritical compared with 538°C subcritical turbines (Luby and Susta, 2002). High strength martensitic stainless steel casting alloys for valve bodies and nozzle boxes for 593°C turbines are used by GE. Supercritical plants generally have high crossover temperatures and so the low pressure turbines have had to be forged from alloys with better fracture toughness and tensile ductility properties. This provides additional freedom to optimise cycle parameters while dispensing with steam cooling, to achieve higher efficiency levels (Logan and Nah, 2002). Armor and others (2003) have outlined the technology and materials advances made in supercritical turbines and the requirements for the future. Flexibility and robustness to rapid ramping were highlighted as areas to address closely.

700°C technology for turbines will also be based on nickel alloys for the hotter parts and new designs are to be produced under the European AD700 project to minimise the proportion of these materials that is needed (Bugge and others, 2003). Table 3 shows information on the composition and status of these materials (Bauer and others, 2003). Although related materials are in use in advanced gas turbines already, large prototype steam turbine components made from the alloys require further testing and demonstration, particularly to establish improved creep strength; steam oxidation could also develop as an issue (DTI, 2002b).

The cost of the nickel alloys needed for the turbines and boilers handling 700°C steam is said to have decreased to around ten times that of the cost of the steels used for headers in current state-of-the-art supercritical plants. Means of keeping the cost under control include use of a horizontal furnace design plus raising of the turbine hall floor and new turbine designs in which the high pressure turbine is split into two turbines, only the first of which, containing nickel alloy components, would take the 700°C. The use of a single intermediate pressure turbine taking reheat steam, and so containing nickel alloy materials, could also serve to reduce the cost (Smith, 2000). Advanced materials of lower density based on titanium for last stage turbine blades have been developed to allow longer blade lengths and so increase the exhaust area.

### Table 2 Materials used by MHI for state-of-the-art-supercritical steam turbines and 538°C subcritical turbines (Luby and Susta, 2002)

<table>
<thead>
<tr>
<th>Component</th>
<th>600°C</th>
<th>538°C</th>
</tr>
</thead>
<tbody>
<tr>
<td>Rotor</td>
<td>New 12 Cr forging</td>
<td>Cr-Mo-V forging</td>
</tr>
<tr>
<td>Nozzle chamber</td>
<td>12 Cr cast steel</td>
<td>2¼ Cr – 1 Mo cast steel</td>
</tr>
<tr>
<td>Inner casing</td>
<td>12 Cr cast steel</td>
<td>1¼ Cr – ½ Mo cast steel</td>
</tr>
<tr>
<td>No. 1 blade ring</td>
<td>12 Cr cast steel</td>
<td>1¼ Cr – ½ Mo cast steel</td>
</tr>
<tr>
<td>No. 2 blade ring</td>
<td>2¼ Cr – Mo cast steel</td>
<td>½ Cr – ½ Mo cast steel</td>
</tr>
<tr>
<td>Outer casing</td>
<td>2¼ Cr – Mo cast steel</td>
<td>1¼ Cr – ½ Mo cast steel</td>
</tr>
<tr>
<td>Rotating blade</td>
<td>Refractory alloy (R-26)</td>
<td>12 Cr forging</td>
</tr>
<tr>
<td>Main steam stop valve</td>
<td>9 Cr forging</td>
<td>2¼ Cr – 1 Mo forging</td>
</tr>
<tr>
<td>Main steam governing valve</td>
<td>9 Cr forging</td>
<td>2¼ Cr – 1 Mo forging</td>
</tr>
</tbody>
</table>

### Table 3 Materials for 700°C steam turbines (Bauer and others, 2003)

<table>
<thead>
<tr>
<th>Composition, %</th>
<th>Application</th>
<th>Status</th>
</tr>
</thead>
<tbody>
<tr>
<td>Ni</td>
<td>Cr</td>
<td>Co</td>
</tr>
<tr>
<td>617</td>
<td>52</td>
<td>22</td>
</tr>
<tr>
<td>625</td>
<td>63.5</td>
<td>21.5</td>
</tr>
<tr>
<td>Waspalloy</td>
<td>56</td>
<td>19.5</td>
</tr>
</tbody>
</table>
The heat available to the steam cycle in a coal-fired combined cycle power station is available at various temperatures and proper use must be made of these different grades of heat for maximising thermal efficiency. In gasification-based cycles, there may be some steam requirements to be satisfied also.

### 5.1 Overall cycle configurations

In combined cycle systems, the heat from the gas cycle is taken from more than one source. A heat source common to all is the waste heat boiler (heat recovery steam generator) that is used to extract useful energy from the gas turbine exhaust gases, but also important are heats released from, for example, combustion in a PFBC system and gasification and gas cooling in an IGCC system.

Combined cycles are initially designed around the gas cycle. The choice of whether to use PFBC or IGCC is the first decision to be made. Coal characteristics will play an important role. Another is scale. PFBC is actually well-suited to low-grade (low calorific value, high ash, high sulphur) coals and waste fuels because of its combustion regime. It is also compact compared with PCC and simpler than IGCC. It is therefore suited for use in urban areas where space is at a premium and a limited capacity is required, while IGCC can offer very low sulphur dioxide (SO$_2$) emissions but usually with less flexibility on fuel type. Eight PFBC plants have been built, mostly using ABB (now ALSTOM Power) technology (Rousaki and Couch, 2000). Most are still operating. The PFBC at Karita, Japan, uses a supercritical steam cycle. To date, IGCC plants using coal as fuel have been built in no greater number than PFBCs, although there are several technology types and suppliers and interest is growing.

Figure 4 in Chapter 2 showed the configuration of PFBC. The proportion of gross power produced by expansion of the flue gas in the gas turbine is about 20% because much of the coal input energy is extracted as heat in the pressurised combustor for steam raising and also because the entry temperature to the expander is low compared with internally fired gas turbines. The fluidised bed has to operate below about 900ºC to prevent agglomeration, sintering and defluidisation. Heat obtained from cooling the turbine exhaust gas is used within the steam cycle. The effect of using the gas turbine is to increase the thermal efficiency of the plant by around three percentage points compared with a PCC plant using the same steam conditions.

The three main sources of heat for an IGCC system are in Table 4. However, there is considerable scope for variations in availability and treatment of medium and low-grade heats in gasification-based cycles because there are many alternative gasifier types and possible gas cleaning systems and environmental control systems. The type of coal to be used is an important factor in determining which gasification technology to design the configuration around. The different types of gasifier and their suitabilities for different coals are reviewed in a recent IEA Clean Coal Centre report (Collot, 2002). The proportion of gross power produced by the gas turbine in an IGCC system, at about 60%, is higher than for PFBC. This is because most of the chemical energy in the coal is removed from the gasifier as chemical energy in the product gas for combustion in the gas turbine combustor.

Table 5. Slurry-feed gasifiers have lower cold gas efficiencies than dry feed systems as part of the heat generated in the gasifier by partial combustion of the coal is absorbed as latent heat in evaporation of the slurry water. Gravitating bed gasifiers such as the BGL gasifier have high cold gas efficiencies because they are counterflow and have long reaction zone residence times. Consequently they generate little steam and are broadly neutral in steam production/demand. The raw gas temperature and composition following quench with aqueous liquor ahead of the product gas cooler is shown in Table 6 (Sander and others, 2003). It contains methane and so has a higher calorific value than the gas from entrained systems as methane is produced as the feed coal is pyrolysed as it descends towards the hottest zone of the gasifier. The steam that is produced by the process is only at low and medium pressure (former in the gas cooler, latter in the gasifier jacket). The two-stage slurry feed entrained E-Gas system as used at the Wabash River repowering project has a higher cold gas efficiency than the single stage slurry feed entrained ChevronTexaco process (now owned by GE Energy) because the second stage of fuel injection captures more of the energy by devolatilising and pyrolysing some of the second stage feed while taking heat out of the syngas produced in the first stage (US DOE, 2000, 2002b).

An example gas composition for the Shell coal gasification process is included in Table 6. Figure 23 shows the process. After a gas quench to take the temperature down from around the 1600ºC that it reaches within the gasifier itself (Hooper, 2003) the gas is at 900ºC, or ~350ºC hotter than that emerging from BGL gasification. The syngas cooler therefore raises high pressure steam. Either type of gasification system can be used as the basis of an IGCC, but

### Table 4 Principal sources of heat for an IGCC system with a syngas cooler

<table>
<thead>
<tr>
<th>Plant area</th>
<th>Heat source</th>
</tr>
</thead>
<tbody>
<tr>
<td>Gasifier cooling</td>
<td>HP or MP steam</td>
</tr>
<tr>
<td>Syngas cooler (waste heat boiler)</td>
<td>HP steam</td>
</tr>
<tr>
<td>Gas turbine waste heat boiler</td>
<td>HP, MP and LP steam</td>
</tr>
<tr>
<td>Various process areas (eg sulphur/sulphuric acid recovery plant)</td>
<td>LP or MP steam</td>
</tr>
</tbody>
</table>
the gas turbine/steam turbine power balances are different (see Table 7). The BGL based system requires all of the HP superheated steam to be produced in the gas turbine’s waste heat boiler, whereas only part of this duty needs to be performed by the waste heat boiler in the Shell cycle. 4–7 MPa MP steam is produced from the water tubes surrounding the Shell gasifier and HP steam is produced by the convective cooler (Hooper, 2003).

The Shell coal gasification process-based Buggenum plant with a Siemens V94.2 gas turbine of 1120ºC inlet temperature and the somewhat similar PRENFLO Puertollano plant with a V94.3 turbine have net efficiencies of 43% and 45%, LHV basis (Scheibner and Wolters, 2002; Hannemann and others, 2003). The Kentucky Pioneer Power Project, based on four BGL moving bed slagging gasifiers, and commencing construction in 2004 has a predicted efficiency of 40% (HHV basis) (US DOE, 2002a). Because a combustion turbine contributes much of the power, IGCC cycles discharge more of the coal input energy at the stack, so sensitivity to stack temperature is high compared with PCC. After allowing for the fact that higher stack temperatures tend to apply under US conditions, with net efficiency implications of perhaps 1–2 percentage points, the combined cycle efficiencies are surprisingly similar for all three, while the balances of energy flows within the cycles differ.

Table 5  Indicative cold gas efficiencies of some gasification processes

<table>
<thead>
<tr>
<th>Process Description</th>
<th>Efficiency</th>
</tr>
</thead>
<tbody>
<tr>
<td>Dry feed entrained (Shell, Buggenum) (Hannemann and others, 2002)</td>
<td>~80%, LHV basis</td>
</tr>
<tr>
<td>Slurry feed entrained ChevronTexaco (Polk Power) (US DOE, 2002c)</td>
<td>~70%, LHV basis 70-75%, HHV basis</td>
</tr>
<tr>
<td>Two-stage slurry feed entrained E-Gas (Wabash River) (US DOE, 2000, 2002b)</td>
<td>~80%, HHV basis</td>
</tr>
<tr>
<td>Moving bed</td>
<td>~90%, LHV basis</td>
</tr>
</tbody>
</table>

Table 6  Example crude gas temperature and volume composition, BGL and Shell gasification systems (Sander and others, 2003; Hooper, 2003; Jones, 2002)

<table>
<thead>
<tr>
<th>Temperature, ºC</th>
<th>H2, %</th>
<th>CO, %</th>
<th>CH4, %</th>
<th>CO2, N2, H2O, balance, %</th>
</tr>
</thead>
<tbody>
<tr>
<td>BGL</td>
<td>550</td>
<td>28</td>
<td>56</td>
<td>7</td>
</tr>
<tr>
<td>Shell</td>
<td>900</td>
<td>25.6</td>
<td>52.9</td>
<td>0</td>
</tr>
</tbody>
</table>

values in brackets are the data immediately above converted to a nominal dry basis

Figure 23  Shell coal gasification process (Hooper, 2003)
BGL system, although producing virtually no net steam, has a high cold gas efficiency to compensate. Gasification cycles incorporating single stage slurry feed entrained gasifiers with a water quench do not under the same conditions achieve cycle thermal efficiencies as high as those from radiant cooler designs because high pressure steam cannot be produced in the gasification/gas clean-up process area and cold gas efficiency is penalised by the evaporation of the slurry water. However, the cost of the gasification/gas clean-up process area is lower.

Table 7, showing the division between gas turbine power and steam turbine power for some existing IGCCs, is a reflection of the relative balances of fuel gas energy and HP steam production. The table also contains some predicted data taken from a study using the ASPEN process simulation system (NETL, 2000).

The IGCC cycles above use conventional gas scrubbing for ammonia and sulphur gases removal from the fuel gas as this is based on commercially proven technologies (such as use of amines or methanol to remove hydrogen sulphide (H₂S) from the gas). Pre-conversion of COS (carbonyl sulphide) to H₂S is also needed so that it can be removed. The sulphur is recovered as elemental sulphur, which has a by-product value. The gaseous contaminants are scrubbed out after final particulates removal, which usually is accomplished by cyclones and/or filters operating at around 300ºC. Gas scrubbing has to be done after heat recovery from the de-dusted or de-tarred gas, which then has to be reheated again before combustion. There is an inevitable energy penalty from the use of these features in IGCC cycles and higher temperature de-dusting and gaseous contaminants removal would allow more efficient cycle configurations. Key requirements of such cycles are reliable technologies for particulates and sulphur gases removal at around 500–600ºC. The former are not reliably proven even on combustor off-gases, while hot desulphurisation systems are at the pilot stage. New sorbents are emerging (Slimane and Williams, 2002; Slimane and others, 2002). The recent impetus to mercury emissions reduction in some countries will need to be accommodated within both cold and hot gas clean-up concepts, as a significant proportion of the mercury from the coal tends to emerge in the stack gases of current designs of IGCC.

Higher efficiency IGCC cycles will be achievable even without hot gas clean-up, using improved gas turbines and a high degree of system integration. An efficiency of 51.5% (net, LHV basis) has been predicted for a cycle using a Siemens V94.3A gas turbine (Pruschek and others, 1999; Haupt and others, 2003). It should however be noted that high degrees of integration do limit flexibility and start-up time, and a lower maximum continuous rating (MCR) efficiency cycle of less capital cost may be preferred as a lower risk investment in many circumstances.

### 5.2 Steam cycles for coal-fired combined cycle systems

In combined cycles the steam cycle configuration follows from the selection of major systems in the gas cycle, although flexibility does exist in steam cycle design. For example, combined heat and power (co-generation) plants have been constructed based on PFBC. Energy outflows from the steam cycle impact on electric power generation from the gas turbine but the comparatively small proportion produced by the gas turbine is unaffected.

Figure 24 shows the power cycle for the 360 MWe Karita supercritical 24.2 MPa/566ºC/593ºC PFBC plant (Veenhuizen and Anderson, 2000). It resembles the cycle of a once-through PCC unit and includes a steam separator for low load operation and start-up, as in other once-through systems. However, the placement of the main heat extraction surfaces is different. All the main tube banks including the superheater and reheater are located in the fluidised bed, while the economiser takes its heat from the turbine exhaust. The similarity of heat availability to PCC allows reheat to be used, although economics restrict this to larger scale units only as here at Karita. The economiser not only heats the high pressure feedwater to main boiler inlet temperature, it also performs some of the low pressure and high pressure feedwater heating in parallel with bleed steam heaters. The air heater is omitted, as the economiser thereby cools the gas turbine exit gases from around 375ºC to stack temperature (~150ºC). Combustion air from the gas turbine compressors passes through an annular tube surrounding the cleaned combustion gases to recover heat. Even for subcritical cycles, a single pressure of evaporation is invariably used as this is sufficient to exploit the high proportion of high grade (high temperature) heat.

The choice of configuration of an IGCC steam cycle will depend on the gasification/gas clean-up combination, but the
The whole integrated system can offer many choices that have implications for the steam cycle (for example, the extent of gas cooling for steam generation relative to that used in exchange with other gas streams). Figure 25 shows the reheat steam cycle for the 250 MWe Polk IGCC plant in the USA owned by Tampa Electric that uses GE’s ChevronTexaco slurry feed gasification technology with a radiant gas cooler for raising high pressure steam (US DOE, 2002c). The gas emerging from the radiant cooler is also cooled in two parallel fire tube boilers (syngas coolers). The reheat steam cycle utilises three pressures of steam (HP at 9.86 MPa, IP at 2.65 MPa and LP at 0.45 MPa). HP and MP steam from the process areas are generated at pressures of 11.48 MPa and 3.00 MPa and let down to the pressures shown above, which are in the steam drums of the gas turbine heat recovery steam generator (waste heat boiler). All main steam superheating and reheating is effected in the waste heat boiler. Net MP and HP flows are into the steam turbine. All the MP and HP steam turbine cycle feedwater heating arrangements are effected using the waste heat boiler. LP steam is extracted from the steam turbine for process use. Recirculating systems with steam drums are normally used in IGCC steam systems as there are many steam and heat demands and sources to be accommodated and means of storing steam are valuable for flexibility. However, once-through supercritical technology may eventually be adopted for the gas turbine heat recovery steam generators on future IGCC plants (Davison and others, 2003).

A cycle for a gasifier fitted with a gas quench and a subsequent sour shift reaction stage is shown in Figure 26 (O’Keefe and others, 2001). The arrangement is predicted to give plant efficiencies of over 40%, LHV basis.

Means of limiting thermal NOx production in the gas turbine
in IGCC systems employ control of the flame temperature using water saturation or nitrogen addition to the fuel gas. If nitrogen is used, this is available from the air separation unit supplying oxygen to the gasification process area. If saturation with water is used, heat is extracted from the steam cycle to heat the fuel gas. Water is more effective for NOx reduction than nitrogen. The energy for heating the fuel gas is not all lost as it is exploited within the combined cycle. At
the Buggenum, Puertollano and, more recently, the Tampa Electric Polk plant, both nitrogen and water saturation are used together (McDaniel and Hornick, 2003; Hannemann and others, 2003).

Heat recovery boilers on gas turbines are sometimes fitted with facilities (duct burners) for direct firing with supplementary fuel to reduce heat transfer surface area requirements or to allow higher steam cycle upper temperatures at reduced gas turbine load. Such a system is unlikely to be worthwhile on IGCC systems because of the additional complication. However, it is common to use natural gas or distillate fuel as a back-up fuel for the gas turbine to maintain availability during gasification or syngas clean-up system outages.

5.3 Design aspects and influence of parameters

5.3.1 PFBC

Carbon utilisation
Carbon utilisation is generally very high in PFBCs, with their efficient contacting of coal and air – for example, 99.5% was obtained at the Tidd PFBC demonstration in the USA (US DOE, 2002a).

Gas turbine
In conventional PFBC, there is limited scope for increasing the efficiency through gas turbine developments since the turbine inlet temperature is set by the fluidised bed combustion temperature, which is limited to below ~900°C to avoid fluidisation problems. One way around this is to use hybrid cycles containing both coal gasification and combustion systems. These aim at realising complementary advantages of each type of system, including exploiting high inlet temperature gas turbines. Many configurations are possible, including feeding PFBC off-gas to a fluidised bed gasifier or burning syngas in PFBC off-gases (deeply cleaned to avoid turbine blade damage by hot contaminants) before expansion through the turbine. Activities are under way in Japan, Europe, and the USA. The A-PFBC process in Japan is an example (Saito and Harada, 2003; CCUJ, 2002). Such hybrid processes are still developmental (Henderson, 2003a).

Steam cycle
The potential gain from higher steam conditions and other steam cycle improvements in PFBC is similar to that discussed for PCC plants in Section 4.3.3, as 80% of the gross power developed is from the steam turbine, with most of the heat for that coming from the coal PFBC combustor. The use of higher steam conditions in the steam cycle of a combined cycle plant has been applied to the Karita 360 MWe PFBC in Japan, which uses a 24.1 MPa/566ºC/566ºC supercritical cycle to achieve a gross efficiency of 42%, HHV basis (estimated to be equivalent to around 44%, LHV basis).

Hot gas filtration
PFBC cycles should gain higher efficiencies from the use of filtration systems rather than cyclones alone because the pressure drop with such systems is less than given by cyclones. They would also give improved reliability of the gas turbine, lower running costs and better environmental performance. However, reliable, fail-safe filtration is essential to realise these benefits. Tests have been carried out on the Escatron plant in Europe, the Wakamatsu PFBC in Japan and at Tidd in the USA (Henderson, 2003a). The reliability of filtration for PFBC systems is not universally accepted as demonstrated.

Auxiliary power demand
Minimising auxiliary power demand is as important in combined cycles as in PCC systems. As an FGD plant is not required for SO₂ removal (the latter is accomplished by...
limestone addition to the combustor), auxiliary power consumption is lower than for PCC. An energy flow diagram for the Karita PFBC is shown in Figure 27 (Tsuji, 2004). Auxiliary power demand (house load), at only 2.3% of gross power, is no more than half of that for PCC. NOx emissions are inherently lower than for PCC, so an SCR (selective catalytic reduction) unit is also not required for most locations, saving a small amount of further power.

5.3.2 IGCC

Gasification technology

The selection of technology depends strongly on the characteristics of the coal for which the system is to be designed. Gasifiers available for IGCC fall into three general types:

- **Entrained bed** (such as ChevronTexaco slurry feed, now owned by GE; Shell dry feed);
- **Fixed (moving) bed** (such as Lurgi and BGL; former British Gas interests in the latter are now owned by Advantica);
- **Fluidised bed** (such as KRW, IGT, HTW and transport gasifiers).

Gasifiers may be designed for ash slagging or they may be non-slagging. Table 8 shows the general suitability of the gasifier types for different coals. Coals with very high silica contents in their mineral matter or with very high ash contents are probably not well suited to use in slagging gasification systems as they are likely to at least suffer an efficiency penalty and at worst not produce a freely flowing slag film on the gasifier wall and cause blockages at the slag outlet. Coals with large quantities of fines or with strongly caking properties are less suited to gravitating bed systems as these cannot be fed easily. Coals with very high moisture contents, can also be difficult to slurry for feeding to slurry fed gasifiers. The suitability of the various types of gasifier for different coals have been reviewed in detail in a separate IEA Clean Coal Centre report (Collot, 2002). The effect on the thermodynamic cycles are the concern here, but the information in Sections 5.1 and 5.2 indicates that the net overall effect on thermal efficiency need not be overriding.

### Table 8 General suitability of gasifier characteristics to different coals

<table>
<thead>
<tr>
<th>Coal property</th>
<th>Gasifier types generally best suited</th>
</tr>
</thead>
<tbody>
<tr>
<td>Large size</td>
<td>Gravitating bed, fluidised bed</td>
</tr>
<tr>
<td>High fines content</td>
<td>Entrained</td>
</tr>
<tr>
<td>Caking</td>
<td>Entrained</td>
</tr>
<tr>
<td>High moisture</td>
<td>Dry feed entrained but drying</td>
</tr>
<tr>
<td>High ash fusion temperatures</td>
<td>Non-slagging</td>
</tr>
<tr>
<td>High ash content</td>
<td>Non-slagging</td>
</tr>
</tbody>
</table>

**Carbon utilisation and cold gas efficiency**

Conversion is also usually about 98% or higher in slagging
gasifiers. As an example, the 150 tpd entrained gasifier in the EAGLE pilot plant in Japan has a carbon utilisation of 99% (Suzuki, 2003). Fluidised bed gasifiers usually exhibit carbon conversion levels no higher than 95% and so they would be used within hybrid gasification/combustion cycles. The residual carbon in the ash taken from the gasifier would be utilised in a combustor. Such cycles have not yet been built (Henderson, 2003a).

The cold gas efficiency of a gasification process is the proportion of the input coal energy that emerges as chemical energy of the fuel gas stream. For a given carbon utilisation, a lower cold gas efficiency implies that more of the coal’s chemical energy is retained as heat, which must be captured efficiently and used within the steam cycle. The net effect was illustrated in the examples in Section 5.1, which showed that good design practice results in similar overall IGCC performance between dry feed technologies of different cold gas efficiencies. When slurry feeding is used for the gasifier, some of this heat is lost because it is trapped as latent heat as the slurry water evaporates. It is then difficult to realise a power generation efficiency that is as high as for dry feeding, although the net effect need not be large. Slurry feed gasifiers are also offered with a water quench for the raw gas in place of use of a radiant gas cooler and, although this will further penalise ‘straight’ IGCC efficiency, it could reduce the efficiency penalty in CO2 capture configurations.

Gas turbine
In an IGCC plant, the gas turbine type is a major determining factor on overall efficiency because it contributes about 60% of the gross power produced by the plant and determines the upper temperature of the heat engine cycle. Figure 28 shows an indicative Sankey diagram for IGCC. Gas turbines for syngas firing are currently offered up to the F-class, with turbine inlet temperatures of 1350–1400°C. Currently, the most efficient IGCC systems have an efficiency of around 45%, LHV basis. The anticipated future upward trend in efficiencies of IGCC cycles is largely based on expected advances in gas turbines, as Table 9 shows. This table also shows estimated further incremental enhancements to the future IGCC efficiency of 51.5% predicted by European workers in the field that was referred to in Section 5.1 and that will also involve continuing improvements in a number of other areas (Pruschek and others, 1999; Pruschek, 2000; Haupt and others, 2003).

The gas turbine advances will further exploit the use of compressor intercooling, the use of reheat, new materials and cooling systems to permit still higher entry temperatures, initially on natural gas-fired versions but flowing through later to syngas firing technology. ALSTOM Power’s GT26B machine, as installed for example at the Enfield natural gas fired combined cycle plant in the UK, already employs reheat (Modern Power Systems, 2000). In general, it will be most efficient and economic to choose the most advanced turbine that can be supplied with full commercial guarantees based on the expected syngas quality. The first H-class gas turbine with a higher firing temperature than the current F-turbines and employing steam cooled blades is currently being tested on natural gas. H-class turbines could increase IGCC efficiency by between 1.3 and 1.4 percentage points. Together with the other gas turbine improvements, they may give ultimate IGCC efficiency improvements of 3–6 percentage points (Davison and others, 2003; Pruschek, 2000).
Syngas temperature at gas turbine combustor inlet

Pruschek and others (1999) have calculated the effect of this parameter on thermal efficiency. Figure 29 shows their results. The efficiency is increased as the syngas temperature is raised because of the direct utilisation of the sensible heat of the gas within the combined cycle.

Steam cycle

As in PCC plants, there is a drive to increase steam conditions in natural gas combined cycle plants, as turbine exhaust temperatures of gas turbines rise with new designs (Franke and others, 2000). GE have introduced steam turbines for their 1370ºC firing temperature F-class gas turbine combined cycles with main steam conditions of 16.5 MPa/565ºC (Modern Power Systems, 2002). This pressure is higher than that of the HP steam from gasifiers (around 11 MPa), presenting something of an integration challenge.

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### Table 9  Estimated further incremental improvements to IGCC efficiencies (Pruschek, 2000)

<table>
<thead>
<tr>
<th>Option</th>
<th>Key parameters</th>
<th>Efficiency increase, % points</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Staged gasification</strong></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Combustion+reduction zone (fuel staging)</td>
<td>Raw gas temperature</td>
<td>1100°C 90.1% LHV</td>
</tr>
<tr>
<td>Reduced raw gas temperature</td>
<td>Cold gas efficiency</td>
<td></td>
</tr>
<tr>
<td>Higher cold gas efficiency</td>
<td></td>
<td></td>
</tr>
<tr>
<td><strong>Chemical raw gas quench</strong></td>
<td>Best quench medium</td>
<td>CH₄</td>
</tr>
<tr>
<td>Alternative medium injected into raw gas to</td>
<td>Temperature after quench</td>
<td>1113ºC</td>
</tr>
<tr>
<td>reduce temperature and increase LHV by</td>
<td>Raw gas LHV</td>
<td>13.1 MJ/kg</td>
</tr>
<tr>
<td>chemical reaction</td>
<td>Quench gas LHV</td>
<td>14.1 MJ/kg</td>
</tr>
<tr>
<td>Considered media: H₂O, CO₂, CH₄, C₂H₆ and</td>
<td></td>
<td></td>
</tr>
<tr>
<td>clean syngas</td>
<td></td>
<td></td>
</tr>
<tr>
<td><strong>Dry high temperature syngas cleaning</strong></td>
<td>Adsorption temperature</td>
<td>500–650ºC</td>
</tr>
<tr>
<td>Dry adsorption processes for contaminants</td>
<td>Fuel gas temperature</td>
<td>375ºC / 500ºC</td>
</tr>
<tr>
<td>No gas cooling so no H₂O condensation</td>
<td></td>
<td></td>
</tr>
<tr>
<td>No process steam, lower pressure drop</td>
<td></td>
<td></td>
</tr>
<tr>
<td><strong>Gas turbines with enhanced TIT</strong></td>
<td>TIT (ISO)</td>
<td>1400ºC</td>
</tr>
<tr>
<td>Higher specific power output</td>
<td>Pressure ratio</td>
<td>30 MPa</td>
</tr>
<tr>
<td>Adapted compressor pressure ratio</td>
<td>GT exhaust gas temperature</td>
<td>633ºC</td>
</tr>
<tr>
<td>Higher exhaust temperature enables</td>
<td>Live steam temperature</td>
<td>600ºC</td>
</tr>
<tr>
<td>improved steam cycle</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Higher NOx and cooling flow</td>
<td></td>
<td></td>
</tr>
<tr>
<td><strong>Reheat gas turbine</strong></td>
<td>TIT (ISO)</td>
<td>1200ºC / 1400ºC</td>
</tr>
<tr>
<td>2nd combustion chamber</td>
<td>Pressure ratio</td>
<td>41 / 53 MPa</td>
</tr>
<tr>
<td>Further increased pressure ratio</td>
<td>GT exhaust gas temperature</td>
<td>604ºC / 641ºC</td>
</tr>
<tr>
<td>Higher exhaust gas temperature</td>
<td>Live steam temperature</td>
<td>584ºC / 600ºC</td>
</tr>
<tr>
<td>Higher specific power output</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Reheat pressure adapted</td>
<td></td>
<td></td>
</tr>
<tr>
<td><strong>Supercritical bottoming steam cycle</strong></td>
<td>Non-reheat GT: Live steam pressure</td>
<td>25 MPa 570ºC</td>
</tr>
<tr>
<td>HRSG with once-through boiler for supercritical</td>
<td>Live steam temperature</td>
<td></td>
</tr>
<tr>
<td>steam and forced circulation for IP and LP</td>
<td>Reheat GT: GT exhaust gas temperature</td>
<td>641ºC 30 MPa</td>
</tr>
<tr>
<td>steam</td>
<td>Live steam pressure</td>
<td>620ºC</td>
</tr>
<tr>
<td>Syngas cooler for reheating IP steam</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Steam turbine efficiency drops with live steam pressure</td>
<td></td>
<td></td>
</tr>
<tr>
<td><strong>Syngas adapted gas turbine</strong></td>
<td>Compressor mass flow:</td>
<td></td>
</tr>
<tr>
<td>Non-integrated ASU concept</td>
<td>N₂ from ASU admixed</td>
<td></td>
</tr>
<tr>
<td>Reduced compressor size</td>
<td>N₂ from ASU not admixed</td>
<td></td>
</tr>
<tr>
<td></td>
<td>Compressor mass flow:</td>
<td></td>
</tr>
<tr>
<td></td>
<td>-4.7%</td>
<td></td>
</tr>
<tr>
<td></td>
<td>-15% (compared with V94.3A)</td>
<td></td>
</tr>
</tbody>
</table>

Advantages concerning operability

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cycles with future gas turbines with their higher exhaust temperatures. For example, Alstom’s existing GT26B gas turbine exhausts at 640ºC (Modern Power Systems, 2000). Re-balancing of the use of the available heat sources from the gasifier/gas clean-up system would also have to be carefully judged to avoid negating any gains from the use of higher steam pressures.

Hot gas cleaning
The application of hot gas cleaning to IGCC was discussed in Section 5.1.

Auxiliary power demand
Auxiliary power consumption varies greatly between different IGCC installations, partly because of the degree of oxygen plant integration employed. The cryogenic air separation unit can take over 50% of the power demand. Total in-plant consumption is typically 10–12% of gross power, for example at the IGCCs with integrated oxygen plant air supply at Buggenum and Puertollano (de Winter, 1992; Méndez-Vigo, 2004). IGCCs with non-integrated oxygen plant tend to have have higher power requirements, for example the Tampa Electric Polk plant uses over 20% of gross power (US DOE, 2002c), although another non-integrated configuration has been predicted to have an in-plant power demand equal to only 12.4% of gross power (O’Keefe and others, 2001). The adoption of ion transport membrane technology (described in Chapter 6) would reduce the power consumption of the oxygen plant by more than a third and raise IGCC efficiencies by around one percentage point while reducing specific capital cost (Armstrong and others, 2003). Other oxygen stream consuming cycles, such as oxy-coal combustion (also described in Chapter 6), would also benefit.

Other equipment areas or systems of significant power demand include coal reception and crushing, coal grinding and/or slurrying, pressurising systems, feed pumps, fans, cooling water pumps, syngas cleaning systems, by-product plants and slag handling.

Other aspects
Gasification pressure is an important parameter. Increasing it increases the heat recovery by the syngas stream and reduces equipment size. However, more energy is needed for pressurising the feed coal and oxygen, and this has to be recovered using a fuel gas expander ahead of combustion in the gas turbine. The 3–4 MPa pressure of dry feed entrained gasifiers appears to lie near the optimum for IGCC efficiency for such systems (Davison and others, 2003). Higher pressures are used in slurry feed gasifiers as pressurising of the feed is simpler to achieve.

Ramping rates of IGCC plants are less than for PCC (or PFBC), especially for highly integrated designs. This is because the various process areas must be kept in balance as load changes. The highly integrated 250 MWe Nuon IGCC at Buggenum has achieved a maximum ramping rate of 3.5 MWe per minute (Wolters, 2003). As has been discussed earlier, much greater ramping rates (of the order of 6% per minute) are readily achievable on once-through PCC units.

5.4 Materials in combined cycle plants
The reliability and availability of gasification combined cycle plants are currently limited by materials considerations. A spare gasifier is currently necessary to achieve availabilities of 90% (Holt and others, 2003). Deterioration of refractory linings frequently occurs after initial slag penetration, but methods of preventing this using phosphate-modified materials are being developed (Dogan and others, 2002).
Another difficulty is that shell-boiler type gas coolers used to cool the raw product gas prior to cleaning can be subject to erosion from ash or char particles as well as damage from deposition leading to excessive temperatures and tube failure. Acid-induced corrosion can arise under the deposits during down time. High temperature corrosion is also an issue, with acidic species such as H₂S and HCl (hydrogen chloride) present and improved corrosion resistant materials are needed (DTI, 2002b).

Much of the expected future progression in the efficiency of IGCC cycles will come from the use of new advanced gas turbines and this was discussed in Section 5.3 and included in Table 9. Developments on these will include the use of new materials and single crystal blading (Jannson, 2003).

PFBC systems on the whole involve less aggressive environments than IGCC. For example, reducing conditions are not encountered so corrosion presents problems that are little different from those in conventional PCC boiler systems. Although the gas turbine operates at a relatively low temperature, it does have to be ruggedised to cope with the increased loading of solid particulates that it encounters compared with turbines burning natural gas or distillates. This would give rise to fouling, erosion and corrosion which have been solved by the turbine suppliers (ALSTOM Power) by using lower gas velocities and coated blades with specially optimised profiles. The development of hot gas filtration media and systems, to reduce the solids loading on the turbine and increase efficiency, has been partially successful. Mechanical failure can occur after ash bridging or thermal shock and chemical attack (Anthony, 2003). At the Wakamatsu plant in Japan, ceramic tube filters, on which filtration occurs on the inner, as opposed to the outer, surfaces of the candles were regarded as a success (Sasatsu and others, 2002). Elsewhere, porous metal filters are being developed as an alternative to ceramics. Anderson and others (2002a) are examining enhancement of the aluminium content of commercial superalloys.

PFBC steam cycles have reached supercritical conditions with the plant at Karita and further advance is in principle possible as materials developed for ultrasupercritical PCC cycles become available, with PFBC using more conservative conditions at first. However, the lack of direct support from ALSTOM, the original developers of the technology, is a severe constraint on further development and deployment.
6 Future plant designs

Coal-fired power plants provide over a third of world electricity and so will be needed for some decades to ensure that power supplies remain secure. Future power cycles based on coal will probably involve new configurations to accommodate carbon dioxide (CO₂) capture and its permanent storage. In their simplest forms, these may consist of flue gas scrubbing systems added after the flue gas desulphurisation (FGD) plant on PCC units, but a range of future possibilities exists, some of which even include new types of turbine to be developed (Henderson, 2003a). Whatever the means to be adopted, they will all involve changes to the energy flows within the plants to some degree. Integration aspects will be important.

6.1 Oxygen production technology

One of the challenges for IGCC and IGCC systems employing CO₂ capture is to reduce the energy requirement for oxygen production. Both the power consumption and cost of conventional cryogenic oxygen plants currently penalise IGCC economics. Ceramic ion transport membranes (ITM) that selectively transport oxygen ions at high temperatures (~600–1000°C) are the basis of a technology for air separation that could avoid these disadvantages (Stein and others, 2002; Prasad and others, 2001), but it will be important to achieve integration of the new pattern of heat flows that this will introduce: incorporation of the ceramic membrane system will involve thermal integration with other hot streams within the plant. ITM systems are expected to be available for IGCC plants within 5–10 years.

A suggested method of integration includes combustion of some of the fuel gas in the air stream feeding the ITM oxygen production modules (see Figure 30). The heat in the spent air stream is utilised by using it as the combustion air of the gas turbine. The oxygen stream is cooled for heat recovery to the steam cycle (Armstrong and others, 2003). Stein and others have evaluated a similar cycle and also one involving indirect heating of the air using gas turbine exhaust gases heated in a duct burner with supplementary fuel (clean syngas). The overall IGCC efficiency improvement was predicted to be around 1 percentage point for the former system but a drop in efficiency occurred for the latter as the steam cycle efficiency was impaired (Stein and others, 2002).

ITM oxygen plant would also benefit the thermal efficiency of a form of PCC with CO₂ capture that would employ firing of the coal in oxygen/recycled flue gas mixtures. This is discussed in Section 6.2.1.

6.2 CO₂ capture

The future challenge of incorporating CO₂ capture systems into coal-fired plant designs brings with it the need to re-evaluate the treatment of energy flows within the cycles. There are many types of plant design that have been proposed for achieving CO₂ capture from both combustion-centred and gasification-centred plants. Some examples are given below and the approaches that may be adopted for heat integration are described. In addition, there is a considerable electrical energy penalty in the need for...
compression and liquefaction of the separated CO₂. Thus in-plant power consumption will be higher than for non-capture plant.

6.2.1 PCC-based systems with CO₂ capture

These systems can be crudely grouped into two categories. The first involves using gas separation systems to remove for storage the CO₂ from the flue gas stream of a relatively conventional PCC plant. The second, more radical, method would use a partially recycled flue gas/oxygen mixture for combustion of the coal, with off-take of CO₂ for storage after condensate removal (‘oxy-fuel’ or ‘oxy-coal’ combustion).

CO₂ separation from flue gas

One of the most commercially established methods for CO₂ capture from gases at low pressure is scrubbing using chemically active agents that are regenerated by heating to release the CO₂ as a concentrated scheme. Aqueous solutions of amines such as MEA (monoethanolamine) or MDEA (methylene diethanolamine) have been proven for use under reducing conditions and are gradually being demonstrated for use for CO₂ capture from flue gas.

The main issue with chemical absorption solvents of particular relevance here is the large energy requirement for regeneration. The raw energy requirement for regeneration of MEA is around 4 MJ/kg CO₂ captured for flue gases with CO₂ concentrations representative of that from PCC (Croiset, 2004). Based on this energy requirement, a PCC plant fitted with a CO₂ chemical scrubber using such a solvent would have a thermal efficiency around 10–12 percentage points lower than a standard PCC plant, including energy for liquefaction and compression of the CO₂ for sequestration. However, improved absorbents and better thermal integration of cycles will be capable of reducing the regeneration energy penalty. For example, a sterically hindered amine (KS-1) in commercial use for urea manufacture produced by Kansai Electric Power and Mitsubishi Heavy Industries, Ltd requires as 0.3 MPa steam 3.2 MJ/kg CO₂ (Mimura and others, 2001), and it has been reported that new solvents under development based on amino acid salt solutions should require only 2.3 MJ/kg CO₂ (Feron, 2004). The power generation efficiency penalty using KS-2 in an optimised cycle is reported at around 22.5% (Gibbins and Crane, 2004).

Low pressure steam is needed for heating the regenerator, which operates at about 125°C. For a pinch temperature in the heat exchanger of 10°C, saturated steam at 0.31 MPa is appropriate. Rather than take this directly from the LP turbine (see, for example Mimura and others, 1998), Croiset and co-workers (Alie, 2003) assumed a let-down turbine fed by 1.18 MPa, 187°C cross-over steam. Other studies have also shown a thermodynamic advantage in using cross-over steam, cooled by heat exchange with high pressure feedwater, rather than using low pressure steam from the LP turbine (Gibbins and Crane, 2003, 2004). Figure 31 shows the suggested configuration. Lower grade heat flows from the CO₂ compressor coolers and the reflux condensers would be utilised in heat exchangers placed in parallel with the normal LP bleed steam heaters. Such systems are potentially suitable for existing plants but, to be economic, these would need also to be upgraded with new boiler, steam turbine and steam systems to reach state-of-the-art supercritical steam conditions. An interesting approach that has been suggested for reducing the cost involves storing the spent solvent to

![Figure 31 Use of cross-over steam for heat for CO₂ scrubber regenerator](Gibbins and Crane, 2004)
allow additional generation at peak periods with solvent regeneration carried out at low electricity demand (and price) periods (Gibbins and Crane, 2003). Dry CO₂ absorption systems based on CO₂ absorption by lithium silicate (to form lithium carbonate) at ~600°C and its reverse reaction above 700°C to release the CO₂ are being developed in Japan (see, for example, Shimomura, 2003; Kato and Moniwa, 2004).

Most of the processes based on physical processes (solvents, adsorbents and membranes) appear less well suited energetically to treating boiler flue gas, where the pressure and CO₂ partial pressure are low, than to treating higher pressure, more concentrated streams, such as are encountered in gasification cycles embodying CO₂ capture (see Section 6.2.2). A promising exception may be electrical swing adsorption (ESA), which uses an electric current passage through the adsorbent for regeneration (Judkins and Burchell, 2002). This technique, currently at the laboratory scale, could be potentially suitable for separation of CO₂ from mixtures containing almost any concentration of the gas, and would have a low energy requirement for regeneration.

In addition to the effect of taking steam from the power cycle for absorbent regeneration, there is also an energy penalty from the additional power for the CO₂ removal systems, notably the exhaust gas fan used to overcome the exhaust gas pressure drop in the absorber column – there could be scope for some improvement of efficiency by use of new absorber column packing materials – and compression of the captured CO₂ stream for transport and storage. An electrical requirement of 69 kWh of electricity per tonne of CO₂ removed, for 90% removal from the flue gas of a PCC plant using MEA, has been given by Feron (2004).

**Oxy-coal combustion**

Oxy-coal combustion involves combustion of the coal in an oxygen/recycled flue gas mixture containing ~35% oxygen (Croiset and Thambimuthu, 2000) instead of air. The CO₂-rich gases from the boiler would be cooled, condensate removed, the recycle stream returned, and the balance of CO₂ taken off for storage. Heat transfer within the boiler would be significantly better than for a conventional PCC system (McDonald and Palkes, 1999) and it is therefore possible that a capital cost saving could be made for a new installation, although the cost of the air separation system would add to this. Boiler efficiency may also be improved, but it is not certain what the overall cycle net thermal efficiency would be compared with scrubbing systems and otherwise similar conditions. The system could be applied to existing PCC boilers as a retrofit. Use of an air heater is not necessary in flue gas recycle systems and changes to heat recovery balances within the boiler and economiser would need to be calculated and flows adjusted accordingly to maintain boiler efficiency. The application of oxy-fuel technology to the retrofit of a power plant and a refinery system showed that an increase in boiler thermal efficiency occurred from the recycle of hot flue gas (Wilkinson and others, 2003).

Efficiency would be lower than for conventional processes without CO₂ recovery, as oxygen production and CO₂ compression and liquefaction would still consume considerable quantities of power. For a cryogenic air separation unit, power consumption for the delivery of 95% O₂ at low pressures is around 200–240 kWh/Mt of O₂. Laboratory and pilot scale tests of the system for PCC have been carried out (Tan and others, 2002). Use of oxygen production using ion transport membranes, described in Section 6.1, would allow a power and cost saving.

### 6.2.2 IGCC systems with CO₂ capture

IGCC with pre-combustion capture of CO₂ is regarded by many as the coal-fired power generation technology likely to be most applied within about two decades because it could give a process that is of higher electrical efficiency than PCC with CO₂ capture that would also be readily configurable for an important range of additional products (hydrogen, liquid fuels and chemicals). Cleaned gasifier product gas would be converted to hydrogen plus CO₂ by addition of steam then use of a shift reaction. The CO₂ would be separated for storage, then the hydrogen burnt in the gas turbine with nitrogen as a diluent. The additional products would be obtained by taking off some of the hydrogen and/or syngas for chemical processing. Among the hurdles needing to be overcome are perceived higher cost and complexity and disappointing reliability and availability even of ‘straight’ IGCC relative to PCC, plus the fact that large gas turbines have not been demonstrated on hydrogen (Henderson, 2003a,b).

The advantage of CO₂ removal before it reaches the gas turbine is that the shifted fuel gas is at elevated pressure and the CO₂ in higher concentration than it would be in the flue gas from the turbine (or a PCC plant). Consequently, it should be removable at lower cost and efficiency penalties by using physical methods, such as physical scrubbing, pressure swing adsorption or membranes, although chemical scrubbing could also be used.

Simulations of IGCC cycles with CO₂ capture generally show an efficiency penalty, relative to straight IGCC, of around 6–8 percentage points, including the energy required to compress and liquefy the CO₂ ready for storage. As an example, Haupt and others (2003) have calculated a 6 percentage points reduction in efficiency for a cycle capturing 90% of the CO₂ by physical scrubbing of shifted gas using Rectisol. This was for the separated CO₂ stream leaving at just above atmospheric pressure. Compression and liquefaction would increase the efficiency penalty. Fiaschi and Lombardi (2001) actually modelled an IGCC cycle with 85 % CO₂ removal by chemical (amine) scrubbing of the shifted gas. The efficiency was calculated at 38.80%, compared with 46.44% for an equivalent conventional IGCC.

There are prospects of reducing the energy penalty of CO₂ removal from IGCC systems to perhaps 4 percentage points by use of newer techniques. For example, the use of a membrane reactor could reduce the energy penalty for capturing CO₂. In this, shift-reaction and CO₂ capture would be carried out in a single reactor equipped with a dense metal membrane permeable to hydrogen.
The use of a water quench gasifier would save the additional energy losses from raising the steam that needs to be added before the shift reactor. Fuel gas shift for hydrogen separation or CO₂ capture (or both) requires addition of an excess of steam over that stoichiometrically required. Thus, the efficiency penalty of incorporating CO₂ capture systems is markedly reduced compared with the radiant cooler cycles. The exothermic shift reaction would allow HP steam to be produced in either case. There are different views on the relative final efficiencies of the CO₂-capture cycles.

A longer term possibility for gasification cycles with CO₂ capture, briefly reviewed by Henderson (2003a), is the application of oxygen combustion of syngas, with expansion of CO₂ or steam/CO₂ mixtures through specially designed turbines (Mathieu and Demaret, 2001; Anderson and others, 2002b, 2003).

6.3 Integrated gasification fuel cell cycles

Use of high temperature fuel cells together with coal gasification in integrated gasification fuel cell (IGFC) cycles, incorporating also gas and steam turbines, will permit higher thermal efficiencies than IGCC. High temperature fuel cell stacks are currently only at the scale of a few MWe, even on natural gas or reformed natural gas-derived feeds (Entchev and Douglas, 2002), so their incorporation within commercial-scale IGFC lies some way off. Fuel cells require very deep cleaning of the fuel gas to remove particulates and contaminants such as hydrogen chloride and hydrogen sulphide, and currently this could only be achieved using cold gas scrubbing systems. However, high temperature syngas cleaning will clearly favour higher cycle thermal efficiencies.

Carbon monoxide would not present a difficulty as it serves as a fuel in molten carbonate fuel cells (MCFCs) and solid oxide fuel cells (SOFCs). The MCFC would require CO₂ to be fed to the cathode as it is one of the reactants for this type of fuel cell.

The MCFC operates at about 600–700°C, while the SOFC operates at about 900–1000°C. Large power generation applications would utilise pressurised cells but the cleaned fuel gas would probably still need to be reduced from gasifier pressure by expansion through an expander (with power recovery) before being fed to the cell stack. The exit gases require extensive heat recovery and exploitation to obtain an efficient power cycle, and integration with a gas turbine or expander would be needed. The fuel cell would effectively replace the combustor in a gas turbine combined cycle by electrochemically oxidising the fuel, with the simultaneous generation of electricity. An example of an IGCC-fuel cell concept is the Japanese EAGLE project (CCUJ, 2002). Figure 32 shows the planned conceptual flowsheet for a commercial scale plant based on the technology. A recent Clean Coal Centre report describes this and a number of other IGFC cycles (Benson, 2001).

Integrated gasification fuel cell cycles with CO₂ capture

IGFC configurations are possible that maintain high efficiency while capturing CO₂ because the fuel and air streams to the fuel cell are not mixed but supplied to opposite electrodes so the CO₂ stream is not diluted with nitrogen. Thus, post fuel cell capture may be used, or pre-fuel cell capture after shift of the syngas is an alternative, akin to an IGCC cycle with CO₂ capture. External utilisation of the issuing 15–20% or so of unconverted fuel gas is also needed, and means for CO₂ capture during this, based on membrane
reactors, can be used for highest CO$_2$ abatement (Haines, 1999; Dijkstra and Jansen, 2002). A variant has been simulated by Kuchonthara and others (2001), in which shifted syngas is using a membrane separator to give hydrogen, which is then fed to an SOFC, with excess hydrogen fired in the gas turbine combustor. For cycles based on MCFCs, the supply of CO$_2$ to the cathode may be accomplished using a combustor. In order to exploit the thermodynamic advantages of all of these cycles, deep cleaning of the syngas will have to be achieved.
7 Conclusions

Coal-fired power generation plants are most commonly based on pulverised coal combustion (PCC) and a water-steam thermodynamic cycle. Heat from combustion is used to raise high pressure superheated steam which is used to drive a turbine to generate power. Coal-fired power plants can also be based on combined gas and steam cycles, which use gas turbines as well as steam turbines. This report has provided an introduction to the principles of both types of plant, with examples and background on the thermodynamics of heat engines.

Heat engines work by exploiting the work done in a cycle of expansions and compressions of a fluid as heat is added and lower temperature heat is rejected. The efficiency is related to the initial and final temperatures of the system, and is favoured by maximising the temperature range over which the engine operates.

A subcritical PCC boiler takes the pressurised preheated boiler feedwater to evaporation point then superheats it to 540°C or above for sending to the HP turbine. A supercritical boiler heats the feedwater beyond the critical point to superheat temperatures. In the steam turbine, the steam is expanded while its energy is converted into mechanical work as it passes over static and moving blades within high pressure (HP), intermediate pressure (IP) and low pressure (LP) turbines, that are usually compounded onto one shaft that drives the generator. Steam emerging from the LP turbines is recondensed then pumped back to the boiler after pre-heating. Sources of losses in turbines include aerodynamic losses, leakage of steam and leaving loss as kinetic energy in the stream issuing from the last LP turbine stage.

The thermal efficiency of state-of-the-art PCC plants is 45–47%, LHV basis, at cold sea water cooling locations. Such plants use main steam conditions well into the supercritical range with pressures approaching 30 MPa and temperatures around 600°C. All large steam turbine cycles use reheat of intermediate pressure steam from the high pressure turbine exit and multiple stages of feedwater pre-heating using steam extracted from the turbine to maximise efficiency.

The influence on performance of various factors and parameters in PCC systems, including main steam temperature and pressure, cooling water temperature, feedwater heating and auxiliary systems, has been described in the main body of the report. Because of the degree of scope for further efficiency improvement through moving to even higher steam conditions, materials development programmes are in progress in different parts of the world to reach them. These and cycle design advances are expected to realise power plant efficiencies well beyond 50%, LHV basis. For boilers, alloys are needed with resistance to gas-side high temperature corrosion and steam-side oxidation and capable of fabrication in large component sizes. Superalloys based on nickel are promising, but are much more expensive than current state-of-the-art materials. However, costs are decreasing and only parts of the plant will need to accommodate the most extreme conditions. For turbines, advanced materials and more effective cooling steam arrangements are being developed. This technology will also be based on nickel alloys and new designs are to be produced to minimise the proportion of these materials that is needed. Related materials are in use in advanced gas turbines already, but prototype steam turbine components made from the alloys require further testing and demonstration.

Combined cycles use the high inlet temperature of gas turbines as the upper temperature in a more complex arrangement. The efficiency of a gas turbine on its own is typically only around 30% because of the high exhaust gas temperature, but the latter is turned to advantage in using it as a major source of energy for the steam cycle. The use of coal as a fuel for a gas turbine necessitates measures to limit the amount of entrained particulates and liquid droplets reaching the turbine to avoid damage. Two principal approaches are used. The first, pressurised fluidised bed combustion (PFBC), employs pressurised combustion of the coal then particulates removal before expansion of the hot flue gas through the turbine expander. The pressurised air feed to the combustor is driven by the expander, so the system resembles a normal gas turbine, but with the combustor located remote from the rest of the turbine. The other approach, known as integrated gasification combined cycle (IGCC), is to convert the feed coal into a fuel gas then to clean the gas before firing it in a gas turbine designed to accommodate the medium calorific value gas. IGCC permits higher turbine inlet temperatures to be achieved. Various configurations are possible and examples have been given in this report. PFBC and IGCC plants exist at a number of locations. IGCC systems could incorporate fuel cells also, for higher efficiency, when these are sufficiently developed in scale.

In PFBC, the effect of using the gas turbine is to increase the thermal efficiency of the plant by around 3 percentage points compared with a PCC plant using the same steam conditions. The similarity of heat availability to PCC allows reheat to be used, although economics restrict this to larger scale units. In IGCC systems, there are many alternative gasifier types and configurations. The type of coal to be used is an important factor in determining which gasification technology to use. The associated water-steam cycle design depends on the gasification and gas clean-up technologies used.

Factors and parameters determining performance of PFBC systems include carbon utilisation, gas turbine characteristics, main steam temperature and pressure, cooling water temperature, gas filtration and auxiliary power consumption. PFBC steam cycles have reached supercritical conditions and efficiencies of 44%, LHV basis, and further advance is in principle possible if materials developed for ultrasupercritical PCC cycles are eventually applied.

For IGCC, the factors considered include gasification
technology, carbon utilisation and cold gas efficiency, gas turbine design, use of hot gas cleaning and auxiliary power consumption. The reliability and availability of gasification combined cycle plants are currently limited by materials considerations, including lifetime of refractory linings and corrosion in syngas coolers. Much of the expected future progression in the efficiency of IGCC cycles beyond their current 45%, LHV basis, will come from the use of new advanced gas turbines, which will also require the use of new materials, as well as technologies such as single crystal blading, compressor intercooling and reheat.

Future power cycles based on coal will include new configurations to accommodate carbon dioxide (CO₂) capture for storage. Whatever the means to be adopted, they will all involve changes to the energy flows within plant designs. PCC-based systems with CO₂ capture can be grouped into two categories. The first involves using gas separation systems to remove for storage the CO₂ from the flue gas stream of a relatively conventional plant. One of the most commercially established methods for CO₂ capture from gases at low pressure is scrubbing using chemically active agents but an issue is the large energy requirement. There is scope for reducing it using improved reagents and more optimised cycles. The second method would use a partially recycled flue gas/oxygen mixture for combustion of the coal, with off-take of CO₂ for storage after condensate removal (‘oxy-coal’ combustion). Boiler efficiency may be improved, but it is not certain how the overall cycle net efficiency or cost would compare with those of scrubbing systems.

IGCC with pre-combustion capture of CO₂ could give a process that is of higher electrical efficiency than PCC with CO₂ capture. Cleaned gasifier product gas would be converted to hydrogen plus CO₂ using a shift reaction, the CO₂ separated, then the hydrogen burnt in the gas turbine. The advantage of CO₂ removal before it reaches the gas turbine is that the shifted fuel gas is at elevated pressure and the CO₂ in higher concentration than it would be in the flue gas from the turbine (or a PCC plant). Consequently, it should be removable at lower cost and efficiency penalties.

There are alternatives to heat engines for converting the chemical energy in the fuel into work. This can avoid the inherent limitations on efficiency that heat engines have. An example is the fuel cell, which uses electrochemical processes to convert the fuel’s chemical energy into electricity. Integrated gasification fuel cell (IGFC) configurations are possible that maintain high efficiency while capturing CO₂.
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