Module 5: Plant design and selection

5.1. Objectives

On completion of this module, the student should be able to demonstrate an understanding of the reasons for choices in power plant design, and the effects these choices have on the operation and life of the plant.

In particular, the student should understand:

- the common features of pressure equipment and the issues surrounding their design, selection and operation
- the mechanisms which cause degradation of power plant components
- the thermodynamic drivers for power plant design
- the economic limits on power plant design
- the main characteristics of major power plant components and sub-systems.

5.2. Principles of power plant design

5.2.1. Type of plant—base load, intermediate, peak load

When power plants were owned by large utilities, energy was generated and despatched according to the lowest costs of generation across the whole enterprise. This tended to bias loading towards the most efficient plant. The lowest cost generation was used for base load, while higher cost generation was used for daily load-changing (intermediate loading), and the highest-cost generation used only for peak loading. The newest coal-fired stations situated adjacent to coal mines tended to be the lowest-cost generation, coal-fired stations having to transport fuel were higher cost, while stations burning oil or gas, particularly at low efficiency, were highest cost. Utility plant tended to be designed for very long life and for the lowest possible operating cost, generally with the expenditure of some extra capital to achieve the performance and longevity required.

Plant which started life as base-loaded, was often demoted to intermediate-loaded and then to peak-loaded as its efficiency and cost of generation was bettered by newer plant. Some plant (e.g. Gladstone P.S.) was designed from new for rapid load changing in accordance with the nature of the electrical transmission system at the time of construction.

5.2.2. Merchant plant

With the advent of the National Electricity Market (NEM), despatch of plant is according only to price bid into the market. This price need have no relationship to the efficiency of the plant or the cost of generation, and is determined by the strategy of the owner at that particular instant. Obviously, owners will not be willing to generate at low profit, or at a loss for long periods of time! Plant which is operated under the unfettered influence of the market may be required to undertake many shut-downs and starts, and rapid load changes. These operations can have a devastating influence on the life of major equipment in a power plant.

Merchant plant built specifically to operate in the NEM is likely to have a lower capital input, sacrificing some of the reliability enhancements of traditional utility plant. Some features of merchant plant may be omission of some of the duplicated machinery usually provided in traditional utility plants to achieve higher reliability, and deletion of beautification works which were seen to be desirable in utility plants, and provision of minimal on-site infrastructure, with reliance on provision of most services by contractors.
With good design, construction, and operation and maintenance practices, merchant plant should still be able to approach or match the performance and reliability of utility plant.

The electricity market can be very complicated, with long-term contracts which bear no relationship to the pool price which is often quoted in the media. For example many organisations (like Ergon) wish to have some certainty in the purchase price so that they can sell this electricity at a profit to us (the consumer). This purchase is done under a long-term contract with the generators. A generator will probably only guarantee that about three quarters of his output will be in service (he has overhauls and occasional shut downs) and does not wish to be exposed to the volatile short term pool price either. The outcome of all this is that on average the pool price is close to the contracted price over the long-term. However the pool price is often seen as the ‘leader’ to where the electricity price is heading. The other quarter of his potential output is then sold at the ‘pool price’ operating at the time.

Generators often bid the despatch of the units at ‘marginal cost’ and that is usually fuel cost so that they can remain in service (shutdown using fuel oil is very expensive and stressful on machinery fatigue concerns). This is why pool prices can be as low as $12 per MWhr (about the equivalent of coal purchase price of around $1.10 per GJ. This does not take into account the return on capital expected of about $10, the operation and maintenance of a further $8–10. These often appear to be forgone for the overnight run, when in reality, the generator is still being paid out of his long-term contractual position.

5.3. Basic steps in plant design

Some major factors influencing the design of a power plant are:

- the nature of the fuel source and its location in relation to major transmission lines, water supplies, and other infrastructure
- fuel transport costs
- availability of cooling water—fresh water, sea water, brackish water.

In establishing a new power plant, the following steps are likely to be followed:

- the fuel type is decided and its characteristics are determined
- the size of machine is developed (no more than 10% of the electricity grid size)
- the type of cooling is determined (wet, dry, sea water)
- the specification is written with performance guarantees
- the boiler designer then decides on furnace area and pressure parts sizing using coal characteristics (coal rank etc.)
- fan sizes and types are designed from coal properties and control philosophies
- bag filters/pollution control gear/ash disposal system are selected.

5.4. Plant selection

5.4.1. Cycle limits

As discussed in the module on thermodynamics, the plant efficiency is determined by the temperature limits between which the turbine operates. The upper temperature limits are determined by the strength, creep resistance and corrosion resistance of the materials available for the boiler, high-pressure pipework and high temperature turbines. The lower temperature limits are determined by the temperature of the heat rejection medium.
5.4.2. Cycle efficiency and key issues

The net efficiency for a 16.6 MPa/538°C/538°C single reheat utility boiler steam cycle is in the range of 34–36 percent HHV. Recent plant designs have incorporated improvements in a number of areas to increase the efficiency of this cycle to between 36 percent and 39 percent:

- Optimised feedwater heating (more heaters and ‘heating above reheat point’ or ‘HARP’)
- Lower turbine exhaust pressure through lower cooling water temperatures and the use of multi-pressure condensers
- Improved steam turbine efficiency through steam flow path design improvements made possible in part by the latest 3-dimensional numerical computational fluid dynamic (CFD) codes
- Reductions in auxiliary power loads at full load and part load, particularly in the environmental equipment area.

To achieve significantly higher efficiencies, it is necessary to substantially increase the steam pressure and temperatures, producing what is generally referred to as ‘ultra supercritical’ steam conditions. Table 1 compares the performance of a conventional 16.6 MPa plant with several sets of advanced conditions. The 16.6 MPa/538 °C/538 °C subcritical pressure (drum: single reheat) and 24.1 MPa/ 538 °C /538 °C supercritical pressure (once-through, single reheat) steam cycles are in wide use today with the availability and reliability of supercritical pressure units rivalling or exceeding typical drum units.

The design of the 31.0 MPa /538 °C /552 °C /566 °C double reheat cycle is generally considered feasible today if the economics justify the expenditures. The double reheat designs have been proven in operation since the early 1960s. The other sets of operating conditions are generally considered developmental, requiring research and field testing before they are truly considered commercially proven. Cycles with operating pressures exceeding 27.5 MPa and main superheat steam temperatures exceeding 550 °C are generally considered to be ultra-supercritical units.

The ultra-supercritical units also generally include once-through circulation at least at full load, double reheat and may include variable or sliding pressure operation to optimise low load thermal efficiency. The highest efficiencies being considered would require operating steam conditions of 40 MPa and temperatures of 650 °C.

Key boiler system issues involved in the design of these advanced power plants include:

- Advanced combustion system/furnace design
- Variable and dual pressure operation
- Spiral and vertical furnace circuitry
- Thermal-hydraulic design
- Boiler materials
- Low level heat recovery
- Advanced SO2 emissions control.

Boiler materials are of great concern, with substantial development efforts being applied worldwide to develop alloys with high performance at acceptable cost.
5.4.2.1. Cycle maximum temperatures

This section has been adapted from ‘Developments in pulverized coal-fired boiler technology’ by Kitto (1996).

A comparison of operating conditions is given for the following power plant types:

A Conventional sub-critical plant with single reheat.
B Supercritical plant with single reheat.
C Supercritical plant with double reheat, increased reheat temperatures.
D Ultra-supercritical plant with double reheat, increased reheat temperatures.
E Ultra-supercritical plant with increased main steam temperature, double reheat with increased reheat temperatures.
F Very high temperature ultra-supercritical with double reheat.

Table 1: Comparison of unit cycle efficiencies for varying maximum steam temperature and pressure configurations (Adapted from: Kitto 1996, p. 5)

<table>
<thead>
<tr>
<th>Operating Parameter</th>
<th>Units</th>
<th>A</th>
<th>B</th>
<th>C</th>
<th>D</th>
<th>E</th>
<th>F</th>
</tr>
</thead>
<tbody>
<tr>
<td>Superheater Outlet Pressure</td>
<td>MPa</td>
<td>16.6</td>
<td>24.1</td>
<td>24.1</td>
<td>31.0</td>
<td>31.0</td>
<td>41.4</td>
</tr>
<tr>
<td>Superheater Outlet Temperature</td>
<td>°C</td>
<td>538</td>
<td>538</td>
<td>538</td>
<td>538</td>
<td>593</td>
<td>649</td>
</tr>
<tr>
<td>1st Reheater Outlet Temperature</td>
<td>°C</td>
<td>538</td>
<td>538</td>
<td>552</td>
<td>552</td>
<td>566</td>
<td>593</td>
</tr>
<tr>
<td>2nd Reheater Outlet Temperature</td>
<td>°C</td>
<td>--</td>
<td>--</td>
<td>566</td>
<td>566</td>
<td>566</td>
<td>593</td>
</tr>
<tr>
<td>Cycle thermal efficiency range max.</td>
<td>%</td>
<td>37</td>
<td>39</td>
<td>42</td>
<td>43</td>
<td>44</td>
<td>47</td>
</tr>
<tr>
<td>Cycle thermal efficiency range min.</td>
<td>%</td>
<td>34</td>
<td>36</td>
<td>38</td>
<td>39</td>
<td>42</td>
<td>43</td>
</tr>
</tbody>
</table>

5.4.2.2. Cycle minimum temperatures

The traditional heat rejection medium was water—either via fresh water rivers or lakes, or to sea water. In the usual case where there is no large continuous supply of water, heat rejection to the air is the norm. If dry cooling is employed, the dry bulb temperature of the air is the lower temperature limit, whereas if evaporative cooling towers are employed, the lower temperature limit can approach the wet bulb temperature of the air. Evaporative cooling consumes large quantities of water.

In Australia, fresh water availability has always been an issue, and has now assumed critical status because of droughts and other competing demands for water.

5.5. Fuel type/s

The primary fuel source under consideration is coal. Coal can have widely-varying properties. Some of the most important from a boiler design viewpoint are:

- specific energy—the higher the specific energy, the lower the fuel consumption rate. (See volatile content, below.) When making efficiency comparisons between plants, one must always be aware of whether the efficiency is based on HHV or LHV.
- moisture content—energy is required to evaporate the moisture during milling.
volatile content—volatiles are the main carriers of hydrogen in the fuel and because of their high reactivity with oxygen, assist with initiating combustion. The combustion products of the hydrogen are water in vapour form. Traditionally, recovery of the latent heat of evaporation of the water vapour has not been attempted because the condensed water contains other combustion products such as oxides of sulphur and nitrogen which form strong acids, posing a corrosion danger to the materials of construction. The loss of this latent heat is a major contributor to the boiler losses. Methods have been developed to reclaim this energy, but they necessarily involve exotic materials ranging from high alloy steels to glass.

because of the general acceptance of the inevitability of this loss, it is common practice, particularly in Europe, to neglect the latent heat of evaporation of the water from fuel hydrogen. When specific energy content is quoted neglecting the latent heat of evaporation of water from the fuel hydrogen, it is termed Lower Heat Value (LHV). When all the available energy in the fuel is considered, it is termed Higher Heat Value (HHV).

ash content—ash is the residue of non-combustible mineral matter in the coal. Its constituents contribute most of the abrasive properties of the coal. Abrasive wear in pulverisers is usually the highest single maintenance cost in coal-fire power plants with the erosion in fuel pipes, burners, and components exposed to flue gas being other large contributors.

ash fusion temperature—ash fusion temperature is the temperature at which ash particles solidify on cooling. This can be reported against observation at several stages melting during heating. The mineral constituents of the ash are the major determining factors on the ash fusion temperature, while the nature of the atmosphere during testing (and combustion) also has an influence. Oxidising atmospheres tend to give higher ash fusion temperatures than reducing atmospheres.

5.5.1. Furnace design to accommodate fuels

Furnaces must be designed to avoid fouling, otherwise the heat transfer surfaces would quickly become blinded by ash and cease to perform their function. When PF is admitted to a furnace it encounters hot gas and a high radiant heat flux from the previously burnt fuel. This raises the temperature of the fuel particles and drives off the lowest-temperature volatile compounds. Hot secondary air is introduced in the same vicinity. The volatiles ignite, raising the temperature of the particles further and completing devolatilisation. The remaining particles consist of carbon (char) and ash constituents. The char particles react at a slower rate with the remaining available oxygen, and ideally the carbon is completely converted to CO₂, reducing the physical size and mass of the particles. The ash constituents will be melted and fused together, forming into minute spheres. The drag coefficient of particles increases dramatically with reducing diameter, so as the particles burn away their velocity and direction conforms ever more closely to the velocity and direction of the surrounding gas flow.

The high temperature particles radiate heat to their surroundings, which in most cases will be other gas and char particles, but in the case of gas at the outside of the fireball, will be the furnace walls. The gases alone in the furnace are not efficient radiators of heat, and the glowing carbon and ash particles suspended in the gas appear as a fireball, enhancing the radiation of visible, infrared and heat wavelengths to the furnace waterwall tubes.
The loss of energy by radiation to the waterwalls reduces the temperature of the gas and ash particles. The gas and particles closest to the furnace walls will be relatively cool, the aim of the designer being to ensure that the ash particles are cool enough not to adhere to the walls. Some molten particles in suspension may collide and agglomerate, forming larger particles that are too heavy to be carried out of the furnace by the gas flow. These larger particles fall to the bottom of the furnace as a component of the furnace bottom ash. In a well-operating furnace, particles impacting the wall and superheater tubes will not adhere strongly, and will fall off under their own weight or with stimulation by sootblowers.

The horizontal cross-sectional dimensions of the furnace are determined by the reaction rate of the fuel and the fusion characteristics of the ash. The furnace walls are arranged to be far enough from the burners that any particles contacting the furnace walls are cool enough that they do not adhere to the walls.

As the gas progresses upwards in the furnace, the char burns out, and the gas temperature decreases as it radiates its energy to the waterwalls. By the time it reaches the top of the furnace the gas will encounter convective heat transfer elements, so its temperature must have reduced low enough that the ash has solidified and will not adhere to any surfaces it encounters. Some particles will inevitably contact and remain attached to surfaces, but as long as they are not strongly bonded and can periodically fall off, the system is still workable. The height of the furnace is therefore also determined by the properties of the ash. The lower the ash fusion temperature, the greater the furnace height required to allow the gas to radiate to the lower furnace exit temperature.

Figure 1: Clinker formation in sheets underneath the nose of a large furnace
(Source: Photo courtesy of Mr Des Covey)
Figure 2: Large clinkers being removed from the furnace bottom in a large utility boiler by a submerged chain conveyor
(Source: Photo courtesy of Mr Des Covey)

Figure 3: Typical temperature profile in a furnace. In this example the burners are installed between the heights of 0m and 6m.
(Source: Author)
For some high-silica coal, the silica particles will emerge from the pulverisers with sharp, angular surfaces. As the fusion temperature of silica is over 1500 °C, the furnace temperature may not be high enough to melt the silica and fuse it into the less-abrasive spherical form. Some types of refractory clay are quite soft and low-wearing in pulverisers and fuel handling equipment, but the furnace temperature may be high enough to sinter the soft clay particles into hard clumps or spheres. In both these circumstances very abrasive ash particles can be produced which are potentially damaging to the components in the flue gas stream.

5.6. Heat transfer

5.6.1.1. Radiation

Radiation heat transfer takes place according to the Stefan-Boltzmann relationship

\[ q = \sigma \cdot A \cdot T^4, \]

where

- \( q \) = rate of heat flow
- \( \sigma \) = Stefan-Boltzmann constant
- \( A \) = radiating area
- \( T \) = absolute temperature

While gas is radiating heat to the furnace walls, the furnace walls are absorbing heat and also re-radiating energy back towards the gas.
High rates of heat transfer are possible while the gas temperatures are high, the furnace wall absorptivity is high, and the furnace walls are at relatively low temperatures. This situation changes when the furnace wall is covered in ash. Some ashes are highly reflective (have low absorptivity) and therefore reflect much of the radiation back into the furnace. Other ashes can form an insulating layer on the heat absorbing surfaces. The insulating layer prevents the flow of heat into the receiving surface, causing the surface temperature of the ash layer to increase. The higher surface temperature of the ash then becomes a more effective radiator, sending more radiation back into the furnace.

Only a bare outline of the principles is given above, as the prediction of radiation from a flame to furnace walls is extremely complex because of the large number of possible radiation paths, and the multiplicity of absorbing and radiating species in the gas.

The advantages of heat transfer by radiation are:

- the high temperature radiating medium does not have to come into contact with the receiving surface
- high heat transfer rates are possible because of the high temperature differences available. (This also constitutes a source of high irreversibility, and a loss of potential thermal efficiency.)

Heat transfer from the combustion zone and the upper furnace is mainly by radiation.

5.6.1.2. Convection

Convection heat transfer is effectively a two-stage process, where

(a) Heat is conducted between a surface and a fluid

(b) Physical movement of the fluid carries the heat towards or away from the surface as sensible or latent heat in the fluid.

The governing relationship is

\[ q = h \cdot A \cdot \Delta T, \]

where

- \( q \) = rate of heat transfer between a surface and a fluid
- \( h \) = heat transfer coefficient
- \( A \) = area of surface
- \( \Delta T \) = temperature difference between the fluid and the surface

Convective heat transfer is strongly influenced by boundary layer phenomena. This effect is incorporated into the heat transfer coefficient. A useful general relationship for a given geometry is

\[ h = C \cdot \text{Re}^{0.8}, \]

where

- \( C \) = a constant
- \( \text{Re} \) = Reynolds number.

Heat transfer rate by convection is proportional to fluid velocity raised to the power 0.8.

Increasing velocity gives increased heat transfer, but the temperature change of the fluid will decrease with increasing velocity.
Heat transfer by convection where phase change is involved (boiling or condensing) can be many times higher than the case where no phase change is involved. This effect is exploited in the operation of thick turbine components where heating rates are kept within desired limits by using superheated steam so that condensation does not occur.

5.6.1.3. Conduction

Conduction is of great importance in transfer of heat through fluid boundary layers, metal components and insulation.

\[ q = -k \cdot A \cdot \frac{dT}{dx} \]

where

- \( q \) = rate of heat transfer through a cross-section perpendicular to the direction of heat flow
- \( A \) = area of cross-section
- \( T \) = temperature
- \( x \) = distance in the direction of heat flow.

As discussed earlier, conduction is also intimately involved in the development of thermal stresses.

5.6.2. Heat transfer modes in boiler components

The dominant heat transfer mode between the combustion products and the furnace walls is radiation. Radiation also dominates in heat transfer to the secondary superheaters suspended at the top of the furnace. High temperature superheaters and reheaters are a mixture of convection and radiation, while convection dominates in the rear pass, with a small contribution from radiation. Economisers and air heaters operate entirely by convection.

Losses from casing surfaces is mainly by conduction, with a small contribution from radiation.

5.6.3. Heat flux monitoring

Measurement of the temperature gradient in furnace waterwall tube walls gives a direct indication of the heat flux by conduction through that tube wall. While such monitors are expensive to install, they are extremely valuable in diagnosing furnace problems, and in optimising operation with troublesome coals such as fouling and slagging types.

Callide 'C' boilers have been instrumented extensively to assist with analysing and controlling this problem.
Figure 5: Photograph of heat flux probes before installation. The thermocouple hot junctions are in the larger diameter section at the centre. The larger diameter section has been factory installed into the standard waterwall tube for simple installation in the furnace. The braced leg houses the thermocouple leads and provides protection during installation and in service. When the assemblies are installed in a vertical waterwall, the protection leg would be oriented horizontally.
(Source: Photo courtesy of Mr Des Covey)

5.6.4. Nose, throat, burner placement

The furnace nose projects across the top of the furnace, forming the combustion chamber into a semi-closed volume and enhancing radiation heat pickup. In doing so it also shields the highest temperature superheater and reheater elements from direct furnace radiation, restricting the heat transfer to those components.

The furnace throat is formed where the waterwall separation is reduced at the bottom of the furnace, reducing the area at the bottom of the furnace where radiation could be lost. Some designs have an offset in the throat sides so that the ash hopper does not ‘see’ the hottest part of the fireball above. The burners are placed in the waterwalls low in the furnace. Some designs (Gladstone) have all the burners installed in one wall. Others fire from both front and rear walls (all the Japanese-built units in Qld. and NSW installed in the 1970s–1990s.) Opposed firing from 2 walls is intended to give a more compact arrangement of burners in the furnace, allowing a lower boiler for a given rating. Others are fired from the 4 corners of the furnace. (Victorian brown coal units, some older CE/ICAL units in NSW.)

The requirements of wall-fired burners is different from corner-fired burners. Wall burners are intended to be stand-alone, whereby each burner forms its own local flame pattern suited to the coal and radiation requirements. Corner burners still require individual stability at low load, while at high load they are contributing to the single fireball in the furnace, and can operate with less swirl and less-intense mixing of fuel and secondary air.

It is claimed by the manufacturers that the single large fireball symmetrical about the furnace vertical centreline gives a predictable and reliable temperature distribution across any horizontal section in the furnace because the burners are grouped in 4s, one to each corner. Putting more burners in service will change the vertical temperature profile, but the horizontal section profile will retain its symmetry. The fraction of the total heat flux to each vertical wall strip will remain essentially constant.

The corner burners are often provided with a tilting mechanism to allow the fireball to be moved up and down in the furnace, changing the ratio of heat pickup between the furnace and the superheaters and reheaters.
5.6.5. Furnace sizing for performance

A furnace must be sized large enough to absorb the heat required to evaporate the total mass flow of steam. The following discussion assumes that the superheaters and reheaters were originally sized correctly for the design conditions. A furnace which is too small will be unable to evaporate the required mass of steam, will operate with high furnace exit temperatures, and is likely to produce high superheated steam temperatures. The high metal temperatures in the superheaters will reduce their creep life expectancy. High desuperheater spray water flows will result in efficiency losses. The high furnace gas exit temperatures are likely to create problems of fouling in the superheater elements.

A furnace which is too large for the required duty will be able to evaporate the required mass of steam, but will have difficulty achieving the correct final steam temperatures. The furnace radiation will reduce the furnace exit gas temperature below optimum, so that the rate of heat transfer by convection in the superheaters and reheaters will be reduced.

Efficiency losses will result from final steam temperatures being lower than design. Steam temperature control could be difficult because the desuperheater sprays will not be in operation unless the final steam temperature set point is set below design. The creep life of superheater and reheater element tubes is likely to be higher than design.

Methods of overcoming small furnace size are:

- overfire air—some of the combustion air is bypassed around the burners so that they are operated at sub-stoichiometric conditions. This causes an increase in flame temperature, increasing heat radiation to the adjacent waterwall tubes. The remaining air is added above the burner zone to provide enough excess oxygen to complete combustion the char. The same technique can be used to reduce the production of oxides of nitrogen (NOx) in the furnace.

- provision of additional economiser surface, to increase the temperature of water entering the waterwall tubes. There is a limit to the scope of this method in natural circulation boilers, as if the water entering the drum is too close to saturation temperature, the natural circulation effect will be suppressed.

5.6.6. Furnace construction

For maximum heat transfer potential, furnaces are constructed so that all the radiation from the flame is intercepted by heat transfer surfaces. The furnace walls are formed from closely-spaced water-filled tubes. The tubes may be almost touching (tangent tube construction) or separated by membranes welded to the tubes (membrane construction).
Furnace Radiation

Water and Steam

Water and Steam

Water and Steam

Radiation Blocking Members

Wall Tube Attachment Member

Tube Attachment Welds

Figure 6: Typical tangent tube construction. A seal casing is required behind the tubes, together with refractory to protect the casing from radiation. (Source: Author)

Figure 7: Typical membrane wall construction. The welded membranes act as heating fins for the tubes and serve to seal the furnace. Fewer tubes are required than for tangent tube construction. The thermal loading on each tube is higher than for tangent tube construction. (Source: Author)

Tangent tube construction is more tolerant to temperature differences between adjacent tubes, because they are not directly connected. More tubes are required to cover a given area than for membrane construction. Tangent tubes can be difficult to weld in-situ because of the restricted access with the close proximity of adjacent tubes. They require additional sealing to obtain an air-tight casing.

Membrane construction is inherently sealed by the fully welded membranes. The membranes provide additional heat transfer surface, receiving heat by radiation and conducting it into the adjacent tubes. The cost of construction is increased by the need to weld the membranes, but this is offset by the reduced number of tubes required. Further cost savings are made by being able to dispense with additional sealing behind the tubes. As
the membranes must experience a temperature gradient from the centre of the membrane to the adjacent tubes, thermal stresses must result. If the heat flux in an area is particularly high the thermal stress may be high enough to initiate cracking. If the cracks propagate into the tubes, forced shutdowns will result. The fatigue effects of cycling stresses due to changing heat flux during ash shedding, sootblowing or water blasting all contribute to additional risk of fatigue failure.

Furnaces of large power plants are rectangular in horizontal section because of the need to place burners and accommodate superheater components. The furnace must be essentially sealed to prevent escape of hot gases and the ingress of unmetered cold air.

Natural circulation boilers have the tubes running vertically in the front, rear and side waterwalls. There are wide variations in the heat flux seen across any particular waterwall and also over its height. Natural circulation boilers can be designed to tolerate this because an increase in heat flux on any tube will cause increased boiling at constant fluid temperature on the inside surface of that tube. The increased steam volume fraction provides an additional driving head for natural circulation, thereby providing additional cooling for that tube. As working pressure is increased, the specific volume of the steam becomes less and the convective driving head becomes less.

Once-through boiler designs usually have tubes arranged in spiral fashion so that each tube wraps around the circumference of the furnace, and is exposed to the full range of heat fluxes incident on the furnace walls. The intention is that all tubes absorb approximately equal amounts of heat in total, and deliver fluid at approximately the same temperature.

![Figure 8: Furnace with spiral-tubed walls for supercritical pressure](Source: Singer 1991, p. 7-35)
Notwithstanding the type of unit, there is a limiting heat flux which can be tolerated in furnace waterwall tubes. Heat radiated from the furnace is absorbed by the tube metal, raising its temperature. Heat flows from the tube metal to the water in the tubes driven by the temperature gradient across the tube wall. In a sub-critical unit with the water at boiling temperature, local boiling occurs at the tube inner surface, with steam bubbles forming and helping to drive natural circulation in the tubes. The heat transfer coefficient via this boiling process is very high, and therefore tube metal temperatures are kept low. If the water flow is not high enough to match the heat flux impinging on the tube outer surface, the steam bubbles cannot be removed fast enough to keep the tube cool, and a blanket of steam may form on the inside of the tube, partially insulating it. This condition is termed departure from nucleate boiling (DNB). The tube wall will then undergo a rapid temperature rise, lowering the metal yield point, causing metallurgical changes and often resulting in premature failure.

Natural circulation boilers are prone to this condition at high-pressures, when the density of the steam bubbles is not much greater than that of liquid water. Some protection against this problem is provided by rifled tubing—where a slow-turning helix is formed by raised ribs inside the tube. The rotation imparted to the water passing through the tubes produces a radial acceleration field whereby the higher density water is centrifuged to the outside of the water flow where it scrubs the tube inner wall while the lower density steam is lifted from the tube wall towards the centre of the tube. The turbulence generated by the rifling also helps to increase the rate of heat transfer.

The Babcock Hitachi 350MW units at Tarong, Callide and Stanwell all have rifled tubes in the lower part of the furnace where the flames are very turbulent and temperatures are uneven around the main fireball.

Figure 9: Sections of rifled tubing, showing stages of development over time
(Source: McDonald & Kim 2001, p. 5)
Figure 10: Comparison of minimum allowable entering velocity for rifled tube and smooth tube, illustrating the greater safety margin with rifled tube. The vertical scale is feet per second.
(Source: Durrant, Babcock & Wilcox 1977)

Boilers at supercritical conditions do not experience this evaporative cooling effect, and hence there must be close control over water flow through the tubes to ensure that tube metal overheating does not occur.

Spiral tube construction is another solution to this problem, where, by inclining the tubes, the same area can be covered by a smaller number of tubes, although each tube run will be of greater length. The reduced number of tubes gives a reduced cross-section for fluid flow, and hence a greater velocity for the same fluid flow rate.
Figure 11: Basic principle where spiral-wall furnace tubes are arranged to occupy the same horizontal space as a greater number of vertical tubes
(Source: Singer 1991, p. 7-35)

Furnace waterwall tubes are usually manufactured from carbon- or low-alloy steel, which has a tendency to react with oxygen in the water and steam, forming a layer of oxide on the inner surface of the tube. The thickness of this oxide layer increases with time, and forms another barrier to heat transfer from the tube to the water. This causes an increase in the tube metal temperature, further increasing the rate of oxide formation and shortening the creep life of the tube. External oxidation of the tubes is also enhanced by this effect. The internal oxide buildup may be removed regularly by chemical cleaning, or be prevented from forming by appropriate feedwater treatment such as the deliberate introduction of controlled, small amounts of oxygen.

The thickness of an undisturbed layer of internal oxide can be used as an indicator of the operating temperature experienced by a tube during its life. This temperature indicator will be lost if the tube is chemically cleaned. In instances where chemical cleans are carried out, meticulous records of oxide thickness prior to the clean should be kept, as well as uncleaned samples of the tube with the oxide layer intact.

5.6.7. Buckstays

In practice, the furnace pressure is continually varying because of variations in the rate of fuel supply, the rate of reaction in the flames and errors in pressure control. The furnace is intended to operate at slightly below atmospheric pressure. The large flat walls of the furnace have almost no inherent bending stiffness, and therefore even small pressure fluctuations in the furnace would cause them to flex. They are given some modest stiffness by providing buckstays—substantial beams or trusses placed to resist the horizontal bending tendency of the waterwalls. The design pressures for the buckstays are quite low—a few kPa—but the very large surface area of the waterwalls multiplies this out to very large forces and bending moments.

The waterwall tubes operate at fluid temperatures of about 350 °C. As the buckstays must be kept at close to ambient temperature to maintain their strength, the differential expansion between the waterwall elements and the buckstays must be accommodated to avoid structural failure. In cases where this effect has not been taken into account or has been inadvertently lost, tube failures are almost certain to occur.
Callide ‘C’ had insufficient clearance in the vertical membrane wall of the super critical boilers. This caused severe buckling of the wall with subsequent cracking of the membrane bar and snapping of the various mounting brackets.

Figure 12: Large vertical crack in a membrane waterwall, caused by insufficient provision for thermal expansion
(Source: Photo courtesy of Mr Des Covey)

Furnaces could be exposed to pressures as high as + 350 kPa due to a fuel explosion, or as low as the full head of the ID fans (say –7 kPa). It is considered uneconomic to provide buckstays for pressure variations even as large as the ID fan differential, and therefore designing for explosion pressures is out of the question. The buckstays therefore keep the furnace walls in the desired shape during normal operation, and control measures are provided to protect the furnace against explosion or implosion. (See Burner Management System.)

5.6.8. Furnace cleaning

Some furnaces are essentially self-cleaning, while others need ash removal with sootblowers to keep the radiation heat transfer rates high enough. Deposits which fuse into clinkers may be too hard to remove by sootblowing but can be removed by water blasting. Red-hot clinker which is hit by a cold water jet experiences thermal shock and undergoes cracking. Water jetting (also called ‘water cannons’), if not controlled tightly can result in water hitting the hot surfaces of the furnace tubes, causing thermal cracking in them as well.

Large load changes can induce temperature changes and thermal expansion differences between the clinker and the boiler components sufficient to break the bond to the clinker and induce shedding.
The highest heat flux areas (just above the burners) are usually cleaned with short stroke sootblowers located at several levels at around 2–3 metres apart. Recent boilers in Germany have been constructed with water cannons as an alternative to sootblowers. With a single wall pressure parts penetration, the water cannon shoots a jet of water through the flame to the opposite side of the boiler, to clean the clinkers from the walls. This one penetration per wall is far better than 8 or 10 on a large boiler, particularly when the cannon can cover such a large area. Callide ‘C’ relies on water cannons to keep clinkers under control.

5.6.9. Ash discharge

Ash particles which agglomerate either in suspension in the furnace or on the heat transfer surfaces will usually be too large to be carried out of the furnace by the gas flow and hence fall to the bottom of the furnace, through the throat into the furnace bottom ash hopper. About 10 percent of the total ash is expected to take this route.

Some heavy particles also settle out in hoppers below the economisers and air heaters. The remainder (about 85 percent) is carried through to the flue gas cleaning plant where it falls into hoppers either for short-term storage or for immediate removal.

5.7. Features common to pressure equipment

5.7.1. Basic design

All pressure parts tend to be basically cylindrical or spherical because of the inherent structural efficiency of this form. (These are the shapes naturally adopted by flexible bodies under internal pressure.) Most components are designed as thin cylinders, where the maximum stress (in the hoop direction) based on the maximum shear stress theory of failure is given by (ref. AS1228 – 1997, eqn. 3.2.1) [2]

\[
\sigma = \frac{p \cdot (d + t)}{2 \cdot \eta \cdot t}, \quad \text{where}
\]

\[
\sigma = \text{hoop stress (MPa)}
\]
\[
p = \text{applied pressure (MPa)}
\]
\[
d = \text{internal diameter (mm)}
\]
\[
t = \text{wall thickness (mm)}
\]
\[
\eta = \text{ligament efficiency. This can be viewed as the inverse of a stress-concentration factor, and accounts for effects such as the loss of load-resisting material due to holes or branches, or a reduction in material strength due to welds.}
\]

In a uniform cylindrical component under internal pressure and without other loading, hoop stresses are 2 x longitudinal stresses. The first consideration of pressure component design is therefore usually hoop stresses, with other stresses being considered as secondary effects.

Fracture mechanics can be used to determine whether a defect in a component will propagate as a crack, and the manner of propagation of that crack. Some cracks will advance at a predictable rate, in which case they can be regarded as (temporarily) safe. Other cracks will accelerate under certain conditions and cause catastrophic failure. Such cracks are extremely dangerous, as they will cause failure without obvious warning.

If a crack can advance through the wall of a component without catastrophic failure, this mode of failure is termed ‘leak before break’ and is preferred, as the component gives its own warning that failure is imminent or has occurred.
5.7.2. Creep

At the temperatures commonly encountered in thermal power plants, many of the components are operating in the creep regime, where the dimensions of components will gradually increase over time in the direction of applied tensile loads (the usual case). Continuity demands that the sectional area perpendicular to the direction of stress will decrease, resulting in increases in the applied stresses. The reverse will apply with components subjected to compressive loads.

A component subjected to an applied tensile load will have its creep rate determined by two competing processes:

- **Primary creep**, when there will be an initial process of continually decreasing creep strain as a result of work hardening. This proceeds according to an inverse exponential time law.
- **Tertiary creep**, when there will be a process of continually accelerating creep resulting from increasing stress (due to reduction of cross-sectional area) and material degradation due to thermally-assisted microstructural changes. This proceeds according to an exponential time law, until rupture occurs.

There may often be circumstances where the rate of reduction of primary creep is matched by the rate of increase of tertiary creep, whereupon an apparent steady state creep rate is observed. This pseudo-steady state is usually referred to as secondary creep. The most modern methods of creep assessment do not rely on the concept of secondary creep.

Most components operate at quite low stress compared to their yield strength at the operating temperature, and consequently spend most of their lives in the tertiary creep mode.

The creep of these components can then be represented by the theta projection relationship of Evans and Wilshire (Wilshire & Burt 2006):

$$\varepsilon_C = \varepsilon_p + \varepsilon_T,$$

where

$$\varepsilon_p = \theta_1 \cdot (1 - \exp(-\theta_2 \cdot t)),$$

$$\varepsilon_T = \theta_3 \cdot (\exp(\theta_4 \cdot t) - 1).$$

If primary creep is small, tertiary creep is the dominant feature throughout the life of the component.

$$\theta_3 = G_3 \cdot \exp\left(\frac{H_3 \cdot \sigma}{\sigma_y}\right)$$

$$\theta_4 = G_4 \cdot \exp\left(-\frac{-Q_4 - H_4 \cdot \sigma}{R \cdot T}\right)$$

(Wilshire & Burt 2006)
where

\[ G_3, G_4, H_3, H_4 \] are all material constants,
\[ t = \text{time(sec.)} \]
\[ \sigma = \text{applied stress (MPa)} \]
\[ \sigma_y = \text{yield stress (MPa)} \]
\[ R = \text{Universal Gas Constant (J/kg\_K)} \]
\[ T = \text{absolute temperature (K)} \]

The nearly exponential relationship of creep strain with time is exploited by several life assessment methods.

These equations also show that creep rate is an exponential function of stress, confirming that creep life is inversely related to the exponential of stress.

Figure 13: Showing a typical creep rupture curve. As the component should not be run to failure, the safe useful life in this instance would be no greater than 60,000 hrs unless close condition monitoring was implemented.

(Source: Author)

Summarising the above, component life is proportional to the negative exponential of stress and to the negative exponential of temperature. Therefore, if components can be operated at lower temperature, only a small decrease is required to achieve a very large increase in creep life. A similar argument applies with stress. Obviously, plants are operated at high temperatures for good thermodynamics reasons, and therefore reduction in operating temperatures will be expected to be accompanied by undesirable effects such as reduced thermal efficiency. If plants are operated at higher temperatures to gain thermal efficiency, plant life will be reduced.
5.7.3. Thermal stresses

Material subjected to temperature gradients will experience internal stresses due to differential thermal expansion. Where these gradients are large, the stresses may be high enough to exceed the elastic limit of the material, causing immediate plastic flow and damage to the component. Even if the stresses are lower than the elastic limit, but are applied for a substantial time, additional creep damage could be incurred. This process is termed creep-fatigue.

5.7.4. Fatigue

Small stresses applied repeatedly a large number of times can also initiate damage, usually characterised by crack initiation and followed by crack propagation leading to failure. This process is termed high-cycle fatigue. Typical causes may be mechanically-induced or flow-induced vibrations, or stresses in shafts due to misalignment.
Figure 16: Fracture surface from a turbine blade. The ‘beach marks’ are typical evidence of a fatigue-induced failure. (Source: Photo courtesy of Mr Des Covey)

Figure 17: Thermal fatigue life curves for a typical superheater outlet header. (Note that the temperature units are Fahrenheit.) (Source: Singer 1991, p. 7-43)

Figure 18: Thermal fatigue curves for a high-pressure steam turbine. (Note that the temperature units are Fahrenheit.) (Source: Singer 1991, p. 7-45)
5.7.5. Low-cycle fatigue

Large stresses need be applied only a small number of cycles to incur significant damage. Even hydrostatic testing can cause stresses to reach levels high enough to cause low-cycle fatigue.

All of the above are life-limiting effects, although not all may have been taken into account in the plant design. Designers may rely instead on placing limits on operations to avoid troublesome combinations of damage mechanisms.

Operators can get the greatest benefit from plants by knowing and exploiting these effects.

Assessment of cyclic stresses (including thermal stresses) can be difficult to assess because of the complexity of the geometry of many items of pressure equipment. German standards such as TRD 301 Annex 1 (ICS 27.040 1996) have documented methods of assessment with algebraic solutions to commonly encountered industrial problems. The diagrams in Figures 15 and 16 show the results of assessment of a header.

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**Figure 19:** Allowable temperature differences for a high temperature header, calculated according to TRD 301 Annex 1  
(Source: Author)
Methods are available to estimate the damage from any single cause or for a combination of causes. These are not always simple.

Two of the well-accepted methods are:
1. ASME Boiler and Pressure Vessel Codes
2. British Energy R5 Codes

Some general rules can be applied:
- If thermal stresses are unavoidable, their effects can be minimised by limiting the magnitude of spatial temperature gradients.
- Careful choice of material combinations can reduce thermal stresses.
- If thermal expansion has an effect for a large proportion of the time in operation, the components or assembly should be designed for minimum stress at the operating temperature.
- Good erection practice can eliminate misalignment problems.
- Low rates of change of temperature can keep thermal stresses to acceptable levels.
- The duration of application of thermal stresses should preferably be kept short.
- Mis-matches in thermal expansion between joined components can be accommodated by designing for low stiffness in components, assemblies and their supports.
5.7.6. Erosion

Many of the components in boilers burning solid fuels are in contact with gases carrying suspended particles. When the suspended particles are as hard as or harder than the boiler components, erosion can occur. Erosion rate increases with particle size, particle hardness (if the particles are softer than the target material), particle angularity and number of particles.

An important relationship is:

\[
\text{Erosion Rate} = k_e \cdot \dot{m} \cdot V_p^n \cdot f(\alpha),
\]

where

- \( \text{Erosion Rate} \) = rate of loss of mass per unit area
- \( k_e \) = a constant incorporating the effects of all the particle characteristics including diameter, shape factor, hardness, density etc.
- \( \dot{m} \) = mass flow rate of particles per unit area
- \( V_p \) = particle velocity
- \( n \) = velocity exponent—usually about 2.5 for the conditions applying in boiler hot flue gas erosion.
- \( \alpha \) = angle of particle approach to surface. 90° is perpendicular to the surface.

Sharp pressure gradients will encourage high velocities. Gas leaks and unintentional bypasses around components and assemblies typically are sources of sharp pressure gradients. Bends and changes of direction can also cause serious imbalances in flow velocities and particle distributions.

Figure 21: Typical shape of function \( f(\alpha) \), showing the variation of erosion rate with angle of attack on a ductile target. Different materials are likely to have variations, but the general relationship will still apply.

(Source: Author)

This indicates that, independent of particle characteristics, erosion rate increases rapidly with increasing velocity of the impacting particle, and varies with the angle of impact. The nature of the angle relationship depends on whether the impacted surface fails by ductile flow or brittle fracture. In ductile materials, erosion rates are highest when the angle of
impact is about 40°–50°. All the base materials used in heat transfer pressure parts have ductile erosion characteristics. In brittle materials, erosion rates are highest when the angle of impact is 90°. Brittle materials can be used for erosion protection, but not for the pressure parts.

The important point is that if gas velocities are kept low, particle erosion will be minimised. Low angles of impact will also result in low erosion rates. Impact angles from 20° to 60° with ductile materials are to be avoided if possible.

5.8. Pressure parts

Some important design features of the pressure parts will now be discussed.

5.8.1. Economisers

Economisers are provided to recover low-temperature heat from the flue gas before it leaves the boiler casing. Operating temperatures are low enough that creep is not normally a design consideration.

5.8.1.1. Plain-tube

The simplest plain-tube type is relatively trouble-free, as the resistance to gas flow is uniform across the whole assembly. The gas-side heat transfer coefficient is low, and therefore this type may not give the best performance for the mass of material used.

5.8.1.2. Finned-tube

The finned tube type has fins welded to the outside of the tubes, increasing the gas-side surface area. Considerable gains (typically > $\times 6$) in heat pickup can be achieved within the same volume as a plain-tube economiser.

The close proximity of the fins creates a significant blockage of the gas flow path, contributes additional friction loss, and is subject to blocking by large particles of ash in the gas stream. The return loops in the tubes are not fitted with fins, so the plain bends can present a low-resistance flow path for the flue gas. This allows hot gas bypassing of the fins, reducing the effectiveness of the heat exchange, while the mis-match in gas-side resistance between the finned tube and the plain tube allows the bypass stream to attain high velocities. This then contributes to erosion. Equalisation of flow path resistances to reduce peak velocities and to force as much flow through the fins is an effective solution.

5.8.1.3. Economiser inlet header

The economiser inlet header is a typical cylindrical pressure component with a small number of large diameter water inlets, and a large number of small diameter outlets to the economiser tube stubs. Water temperature can vary widely and at a fast rate, particularly when high temperature feed heaters are suddenly removed from service. The rapid quenching of the bores of the outlet holes can initiate cracking from thermal stresses. Thermal protection sleeves in all of the outlet holes may overcome this problem, but at a significant cost in manufacture, reliability and maintenance. Control of thermal transients in the feedwater to the boiler is an effective control.

The economiser inlet header can experience high localised velocities. Certain combinations of feedwater chemistry, temperature and velocity can induce flow-accelerated corrosion (FAC), a phenomenon where the fragile products of corrosion are removed by the high velocity fluid, exposing fresh metal to further corrosion. Conditions from the high temperature feed heaters to the economiser inlet header are such as to maximise the risk of FAC, with the potential to reduce the wall thickness of the header shell. At its original
thickness the header should have been a very safe component, whereas in a thinned condition the header could fail catastrophically.

It is not feasible to alter the temperature, redesign with much larger diameter header shell and outlet holes and stubs to reduce velocities is likely to be expensive, whereas control of the feedwater chemistry provides a reliable solution.

5.8.2. Drum

The drum is a large cylindrical component with many penetrations for tubes, pipes and downcomers. Its temperature is low enough for creep not to be a cause for concern. Its large diameter and high operating pressure results in substantial wall thickness—typically about 130 mm on a modern boiler. This is likely to be the thickest metal pressure component on the boiler, and renders it susceptible to damage from thermal stresses. To control thermal stresses, limits are placed on drum temperatures such as:

- difference between inner and outer surface
- difference between top and bottom.
- rate of change of temperature.

5.8.3. Downcomers

Downcomers are effectively large pipes conveying water from the drum to the lower waterwall headers. They need to have enough flexibility designed in to allow for the small temperature differences between the water from the drum and the average metal temperature in the waterwalls. As they will be of substantial size and mass, their supports need careful attention in design, construction, operation and maintenance to ensure that unintended forces and bending moments are not generated internally or transmitted to the drum and headers.

5.8.4. Superheaters—radiant, convective

Superheaters consist of inlet headers, heat exchange tubes and outlet headers. The inlet headers are pressurised cylinders. Some potential problems are thermal shocks, bending stresses due to incorrect supports, and bending stresses from attached pipework.

The heat exchange tubes are usually the most critical for material selection, being exposed to flue gas on the outside, steam on the inside, and operating at temperatures between those of the gas the steam. The principal mode of failure is creep. Material thickness is increased in steps along the direction of steam flow to decrease hoop stresses to compensate for the increase in metal temperature and the corresponding reduction in creep strength. When metal temperatures become too high for one material, a transition is made to another material with higher creep strength. Creep-related failures can occur at transition welds between different materials because of different coefficients of thermal expansion. These transitions must be designed carefully to avoid large stress buildups, sometimes involving a transition with a metal of thermal expansion coefficient mid-way between the two target materials. Convective low temperature superheaters are arranged with steam flow in counterflow to the gas flow for efficient heat exchange.

The highest temperature superheaters often have the steam and gas flows parallel. That is, increasing steam and metal temperatures are paired with decreasing gas temperature so that steam and metal temperatures are self-limiting to a certain extent.
5.8.4.1. Internal oxidation

Steam will usually react with the tube material at the inner surface. The reaction obeys an Arrhenius relationship where reaction rate is exponentially proportional to absolute temperature. The internal oxide layer acts as a resistance to heat flow, causing an increase in tube metal temperature similar to the case with water wall tubes. Increasing metal temperature then increases the rate of oxide formation and reduces the creep strength of the metal.

If the oxide layer is tightly adherent, its thickness will continue to increase, causing premature creep failure of the tube. If it is stable in service, but detaches after large temperature changes (as for austenitic stainless steel Grades 321 and 347), the shed scale can damage turbine blades or restrict flow through the tubes when the boiler returns to normal service. The resulting overheating then causes short-term rupture due to overheating.

This problem only becomes serious at temperatures above about 550 °C for low alloy steels, and approaching 600 °C for austenitic stainless steels. It can be overcome by correct material selection at the lower temperatures, and by surface treatment of austenitic stainless tubes such as shot blasting or chromizing. Chromizing is a process where chromium is diffused into the surface of a metal by dissociating a chromium-bearing compound in close proximity to the surface at high temperature. Diffusion of the chromium atoms into the surface of the base metal is facilitated by the high temperature.

Figure 22: Illustration of the increased growth rate of internal oxide in the area of highest heat transfer (and therefore highest metal temperature). The insulating effect of the internal oxide raises the tube metal temperature and favours accelerated growth at the corresponding location on the outside of the tube.

(Source: Author)
5.8.4.2. External oxidation

The outer surface of the tubes is exposed to high temperature gas containing some oxygen. Oxidation will occur, also according to an Arrhenius-type relationship. As reaction rate is exponentially related to absolute temperature, by keeping metal temperatures down, oxidation rates will be reduced. Refer Figure 26.

Oxide layers have virtually no strength, so loss of metal due to oxidation will increase operating stresses, with a corresponding exponential increase in creep rate.

5.8.4.3. Erosion

Superheater tubes may experience erosion due to particle impact in the flue gas stream, or to particles entrained in sootblower steam jets. The conditions is easily recognised by visual inspection or direct measurement. Feeling by hand is one of the quickest and most sensitive methods of detecting external erosion of tubes.

5.8.4.4. Supports

The performance of superheaters elements is predictable only when the tubes remain in their designed positions. If they move out of position, they can be exposed to different radiation intensities, gas velocities, and sootblower jet velocities, rendering their performance and life uncertain. Supports must therefore be designed and maintained to ensure that the elements are always kept in their correct location both in the gas space and in relation to one another.
5.8.4.5. Temperature deviations

Gas temperatures will inevitably be unevenly distributed across the width of the furnace, leading to uneven metal temperatures and steam temperatures across the width of the superheater. Steam temperature is controlled on the average steam temperature leaving the outlet header, therefore there will be departures in temperature above and below the measured outlet value. Superheater elements subjected to the higher temperatures will experience higher rates of consumption of creep life than expected from the measured mean value. The outlet header will see the same deviations, and will experience localised creep rates higher than the expected mean.

Tuning of burners and firing patterns can do a lot to minimise these temperature deviations.

Figure 24: Results of statistical analysis of the temperature data for a secondary superheater outlet header for 1 year. The low temperatures towards the sides of the boiler are the result of low gas temperatures in close proximity to the furnace walls.

(Source: Author)
Figure 25: Results of statistical analysis of the temperature data for a tertiary superheater outlet header for 1 year. The low temperatures towards the sides of the boiler are likely to be the result of low gas temperatures near the furnace walls, while the dip in the middle is probably due to crossing over flows from the outside of the secondary superheater to the middle of the tertiary superheater.

(Source: Author)

When the deviations are consistent over time, orifices may be installed in the header stubs to restrict flow in low temperature elements, forcing more flow through high temperature elements, reducing their temperature towards the mean value and increasing the life of the tubes and headers. The orifices impose an additional pressure loss through the superheater, which appears as a small efficiency loss in the boiler.

5.8.5. Outlet headers

Temperature deviations and rates of change are likely to be highest at the outlet header. Changes in steam temperature with time can induce thermal fatigue cracking in the stub penetrations in the header. Techniques are now well developed to simulate this effect and to calculate its effect on header life. Control measures can be put in place to limit the rate of change of temperature to minimise thermal fatigue damage.

5.8.6. Desuperheaters

Power plant desuperheaters are spray types. Desuperheaters are installed in the transfer pipes between superheater stages, and water is sprayed into the pipe to cool the steam. Variation of the steam and water flow can cause extreme thermal shocks on the inside of the pipe, running the risk of thermal fatigue cracking. A separate liner (a ‘thermal sleeve’) is installed in the inside of the pipe to keep the water and cooled steam flow away from the pipe surface until the flow has mixed properly. If the device is designed properly, the water will be finely atomised and directed into the steam flow in such a direction that it will evaporate without contacting the liner or the pipe wall. Problems are often experienced with the failure of the attachment welds between the sleeve and the pipe. This is usually related to differential thermal expansion.
5.8.6.1. Nozzle fracture

The spray nozzles are mounted on a support pipe with its axis perpendicular to the steam flow. Vortex shedding in the steam flow can excite vibrations in the spray assembly causing fatigue failures. Attention to the design of the support pipe and its attachment to the pipe wall can avoid this problem.

In summary, good desuperheater designs are trouble-free, while poor designs cause major problems.

5.8.7. Reheaters

Reheaters have the same basic characteristics as superheaters, and the same types of problems. Reheaters are often designed with a small number of intermediate headers (or none at all) between stages, thus limiting the mixing which would even out temperature deviations. Such designs will therefore be subject to the widest temperature deviations, and the widest distribution in life expectations for tubes and headers.

Reheater steam flows are usually counterflow to the gas flow, as the gas temperatures are lower than those over the superheaters.

5.8.8. Boiler suspension

Large boilers tend to be supported from the top, with virtually the whole weight hanging from the boiler house structure. This has several functions:

- Thermal expansion is all downwards from the top, and easy to account for.
- Most of the heaviest individual items—the drum and superheater headers can be held in fixed positions.
- The support structural elements can be kept compact and relatively low cost.
- With most of the components being vertical and in tension, buckling resistance does not need to be provided, helping to lower costs.
- It is relatively easy to arrange for the support hangers to be out of the flue gas stream, allowing them to operate at lower temperatures and hence have smaller sections.

When the boiler is erected, each of the hangers is adjusted to carry its correct load. Relaxation of hangers in service and deformation of components can re-distribute the loads within the boiler, imposing unintended loads. Bending moments within headers and additional forces and moments on headers and pipework can be consequences.

Techniques are available to ‘weigh’ boilers by applying lifting forces to each support hanger until it just lifts off its support beam. Once the existing loads are known, adjustments can be made to return the hanger loads to the correct values.

5.8.9. Pipework

Pipework incorporates bends, support lugs, instrument tapping points and branches. It is subject to the degradation mechanisms of creep and fatigue, and may also be subjected to thermal fatigue and corrosion.

Attention is required to ensure that supports carry the weight of the pipe in the intended manner, and that insulation is in place, ensuring that no unforeseen temperature gradients are imposed.
Bends of uniform thickness impose higher stresses at the intrados than in a straight pipe of the same thickness. The stress at the extrados is lower than in an equivalent straight pipe. The method of bending can reduce the thickness at the extrados, so that in a loosely specified bend the stress at the intrados and extrados will be unpredictable.

Welds in heavy section pipes can be subject to accelerated creep failure because a region in the weld heat affected zone has much lower creep properties than the expected properties for the rest of the material. This problem has emerged only after years of operation, so that millions of such welds are in service worldwide and pose an on-going hazard. Good welding practice can minimise this problem.

Some pipe is manufactured by rolling plate into a cylinder and performing a longitudinal seam weld. This weld is subject to the full hoop stress and has been shown to have an unacceptably high failure rate. Several catastrophic failures have occurred worldwide in hot reheat pipe and seam-welded headers with fatal results.

5.9. Auxiliary plant

5.9.1. Fans

The two main types of fans employed in power plants are centrifugal and axial flow. Mixed flow fans are also likely to occupy specialised niches.

5.9.1.1. A note on power

For any fluid motivating device operating on an incompressible medium,

$$\text{FluidPower} = \text{VolumetricFlow} \times \text{PressureRise}.$$ 

For the purposes of this argument, power plant fans can be considered to be operating on an incompressible medium.

$$\text{Volume} \propto R \cdot T$$ where

$$R = \text{Universal Gas Constant}$$

$$T = \text{Absolute Temperature}$$

Therefore, if there is a choice, to minimise its power requirement the fan should be placed at the lowest possible temperature.

Practical considerations could rule against this, such as where condensation could create an additional risk of corrosion.

5.9.1.2. A note on efficiency

In cases where the system pressure drop is proportional to (flow)$^2$, efficient flow control may be achieved by varying the speed of the fan.

$$dP_{\text{System}} = k_1 \cdot Q^2$$

As pressure rise through the fan is proportional to (speed)$^2$,

$$dP_{\text{Fan}} = k_2 \cdot N^2$$

flow will then be directly proportional to speed.
\[ Q = k_3 \cdot N \]

where

- \( dP_{\text{System}} \) = the total pressure loss in the system
- \( dP_{\text{Fan}} \) = pressure rise through the fan, \( = dP_{\text{System}} \)
- \( Q \) = volumetric flow rate (arbitrary units)
- \( N \) = speed of rotation of fan (arbitrary units)

\( k_1, k_2, k_3 \) are constants of proportionality, units consistent with the units of \( Q \) and \( N \).

In a system with a square-law resistance curve and a well-matched fan, speed control can give efficient flow control, because the full-flow efficiency will be maintained throughout the flow range. The further the departure of the system resistance from a simple square law, the greater the departure of the fan from its peak efficiency duty point, and less efficient the whole system will be. A typical departure is where a control device maintains a constant pressure difference across a component. Provision of a high-efficiency variable speed driver for the fan is a separate but equally important technical challenge.

![Figure 26: Comparison of power consumption for various fan control modes](Source Singer 1991, p. 14-18)

### 5.9.2. Centrifugal fans

#### Physical Construction

The type of fan normally employed in power plants is a high specific speed type, with backward-curved aerofoil-section impeller vanes. It runs at constant speed and its capacity is controlled by variable inlet vanes. The heavy impeller usually requires a simply-supported shaft and bearing arrangement. This arrangement renders impractical the ideal axial gas approach to the inlet, and instead a side inlet box is provided. A scroll casing on the discharge side is provided to attempt to efficiently convert the velocity head of the air emerging from the impeller to static pressure.

The heavy construction of this type allows it to be tolerant of the passage of occasional small solid objects, and moderate dust loadings.
The small number of moving parts and simple construction contribute to low maintenance costs. High gas temperatures can be tolerated, and are limited by the strength characteristics of the construction materials.

Figure 27: Centrifugal fan blade types. The discharge angle of the airflow from the blades determines the pressure and power characteristics at constant speed. (Source: Singer 1991, p. 14-7)

With radial discharge, the gas is accelerated so that the tangential velocity component is equal to the tangential velocity of the blade tip. (Ignoring slip effects.) This results in a constant pressure characteristic with varying flow, and a continuously increasing power characteristic.

With backward-curved blades, the tangential velocity component reduces with increasing flow, producing a falling pressure characteristic with increasing flow, and a limiting power characteristic.
5.9.2.1. Operational characteristics

Over most of the useful operating range the fan has a falling pressure-flow characteristic. That is, pressure rise decreases with increasing flow. This characteristic is useful for parallel load-sharing, and is inherently non-overloading.

The variable inlet vanes impart swirl to the flow before the inlet to the impeller, altering the relative velocity between the incident flow and the impeller vanes and changing the pressure rise through the impeller. Part-load efficiency tends to be poor.
At low flows the inlet swirl from the variable inlet vanes can be so high that an unstable free vortex forms in the eye of the impeller, resulting in severe buffeting on the casing. A series of axially-disposed stationary radial vanes may be provided, extending from the shaft outwards to a small fraction of the radius of the inlet cone to intercept and suppress the inner core of this vortex where tangential velocities are highest. At high loads the gas flow from the variable inlet vanes into the impeller is largely axial and therefore the stationary vanes have little effect on performance.
Low specific speed (radial vane) impeller centrifugal fans are capable of a high-pressure rise in a single stage. The backward curved impeller vanes in high capacity fans have a lower pressure rise per stage. The high rotational inertia of this class of fan means that variable speed control is impractical if rapid load response is required.

In large sizes and high single stage pressure rises, the operating stresses in the impellers are very high, requiring the use of high-strength steels and therefore special fabricating and repair procedures. The high operating stress levels render this class of fans unlikely candidates for variable speed control because of the risk of low-cycle fatigue cracking.

The slow shaft speed and modest number of impeller vanes (usually about 12) causes the generation of low fundamental noise frequencies which can be difficult to absorb or cancel out. The large flat surfaces on the casing have low inherent stiffness and are efficient radiators of noise.

Various treatments at the scroll cutoff have been used to reduce pressure fluctuations at this primary noise generation location.

5.10. Axial flow fans

5.10.1. Physical construction

These fans are usually constant speed axial flow type with variable rotor blades. They are usually of overhung impeller construction. The design can be arranged to give good entry and exit conditions for the gas flow. The major elements of the rotor can be designed to run at moderate stresses. The blades are usually bolted to carriers which form part of the variable control mechanism. The variable blade mechanism tends to be complex, and careful attention is required to minimise friction in the blade carriers. Maintenance of the variable blade mechanism can be expensive.

The casing is basically cylindrical and can be produced at modest cost. Rotation speeds tend to be high in order to achieve the required pressure rise in a single stage. This then places severe duty requirements on bearings, where rolling-contact bearings are employed to minimise friction losses. High precision in manufacture and maintenance is required in order to keep blade tip clearances small, as efficiency losses will result from large clearances, and blade damage could arise from clearances which are too small. The impeller blades are necessarily cantilevered from the impeller hub outside radius, and are intolerant to the ingestion of foreign objects. The variable blade mechanism and the multitude of internal bearings are unsuitable for operation in a dusty environment.

Variable rotor blade axial flow types are not suitable for very high temperature use because of temperature limitations on the control mechanism. Obviously, technical solutions could be developed to overcome all these drawbacks, but at the expense of increased complication, manufacturing cost and cost of maintenance.
Figure 32: Two-stage variable pitch axial flow fan for induced draft service.  
(Source: Singer 1991, p. 17-10)

Figure 33: View of adjustable airfoil blading on an axial flow fan.  
(Source: Singer 1991, p. 14-18)
5.10.2. Operational characteristics

Axial flow fans tend to be more susceptible than centrifugal fans to stalling, and fans must be selected to have adequate margin above stall to avoid unpredictable loss of air supply to the combustion process.

Control by varying the angle of the impeller blades alters the angle of attack of the blades on the air flow, thereby changing the lift of each blade and the pressure rise of the fan. Efficiency remains high throughout the operating range rendering variable speed drive unnecessary.

Figure 34: Typical axial flow fan performance characteristics, with variable-pitch blade control. Efficiency remains reasonably high throughout most of the operating range. Fan stall points (discontinuities in P-V curves) are above the normal operating conditions. The safety margin is decreasing with increasing load.
(Source: Singer 1991, p. 14-16)

The high rotation speeds and the large number of blades required produces fundamental noise frequencies at relatively high frequencies where it is relatively economical to treat. The cylindrical casing in the area of noise generation is structurally efficient with high inherent stiffness, limiting noise transmission.
5.10.3. Fan selection

The following table is a summary of the principal duties and fan selections relevant to large coal-fired power plant. Not all the fan types listed below are required in all plants.

Table 2: Typical performance requirements for various power station fans

<table>
<thead>
<tr>
<th>Fan Purpose</th>
<th>Approx. Pres. Rise (kPa)</th>
<th>Flow</th>
<th>Max. Temp. (°C)</th>
<th>Fluid</th>
<th>Fan Type</th>
</tr>
</thead>
<tbody>
<tr>
<td>Forced Draft</td>
<td>8</td>
<td>High</td>
<td>70</td>
<td>Clean air</td>
<td>Single stage, high specific speed centrifugal/single stage axial flow</td>
</tr>
<tr>
<td>Primary Air</td>
<td>10</td>
<td>Moderate</td>
<td>300 [^1] /70</td>
<td>Clean/dust-laden air</td>
<td>Single stage, low specific speed centrifugal</td>
</tr>
<tr>
<td>Pulveriser Sealing Air</td>
<td>20</td>
<td>Low</td>
<td>50</td>
<td>Clean air</td>
<td>Single/multi-stage, low specific speed centrifugal</td>
</tr>
<tr>
<td>Pulveriser Exhauster</td>
<td>10</td>
<td>Moderate</td>
<td>100</td>
<td>Dust-laden air/gas [^2]</td>
<td>Single stage, low specific speed centrifugal</td>
</tr>
<tr>
<td>Scanner Cooling</td>
<td>10</td>
<td>Low</td>
<td>50</td>
<td>Clean air</td>
<td>Single/multi-stage, low specific speed centrifugal</td>
</tr>
<tr>
<td>Induced Draft</td>
<td>7</td>
<td>High</td>
<td>150</td>
<td>Clean/dusty flue gas [^3]</td>
<td>Single stage low specific speed centrifugal/single/multi-stage axial flow</td>
</tr>
</tbody>
</table>

\[^1\] Depends on whether the air to the fan is taken from before or after the air heater. (See notes on air heater types.)

\[^2\] Not required on most modern roll-type pulverisers where high primary air pressure is available. An exhauster allows pulveriser to operate at pressures below atmospheric, reducing dust leakage problems. Erosion on exhausters due to PF can be severe. Some pulveriser systems use furnace gas rather than hot air for drying coal.

\[^3\] Modern fabric filter systems produce flue gas which is substantially cleaner than older electrostatic precipitators. Special erosion protection is required on fans handling gas with a high dust loading.

5.10.4. Airheater

Two main types of air heaters are used in coal-fired plant—regenerative and recuperative.

5.10.4.1. Regenerative Ljungstrom/Rothemuhle

Regenerative types pass the heat exchange surface alternately through hot flue gas and cold air. While in the hot gas, the corrugated sheet metal elements absorb heat through both surfaces and undergo a temperature rise. When placed in the cold air flow, this heat is given up to the air through both surfaces, reducing the metal temperature again.
The original design from Ljungstrom moves the heat exchange elements through stationary ducts. The later Rothemuhle design keeps the heavy elements stationary and instead rotates hoods which switch the air and gas flows as required.

This general class of airheater has high performance for a compact size, and can be provided at modest cost. Their compact size allows them to be located close to the boiler casing, minimising the length of run of hot gas ductwork.

Some designs are supplied with different partitioned sections to provide hot air at different temperatures for different duties, e.g., highest temperature primary air for pulverisers, and secondary air for the burner windbox at a lower temperature.
Regenerative types are the most common in modern power plants.

One of the most important features of this type is the reliance on sliding or close-proximity seals to minimise leakage from the high-pressure air to the low-pressure gas side. Ineffective seals can allow substantial air leakage, wasting fan power, upsetting the precision of the controls, and losing thermal efficiency. Controllable-gap seals are now available to minimise leakage in service.

As both the air and gas pass over the same heat transfer surfaces, some dust is left behind and is picked up in the air. This then makes its way into the windbox, dampers and air registers, promoting wear and erosion. The heat exchange elements only reach low temperatures for a short time and hence are not particularly prone to cold end condensation and corrosion.

With their close proximity to the boiler and small gas passage size, conventional design regenerative air heaters can have their surfaces contaminated during startup with unburnt fuel oil. As the gas and air temperatures rise during operation this oil can ignite and damage the air heater. The ready availability of air facilitates ignition and sustains combustion. Element temperatures can then be high enough to melt the metal.

Mechanical drives are required to move the elements or the hoods. While the power required is small, the torque can be very high. If the drive mechanism has moving parts in the gas stream, wear rates can be high.

In the event of failure of the drive, the air heater ceases to operate almost immediately. The result is uncooled gas passing through into the flue gas cleaning plant, with the potential to destroy fabric filters. The temperature of air sent to the combustion system drops suddenly, destabilising combustion.
5.10.4.2. Recuperative air heaters

Recuperative air heaters are tubular types with the gas passing through metal tubes carried in metal tube plates. The air passes around the outside of the tubes, usually in multiple successive cross-flow which approximates counterflow.

Heat passes into the tube walls from the inner surface only, and is given up to the air from the outer surface only. As a consequence tubular air heaters tend to be much larger than regenerative types, and therefore more expensive for the same performance. They are usually too large to be placed immediately adjacent to the main boiler casing.

Separate sections for different temperatures and pressures are easy to arrange with this type. As the air and gas are kept separate at all times, there is no contamination of the air stream with dust and the primary and secondary air dampers, windbox and air registers are kept clean.

This design can allow tube and tubeplate metal temperatures to go below the dew point of the flue gas, causing localised corrosion. This is overcome by recirculating enough hot air to the FD fan inlet to present the cold end of the air heater with warmer air, keeping the metal temperature above the acid dew point. The cost of this measure is a small increase in FD fan power.

Fouling tends not to be a problem, as oil residue and the like are not exposed to air, and therefore are unlikely to ignite. Once the unit is up to operating temperature, such deposits should be evaporated off or scrubbed away by the dust in the flue gas.

The large size of tubular air heaters can create problems with thermal expansion. The large mass of metal can increase its temperature by over 300 °C while the supporting structure remains close to ambient temperature. Provision must be made to accommodate the resulting expansion otherwise the airheater will be damaged. This is not a particularly difficult task, although the sheer size and number of parts can increase construction costs.

The large size of this type creates a temptation for designers to use casings of rectangular plan section and having large flat sides. This makes the simplest design unsuitable for high air pressures, with substantial stiffening being required, or limits to be applied to operations to avoid high air pressures.

Having no moving parts, this design has no drives. Being static, it always performs its function as a heat exchanger and cannot send hot gas to the flue gas cleaning plant or cold air to the combustion system. Maintenance costs tend to be low.

In summary, tubular airheaters have only one disadvantage—cost. When whole-of-life costs are considered, a good case can be made for their more widespread use.

Summary

When you finish this module you need to be able to describe the reasoning behind selection of the various components of a steam power plant. You also need to understand the factors which limit the performance and life of plant.

The learning objectives at the start of this module provide a detailed breakdown of the task described above. Make sure that you can do each activity listed in the learning objectives.

If you feel that you cannot achieve the learning objectives for this module, work through this Study Guide again and read the relevant sections from recommended books.

Remember that if you need assistance in your study, the lecturer and other University staff are there to assist you. We are only a phone call away.
# Checklist

Use the following checklist to identify whether you achieved the essential elements of each of the enabling objectives and learning objectives in this module.

## Performance criteria

<table>
<thead>
<tr>
<th>Basic Plant Design and Cycle Limits</th>
</tr>
</thead>
<tbody>
<tr>
<td>- Describe how operation of plant within the National Electricity Market can influence plant design and operation.</td>
</tr>
<tr>
<td>- Describe the factors which determine the limiting temperatures for a coal-fired steam plant.</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Fuel Types</th>
</tr>
</thead>
<tbody>
<tr>
<td>- Describe how fuel types and properties affect the design of boilers and associated equipment.</td>
</tr>
<tr>
<td>- Describe the heat transfer processes, the areas in which they dominate, and the effects on design.</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Furnace Design and Construction</th>
</tr>
</thead>
<tbody>
<tr>
<td>- Describe the essential features of furnace construction and how these are achieved in different designs.</td>
</tr>
<tr>
<td>- Describe the difference between once-through and recirculation boilers and their main features.</td>
</tr>
<tr>
<td>- Describe the phenomenon of departure from nucleate boiling and the effect this has on plant design and operation.</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Pressure Parts</th>
</tr>
</thead>
<tbody>
<tr>
<td>- Describe the degradation effects applying in high temperature pressure equipment and their effects on plant life and design. Understand the separate and combined effects of creep and fatigue.</td>
</tr>
<tr>
<td>- Describe the process of erosion and the factors which influence it, and the influence this has on design.</td>
</tr>
<tr>
<td>- Describe the process of oxidation, where it occurs, and the effect it has on plant life.</td>
</tr>
<tr>
<td>- Describe the underlying principles of pressure equipment design, and the how operational deviations from design conditions affect plant life.</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Auxiliary Plant</th>
</tr>
</thead>
<tbody>
<tr>
<td>- Describe the essential performance characteristics of the main types of fans and how this affects their selection.</td>
</tr>
<tr>
<td>- Describe the control processes used with the main types of fans, and the effects these have on efficiency.</td>
</tr>
<tr>
<td>- Describe the construction features of the main types of fans.</td>
</tr>
<tr>
<td>- Describe the main types of airheaters used in power plants, together with their advantages and disadvantages.</td>
</tr>
</tbody>
</table>
References


[7] Photos courtesy of Mr. Des Covey.