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Report on limitations of combustion-based generation technologies

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Name of the report: Report on limitations of combustion-based generation technologies

Key words: power plant, engine, boiler, furnace, control, steam cycle, minimum load, ramp rate, limitations

Summary

The report summarizes flexibility limitations of engine– and boiler-based energy production. To allow the reader to easier understand the limitations, the operation principles of the engines and the boiler-based plants are also shortly described.

Engine-based units offer excellend load following capability and can easily respond to very fast power change demands. Alhough small amount of addidional power can be obtained fast from the heat reserves of boilers, boiler-based units react in general slower to load changes. Fluctuations in production of solar and wind power are however not in second scale but more in minute and hour scale and boilers can react in this time scale. The biggest challenge for boilers comes from the need for low load operation or temporary shutdowns in situations when the amount of solar and wind power is large. Startup of a boiler is a slow operation and to reduce the need for shutdowns, the goal of energy producers is today to run the boiler at as low load as possible in these situations. The report discusses the problems related to low load operation and fast load changes.

Helsinki October 2016

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1 Background

Future energy systems will combine a wide range of centralized and local energy generation resources including a growing share of renewables and energy storages. In combustion based power production, new solutions to increase fuel and operational flexibility are required to respond to the increase in intermittent energy sources like wind and solar. The role of present units for combined heat and power (CHP) generation will also change.

All power systems will need some reserve capacity to keep the frequency stable and prevent wide blackout in case of power plant or grid failure. Common principles for procurement and settlement of frequency containment reserves, frequency restoration reserves and replacement reserves and common methodology for the activation of those reserves are given in ENTSO-E Network Code on Electricity Balancing /1/. Reserve requirements are defined in the grid code prepared by the transmission system operator.

Stabilizing reserve capacity will consist of power plants providing frequency control: they keep the demand and production in balance. Frequency control power plants must be in operation, i.e. “spinning”, and adjust their load all the time. /2/

The emergency reserve that maintains system stability in case of a production failure can be divided into three subcategories – primary, secondary and tertiary. Response times for each are categorised by a country’s grid codes. /2/ The primary reserve has to be large enough to compensate lost capacity of a large basic load power station and the primary reserve has to be spinning since the required response time is only seconds and ramp up time to full output tens of seconds. The main role of the primary reserve is to stop the frequency dip after a failure. Secondary reserve is needed in order to relieve the primary reserve back to “waiting position”. Responding time is short, typically 30-60 seconds. Tertiary reserve is backup for fast reserves.

Fluctuations in wind and solar power are typically slower since their production is spread out across a wide geographical area and thus the reserves required to compensate changes in the availability of these energy sources can have longer response times.

In the project *Flexible Energy Systems – FLEXe* (2015-2016) the target is to find solutions for future energy systems. The project includes investigations both on energy system level and on local producer level. The present report produced in FLEXe Task 4.1 summarizes current limitations and potential boundaries of combustion-based generation technologies. Both operational limitations and fuel flexibility are discussed. One objective of the report is to provide a basis for description of the units and their limitations in models for energy system level. Simultaneously, the summary is intended to serve as a means to improve the general understanding of the limitations and possibilities of the combustion based power production. The report describes the limitations that boiler-based and engine-driven power plants pose on system flexibility today. In the future, combustion-based power plants have to change their role from providing base-load power to generating fluctuating peak and back-up power to control and stabilize the electricity system. The main considerations are related to load response capabilities, allowable minimum loads, and the impacts on, e.g., combustion processes, emissions levels, plant components and economy. The energy production technologies that to varying extent are covered by the report are combustion engines, gas turbines, and boiler-based energy production, including pulverized coal fired boilers, bubbling and circulating fluidized bed boilers, grate furnaces and recovery boilers. In addition to operation flexibility, flexibility in terms of selection of fuels is also discussed. For boiler-based production, operation and operational limitations and control of the boilers, the water-steam cycle and the turbines are described in the report.

2 Engine-based power plant

2.1 Overview of the technology

Prime movers of engine power plants are typically four stroke medium speed internal combustion engines. The capacity of a single plant can be even over 500 MW but it consists of several engines or more precisely generating sets, each engine is connected to a generator via a flywheel and coupling. The largest engine power plant now is a 573 MW plant consisting of 38 engines. /3/ The output of power plant's single generating set is most often 5…20 MW. Power plants having only one generating set exists while large plants consist even of tens of units. Several generating sets means very high efficiency in a wide output range: single units can be run in the optimal range and the required output defines the number of units taken to use. Efficiency of a generating set can be around 50 % in simple cycle, and if waste heat is utilized, (combined cycle) electricity efficiency can increase 4…5 % units.

Wide range of fuels from low heating value gas to biofuels, high viscosity heavy fuel oil (HFO), emulsions or even crude oil are possible fuels for engine power plant. The main fuel affects the selected engine power plant technology, i.e. the design of the fuel handling, the engine itself and the required exhaust gas aftertreatment system. Combinations of two different fuels are common. Dual fuel engines using normally low pressure natural gas as main fuel and small amount of liquid diesel fuel to ignite the gas–air mixture are widely used. Another possibility is to inject gaseous and liquid fuels directly to the combustion chamber. Common to both techniques is that if gas for some reason is not available they switch to use liquid fuel only without interruption or power loss.

Engine power plants are used for base load, intermediate load, and peak load operation and for grid stability management. Due to a short reacting time, engine power plants are a good choice for balancing renewable wind and solar power or for peak load operation. Not only quick start but easy ramping and high part load efficiency are important features. Generating units that can be operated in parallel and deployed as needed to match the changing power requirements have already now in many areas an important role: the stability of electric transmission grids with high wind or solar production capacity is based on balancing engine power.

There are no special requirements limiting the location. Power plants can be near the large consumers (cities, factories) or next to local fuel supply thus reducing either transmission losses or fuel transportation need. Locally produced power can serve the customer specific needs. Often also heat can be utilized which increases the overall efficiency to very high level. Engines have a closed loop cooling system which means that water need is very small in simple cycle power plants. /4/ In combined cycle plants water consumption depends on the cooling system. Dry cooling system needs no cooling water but it is not suitable for power plants having large cooling needs /4/. Cooling towers are commonly used to condensate and cool the steam. Some make up water is needed to compensate the evaporation losses and maintain cooling water quality. A once through cooling system is possible if the power plant is located next to the sea, a lake or a river but due to environmental regulations recirculating cooling systems are most common systems today.

Emission legislations define the frame in which equipment manufacturers and end users can operate. Exhaust emission levels are normally adjusted according to local regulations. For engine power plants the key emissions are sulphur oxides (SOx), nitrogen oxides (NOx) and particulates, and also methane, if natural gas is the main fuel. The fuel affects emissions as well as the engine technology. Engine out SOx emission can be calculated directly from sulphur content of the fuel. Exhaust gas aftertreatment systems are used if needed. A lot of emphasis is also put on greenhouse gas (GHG) emissions. GHG emissions depend on the fuel and on engine efficiency. Advanced engine automation is one of the key enablers to provide efficient and clean engine power.

Straight citation taken from International Energy Agency's report Tracking Clean Energy Progress 2014 /5/, below heading Natural Gas-Fired Power: "Rising flexibility needs make internal combustion engines (ICEs) increasingly attractive for power, as single-unit plants (< 20 MW), stacked in so-called "bank" or "cascade" plants (20 MW to 200 MW), or operated with a combined steam cycle (> 250 MW). At 48% full-load efficiency, ICEs outperform OCGTs (< 42%) but fall short of CCGTs (< 61%), while having better flexibility and part-load efficiencies." OCGT refers to an open cycle gas turbine and CCGT to a combined cycle gas turbine.

2.2 Operation at reduced load

Operation at reduced load does not decrease the efficiency of an engine power plant significantly. Efficiency curve vs load of a single generating set differs slightly with different techniques but generally efficiency is high even at part loads. At half load, efficiency is in the worst case only about 4 %-units lower than at full load. At full load efficiency of both a lean-burn spark ignited gas engine and a four stroke diesel engine is at best over 50 % /6, 7/. Combined cycle power plant efficiency can be over 53 % /6/. Source /8/ gives 45,8 % efficiency for single cycle power plant and 50 % for a combined cycle plant. Wärtsilä gives 50 % simple cycle efficiency for a lean-burn gas engine (50SG) power plant, 48 % efficiency for a dual fuel engine (50DF) power plant in gas mode and 47 % efficiency for a liquid fuel engine (46/50DF) power plant /9/.

In practice, an engine power plant consists of more than one engine, often several engines. That's why the load of a single engine is high, and the required power adjusting is done mainly by changing the number of running engines. The efficiency of the plant is thus stable on a wide output range (Figure 1).

Ambient temperature or air pressure (altitude) has minimal impact on efficiency. Turbocharging and charge air cooling can compensate changes of ambient conditions until about 40 ^{o}C temperature and keep the efficiency high also at part loads. At altitudes of about 2000 meters over sea level the efficiency starts to decrease. /3/

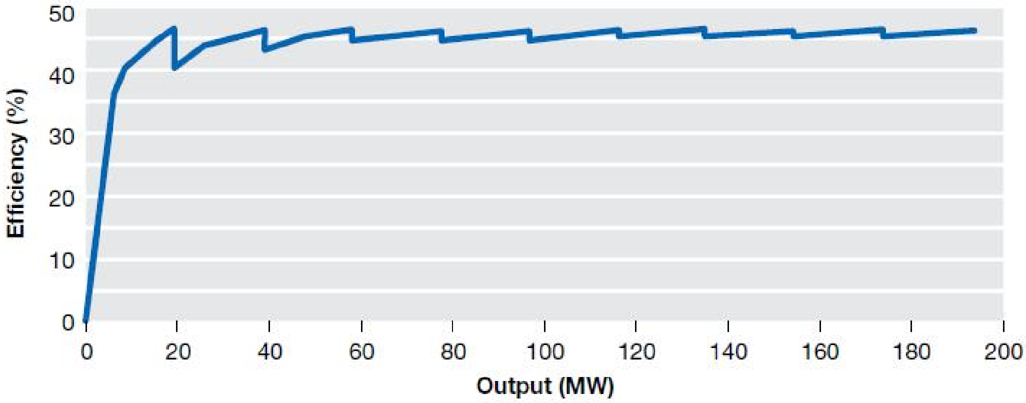


Figure 1. Multiunit engine power plant efficiency vs power output. /6/

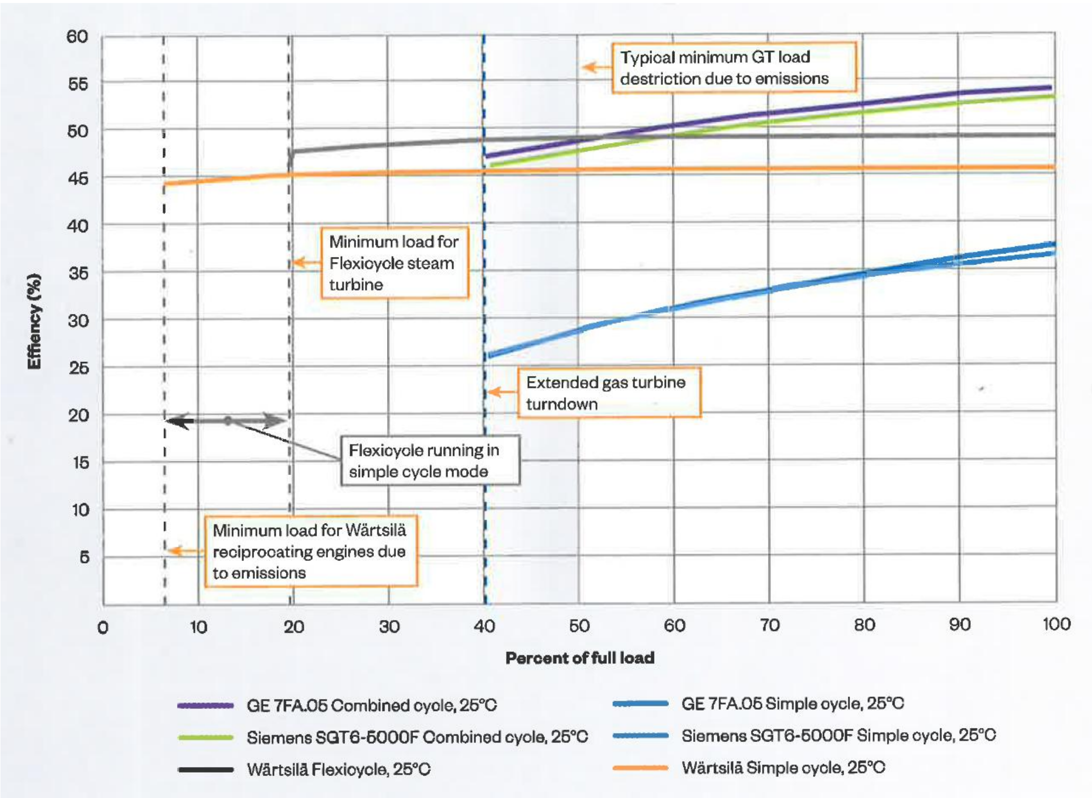


Figure 2. Efficiency curves of some engine and gas turbine power plants at 25 °C ambient temperature.. /10/

Gas turbines are often seen as the closest rival to internal combustion engines. If only efficiency at reduced load is considered combined cycle gas turbines (CCGT) win engines at high loads (Fig. 2) but engines have wider possible loading range and the efficiency curve is flat. It means for example that engines can be run on low load and used as spinning reserve without efficiency penalty. On the other hand efficiency gap to CCGT on full load is a limitation at base load type plants. Efficiency of a simple cycle gas turbines is always below that of internal combustion engine. GE gives 42,6 % simple cycle and 56 % combined cycle efficiency for a 54 MW aeroderivative gas turbine. /11/

The huge difference between simple cycle and combined cycle gas turbine efficiencies comes from the fact that an industrial turbine's gas temperature after the turbine is very high and a lot of energy can be taken from the exhaust gases by means of a steam turbine. The exhaust gas temperature of a corresponding engine power plantis about 350 ^{o} and the possibility to increase the efficiency by utilizing exhaust gas energy is clearly smaller.

The minimum load for engines is very low (Fig. 2). At low load the burning temperature is low and burning is not as complete as at high load. Some exhaust gas components, typically hydrocarbons (HC) or carbon monoxide (CO), increase to too high levels. The minimum load for a combined cycle engine power plant (Flexicycle in Fig. 2) is around 20 %. At lower loads, steam production for a steam turbine is too low and so the power plant must be operated in single cycle. Limits shown in Fig. 2 are not absolute but more indicative. The source /9/ gives 3 % as the minimum load of engine power plants. Both emission regulations and engine and gas turbine techniques change/develop continuously.

2.3 Ramp rate limitations

The ramp rate is defined as change of output within one minute when the power station is running and connected to the network /12/. Table 1 gives typical ramp rates for different type prime movers like a single engine, a gas turbine etc.; the figures are from year 2009. Need for stabilizing power production has increased in recent years due to increase in wind and solar power production and the ramping capability of engine power plants has been improved clearly. Wärtsilä reports over 100 % ramp rates for all types of engine power plants (gas, dual-fuel, liquid fuel) /9/. It means that rapid changes can be made in a few seconds or tens of seconds depending on the type of the engine power plant.

Figure 3 shows ramp rates for two 200 MW lean-burn gas engine power plants and equal size gas turbine power plants. Operational ramp rate refers to conditions in which the power plant is running in stabilized conditions, and not in the warming-up phase. The ramp rate for a lean-burn gas engine power plant is over 100 % and the loading rate in units MW/minute is over the plants nominal power since the plant can ramp up from very low load to full load in less than a minute. Aeroderivative gas turbines can also have around 100 % ramp rate /11/. Caterpillar does not report the ramp rate for its own gas engine plants but says that once synchronized to the grid, the plants can increase their output almost immediately in response to major generation or transmission outages. /13/

A fast ramp rate is clearly a strong point in favour of the engine power plants. Engine power plants have at best excellent ramping rates, comparable to that of hydro power.

Table 1. Maximum ramp rates of different type power plants. /12/

Prime mover	Max ramp rate (%/min)	range (%)
Hydro power	40	0-100
Diesel engines		
- emergency	100	0-100
- normal	6	0-100
Gas engines		
- emergency	20	40-100
- normal	6	40-100
Aero derivative gas turbines		
- emergency	20	40-100
- normal	6	40-100
Industrial gas turbines	8	40-100
Steam turbines		
- oil-fired	5-10	40-90
- coal-fired	2-4	60-90
- lignite-fired	1-2	60-90
Nuclear plant	1-5	60-90

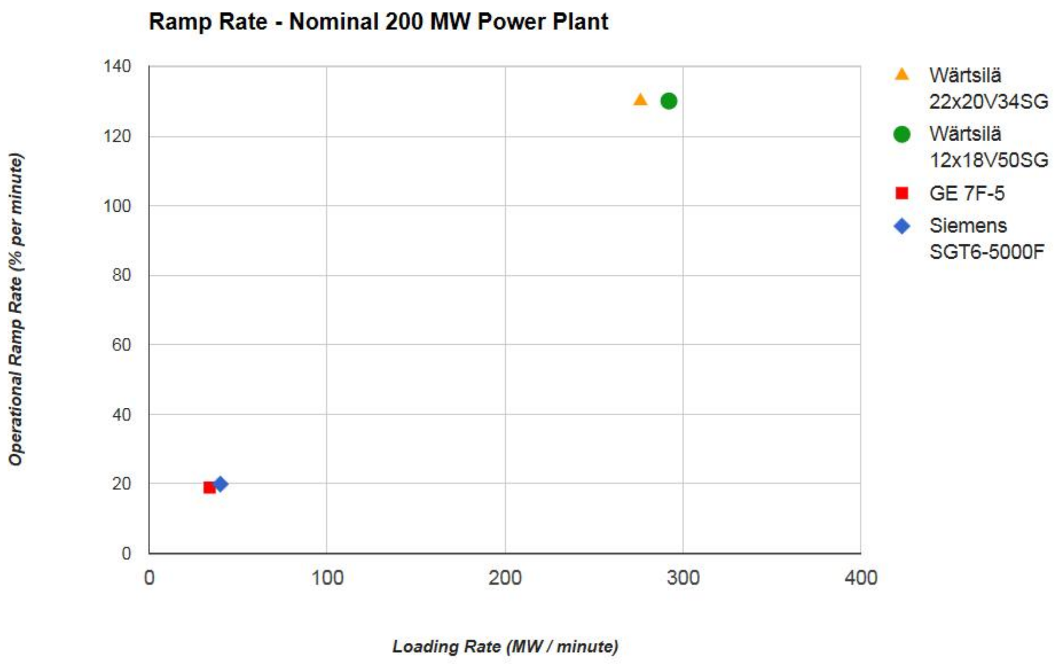


Figure 3. Operational ramp rates and loading rates of some 200 MW engine power plants and industrial gas turbine power plants. /14/

2.4 Start-up limitations

Engine power plants can be started and run to full load within minutes /15, 13, 16, 17/. The time needed depends on the type of the power plant technology and preconditions (hot start or cold start). Accordning to Wärtsilä, hot start means that cooling water is preheated and kept continuously at certain temperature level, lubrication system circulates oil and thus keeps the engine and the generator bearings prelubricated and the engine is cycling slowly. /15/

A lean burn gas engine power plant can reach full load in two minutes under hot start conditions (Fig. 4). Under cold start up conditions the Wärtsilä 34SG power plant can reach full load in 10 minutes and the 50SG in 12 minutes. /15/ Small emergency diesel generators using diesel fuel or light fuel oil as fuel can reach full load fast but the heavy/more viscous fuel might need some preparation time before start.

Restart after shut down can be conducted shortly after shut down. For example for the lean-burn gas engine 50SG shut down from stop signal takes roughly 3 minutes and after another 3 minutes the genset is ready to start again /18/.

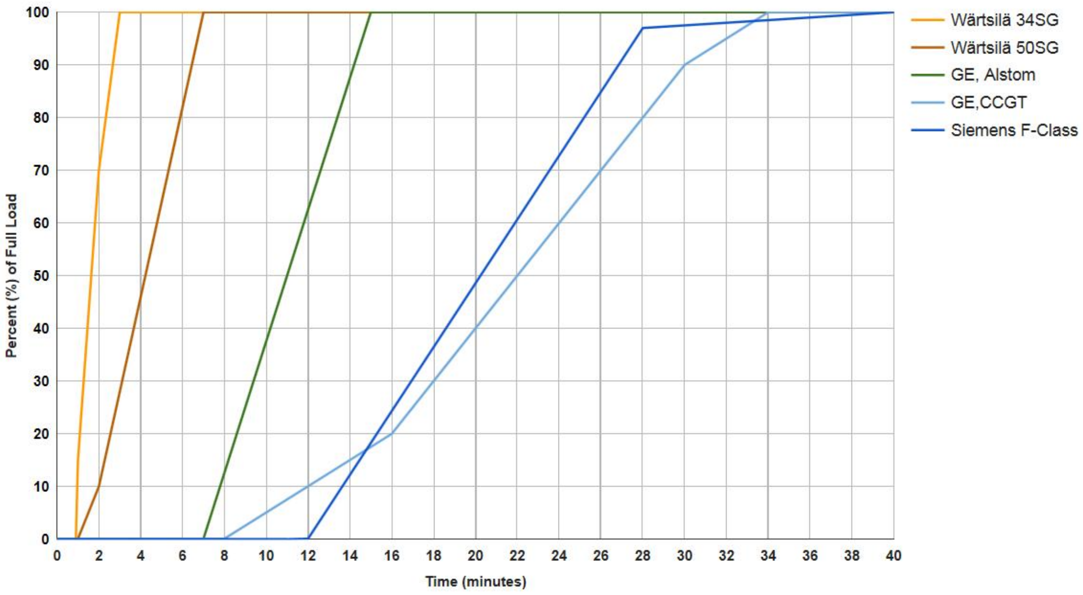


Figure 4. Start up times of some engine and gas turbine power plants. Yellow line: Wärtsilä 34SG power plant and brown line Wärtsilä 50SG power plant both under hot start conditions: 70º cooling water temp; prelubrication of the engine and the generator bearings. Green line: Simple cycle industrial (heavy duty) gas turbine under hot start conditions: GE,Alstom. Light blue line: GE FlexEfficiency CCGT under hot start conditions: purge credit; Rapid Response; start up within 8 hours of shutdown. Dark blue line: Siemens F-class CCGT under hot start conditions: auxiliary steam, stack dampers maintain HRSG temperature and pressure. /15/

3 Boiler-based power plants

3.1 General

In boiler-based power plants, the energy contained in the fuels is converted to heat in a combustion process inside the boiler furnace. The heat is transferred to water to produce superheated steam for the steam turbine that converts the thermal energy to mechanical energy that is converted further to electricity by the generator. Especially in the north, a big share of the boiler-based production units are cogeneration plants (CHP, Combined Heat and Power) that utilize low-grade heat for district heating and process steam. An auxiliary condenser is sometimes added to CHP plants to generate artificial thermal load for the boiler to reduce the minimum supplied district heating load and/or during part load operation to increase electricity production. The extra heat is typically dumped into a lake or the sea.

The steam cycle of a boiler-based power plant is fairly similar for all the different boiler types and it is the predominant part of the power plant. Thus in the following the description of boiler-based production starts with the steam cycle. Several types of furnaces are used in the boilers. Since the flexibility limitations of the different processes significantly vary, the different boiler furnace types are described separately in the following sections. After the steam cycle and the different furnace types, the operation and limitations of the turbine and control approaches used in boiler-based power plants are discussed.

A boiler can convert the fuel bound chemical energy to thermal energy with an efficiency that is at maximum in the order of 90%. Energy contained in the flue gases causes the biggest energy loss. This loss increases at low loads when it is usually necessary to operate at higher air to fuel ratio to guarantee good mixing in the furnace. This leads to a bigger amount of flue gases per the amount of burned fuel. Other major losses are due to the loss of combustible matter (unburned carbon) with bottom and fly ash, heat contained in removed ash, heat transfer through boiler walls to the surroundings, and the energy used to run the boiler (mills, fans, pumps, conveyors etc.). At high loads the efficiency also drops somewhat. One reason for this is the increase in the energy used for boiler operation (mainly by pumps and fans). Another reason can be that the heat transfer surfaces are at full load not capable of transferring all heat power from the combustion gases to the water/steam cycle. Also design of the water preheating arrangement affects the efficiency at different loads.

In utility power plants, steam turbines and generators convert thermal energy into electricity. Typically in the order of 40% of fuel power can be converted to electricity. The maximum efficiency at which fuel energy can be converted to electricity is restricted by laws of thermodynamics. In practice the maximum share of electricity at different loads depends also on the efficiency of the turbine and the generator. By increasing the steam temperature, power plants with a somewhat higher share of produced electricity can be built. In these boilers, more expensive materials need to be used in the pipes to allow the increase in temperature and pressure. In CHP plants in which the low-grade (i.e. low temperature) heat is used e.g. for district heating, typically 60-95% of the fuel energy can be utilized as electricity and heat. Electricity price and demand of heat determine in practice how the CHP plant is operated and what are the actual shares of produced heat and electricity.

3.2 Steam cycle

In steam boilers, water is commonly used as the heat transfer medium because of its cheap price, good availability, and suitable thermodynamic properties such as specific heat capacity and latent heat. There are also other mediums used such as different type of oils. However, the use of those so-called organic cycle processes is still quite rare and for this reason they are excluded from this report.

The thermal cycle utilized in steam power plants is the so-called Rankine cycle. The simplified water/steam cycle of the power plant process consists of a steam generator (boiler), steam turbine, condenser, and a return pump. The operation of thermal cycles is typically described with *Ts* or *hs* diagrams, where *T* is water/steam temperature, *s* is entropy and *h* is enthalpy. An ideal Rankine cycle is presented in Fig. 5. In the first step, water is compressed to a boiler pressure by a feed water pump, 5 -> 6 in Fig. 5. In ideal (isentropic) compression, work entropy remains constant and temperature is increased. After compression water is heated in preheaters and evaporator to the boiling point, 6 -> 7. During the boiling process in constant pressure, temperature remains constant and entropy increases, 7 -> 8. After all water is converted to saturated steam, it starts to superheat, 8 -> 1. At point 1, live steam has reached its nominal temperature at boiler pressure. After that live steam is expanded in a high-pressure turbine. In an ideal (isentropic) expansion, entropy remains constant, pressure and temperature drops, 1 -> 2. After the first stage expansion, steam is returned back to the boiler to be reheated. At constant pressure, steam temperature and entropy are increased to values at point 3. In the second expansion stage, steam pressure and temperature are dropped to the level at a condenser, 3 -> 4, and steam is condensed back to liquid phase. Next condensate is pumped back to the boiler and the cycle is closed, 4 -> 5.

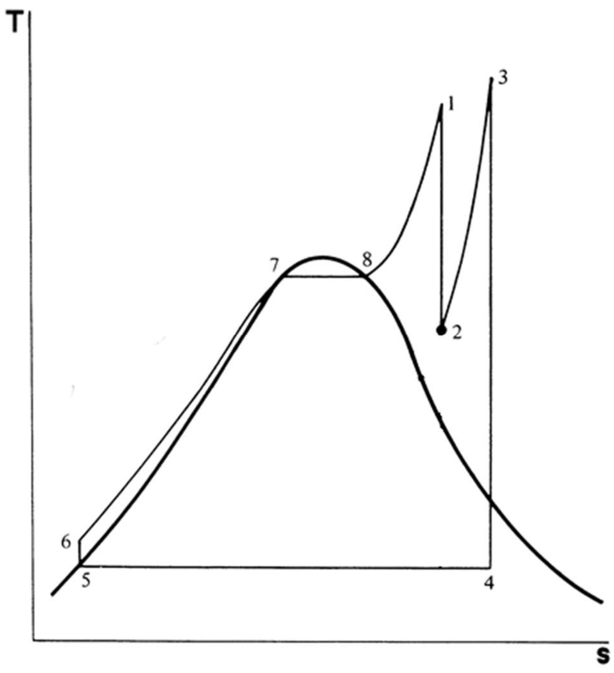


Figure 5. Ts diagram of an ideal Ranking cycle.

3.2.1 General description

Steam boilers are classified according to the structure of a water/steam system to drum boilers and once through boilers. In drum boilers, the water/steam process is divided into two separate parts: evaporation and superheating. A mixture of saturated water and steam circulates in the evaporation loop consisting of a drum, downcomers and riser tubes covering furnace walls. Water and steam are separated from each other in the drum. Saturated water with added feed water keeps on circulating in the evaporator loop and saturated steam is led to the super heating section to be heated up to the final live steam temperature. The simplified structure of the water/steam system of a drum boiler is depicted in Fig. 6. After the first turbine stage (high-pressure stage), steam is returned back to the boiler to be reheated before entering the further turbine stages. Reheating improves the thermal efficiency of the turbine process and it also reduces water content of moist steam in the low-pressure end of the turbine. This protects rotor blades in a low-pressure section of the turbine from water drop caused erosion. In some combined heat and power (CHP) plants reheaters are not applied, because the outlet pressure of the steam turbine (back pressure) is designed to be so high that this moist steam problem does not exist. Extracted steam from the turbine is also used for preheating of condensate and feed water in the feed water tank and line after a feed water pump.

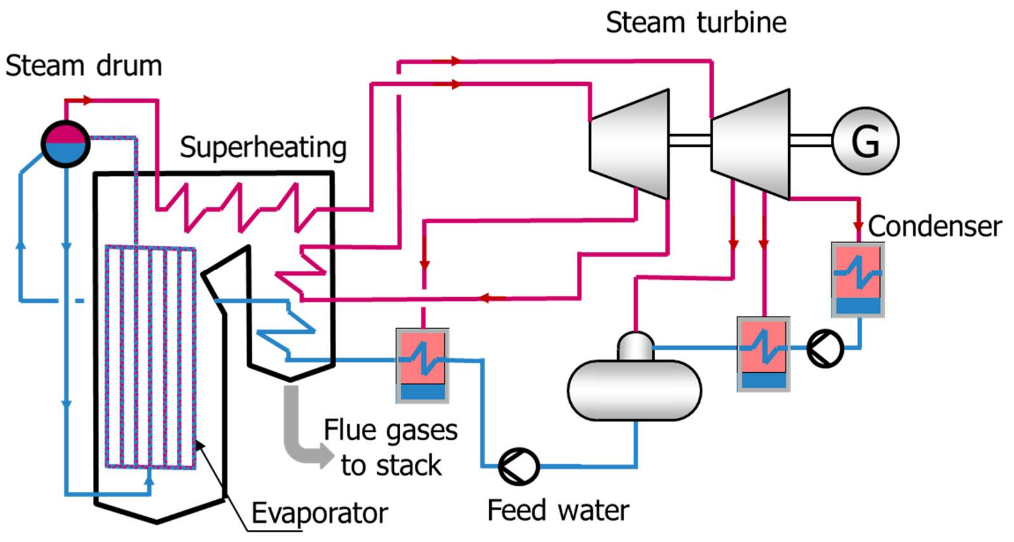


Figure 6. A simplified structure and connections of the water/steam system of a drum boiler and a steam turbine.

In once through boilers, there is no evaporator loop, but water is flowing straight forward through the boiler tubes first warming up to the boiling point, then vaporizing, and finally superheating. There are two types of once through boilers, Benson type and Sulzer type. In a Sulzer type boiler, vaporization and superheating zones are separated from each other by so-called Sulzer bottle. The role of the Sulzer bottle is a little bit similar to a steam drum, to separate water from steam, but in Sulzer boiler the saturated water is not circulated back to the evaporator but it is blown out. Proportion of water in water/steam mixture at the Sulzer bottle inlet is typically 2-4 % of the total mass flow while in drum boiler it is typically about 90%. In the Benson boiler, there is no separation of water and steam, but all the water is flowing through the boiler from preheating economizer to final superheaters.

Sulzer and Benson type once through boilers are operated with different strategies. In Sulzer boilers, evaporator and superheating zones have the fixed separation point, the Sulzer bottle. Fuel and water feeds are adjusted so that app. 96 – 98 % of feed water is vaporized before the Sulzer bottle. In Benson boilers, vaporization and superheating zones are floating according to a boiler load. This floating is very useful especially at part load operation, because the shares of fuel powers directed to vaporization and superheating can be affected by feed water flow, i.e. at which point in a heat exchanger all water is vaporized and superheating will start. In boilers with fixed heat exchange surfaces for evaporation and superheating (drum and Sulzer) there is a common problem to maintain the nominal live steam temperature at part load operation. This is very sensitive for correct dimensioning of heat exchangers and properties of applied fuels. The basic structure of once through boilers is depicted in Fig. 7.

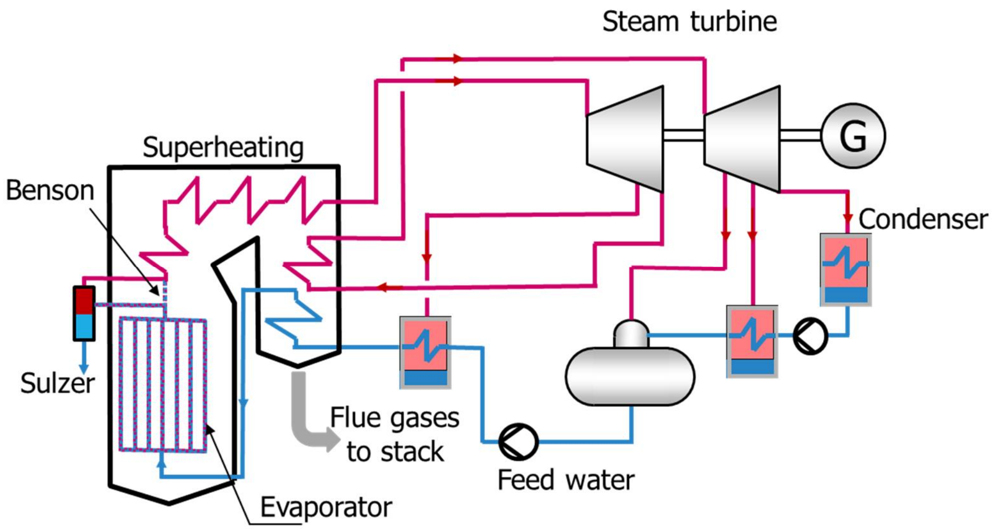


Figure 7. A simplified structure and connections of the water/steam system of a once through boiler and a steam turbine. In the Benson boiler water/steam is flowing directly from evaporation to superheating (dashed line) while in Sulzer boiler it flows through the Sulzer bottle, where remaining water is separated from saturated steam.

Benson type boilers are more commonly used because of their simpler and cheaper structure. Sulzer type boilers are applied mainly when there is expected to be problems with the boiler water quality. In Sulzer type boilers, impurities like dissolved lime and other minerals in feed water are blown out so that they won't be collected in the superheaters and finally plugging the lines.

Another possibility to classify boilers is according to the operation pressure. Boilers can be classified to sub- and super critical boilers depending on if the boiler is operated below or above the critical point of water (647 K and 22.064 MPa). All super critical boilers are once through type, because when operating at super critical area water doesn't have different phases (liquid, vapor), but it is a supercritical fluid which is heated up to the devoted temperature. Thus the drum cannot be used to separate steam from water, because those different phases do not exist. The benefit for using super critical operation range is the improved thermal efficiency and higher power output compared to physically same size sub-critical boiler. In the point of view of operation there are no big differences if the once through boiler is sub- or supercritical.

3.2.2 Ramp rate limitations

Load change rate of the boiler depends on the fuel handling system, combustion method and water/steam system. The dynamic response of the combustion system, i.e. how fast fuel bound chemical energy can be converted to thermal energy, depends on

- fuel properties
    - reactivity
    - proportion of volatile components
    - effective reaction area
    - particle size

    - moisture
- fuel preparation and feeding system
    - drying
    - milling -> particle size
    - direct mill combustion (slow mill dynamics included) or feeding combustion ready fuel from a storage (fast response)
    - delays in feeding system
- applied combustion method (discussed in Sections 3.3-3.6)
    - fast: pulverized fuel combustion, oil and gas firing
    - medium fast: bubbling and circulating fluidized beds
    - slow: grate firing.

Dynamic properties of the water/steam system are based on heat transfer capacity and the amount of energy stored in boiler structures. Heat transfer capacity of a heat exchanger is proportional to temperature difference across the heat transfer surfaces and flow rates of mediums in the primary and secondary sides. Thus, if we want to change heat power transferred by a heat exchanger, we have to change either temperatures, flow rates, or both.

As a result of an increased power output from a boiler, gas flows inside the furnace and water/steam flows will be increased. Temperature of the saturated water/steam mixture in the evaporator does not change when the boiler is operated at constant pressure. Actually there is a small change in drum pressure, because the controlled pressure is live steam pressure and pressure loss across the superheaters is a function of steam flow. Thus with higher loads (steam flows) drum pressure is also a little bit higher. However, in this case the pressure related temperature change is small.

Adiabatic flame temperature does not change as a function of fuel power, thus radiative heat transfer from flames to heat exchanger surfaces remains constant. With increased load, gas temperatures in the furnace tend to rise because the heat absorption capacity of furnace walls does not increase directly proportional to released fuel power. This means that with higher loads a bigger share of released fuel power is directed to superheaters, and superheated steam temperature increases. As a result, with increasing load also spray water flows in the attemperators cooling down the controlled steam temperature are increased compensating the decreased share of evaporation in the furnace.

Dynamic properties of the water/steam system consist of two parts: transfer of heat power from furnace radiation and hot flue gases to water/steam flow and behavior of energy stocks stored in heat exchanger structures. From hot flue gases, thermal energy is first transferred to the outer surface of metal tube walls by radiative and convective heat transfer. After that, the heat power is conducted through the tube walls, and finally from inner tube surface to water/steam flow by convection. Convective and radiative heat transfer between medium and tube surfaces is fast compared with conductive heat transfer through tube walls. Thus, the thicker tube walls and the bigger steel mass in the heat exchanger the slower the heat transfer dynamics from boiler gas side to water/steam side. Heat transfer efficiency from hot flue gases to heat exchanger structures depends, among other things, on a cleanliness of heat transfer surfaces. The more there is soot and ash on the surfaces the slower and more ineffective the heat transfer efficiency is. Thus heat transfer dynamics in the boiler is not constant, but it varies as a function of time.

The hot steel mass of water/steam system structures is an energy storage which can be utilized during transient operation. Boiling point of water is a function of pressure. The boiling point is increased along with the pressure until the system pressure reaches the critical point. Above this point boiling temperature cannot be defined because no boiling occurs any more.

In fixed pressure drum boilers, steam pressure can be either stabilized accurately to the set point value (turbine follows control structure, known also as turbine front pressure control) or it can be allowed to fluctuate between predefined limits (boiler follows control structure). Principles of these different control structures are explained in Section 3.8. If the boiler pressure is allowed to fluctuate along with a load change, it is possible to improve the momentary load change rate by utilizing the stored energy inside the boiler. In case of increasing steam flow, outflow from the boiler can be controlled to be bigger than the generated steam flow resulting in a pressure drop in the boiler. As a result of the pressure drop also boiling point of saturated water in the evaporator is dropped causing additional generation of saturated steam. The latent heat needed for additional evaporating is extracted from heat exchanger tubes. As a result, also temperature will drop and the system will set down to a new equilibrium state. This momentary additional steam generation capacity is called storage capacity of a boiler, C_e. It is defined as how much extra steam power a boiler can generate as a function of unit pressure drop.

$$C_e = \frac{\int_{t_1}^{t_2}[h(t)q_m(t) - h(t_1)q_m(t_1)]dt}{p(t_2) - p(t_1)}$$

where h is steam enthalpy, q_m is steam mass flow and p is steam pressure. How much extra steam will be generated depends on how big pressure and temperature changes are allowed.

Pressure change is not so critical for boiler structures. However, it may lead to problems with drum level control because of the shrink and swell effect taking place in the drum. This means that when steam flow from the boiler increases and the actual amount of stored water/steam mixture in the evaporator and drum consequently decreases, at first the measured mixture level in the drum increases due to the pressure drop in the drum. Pressure drop related additional evaporation of saturated water increases the specific volume of water-steam mixture raising the drum level. Just after a while the drum level turns to go down due to decreasing water/steam storage in the evaporator and drum.

This is because the pressure drop caused by increased outlet flow will vaporize saturated water in riser tubes and drum causing level rise in the drum, drum water is swelling. In case when feed water flow into the boiler is increased, the subcooled feed water will cool the saturated water/steam mixture and saturated steam bubbles will condensate to liquid and water/steam mixture shrinks resulting to level drop in the drum. This so called non minimum phase response makes normal feedback control very difficult and ineffective, because the controller must be tuned very slow so that it won't react to this first wrong direction response of the controlled variable. A commonly used three element control structure for drum level control is introduced in Section 3.8.

Temperature change caused by pressure change in the boiler is more critical. Pressure change originated temperature transient in the evaporator and steam drum must be kept within permissible limits. Steam drum is a thick walled pressure vessel and it is more sensitive to temperature transients than evaporator tubes. Calculation methods for maximum allowed temperature gradients for a steam drum are defined in a standard EN12952. When utilizing boiler's storage capacity in load tracking control, it is important that the maximum allowed gradients are not exceeded.

Once through boilers are typically operated at sliding pressure mode, which means that steam pressure is increased along with the load. Sliding pressure mode saves pumping costs, because at partial loads less pumping power is needed. In pure sliding pressure mode control valve of the steam turbine is set to be 100% open. Thus boiler pressure cannot be manipulated by the turbine control valve. In this case the thermal storages of boiler structures cannot be utilized, but the load rate change depends only on the combustion and heat transfer dynamics from the furnace side to the water/steam side of the boiler. Another option for sliding pressure operation is so-called supervised sliding pressure operation. This means that during a steady state operation turbine valve is operated app. at 90% position, so that live steam pressure is throttled a little before the turbine inlet. In case of a rapid load increase the valve will be quickly opened to 100% position, which gives a temporary boost to power output of the boiler. The size of the boost is smaller compared to the utilization of the storage capacity in fixed pressure drum boilers. The size of the boosting effect depends on how much the turbine valve is throttled. The more throttling, the more boost for temporary load change, but this will increase pumping cost in steady state operation.

As a conclusion of the load change rate of the water/steam system it can be stated that most probably the limits will come from boiler control side, not from structural limitations. Actuators for fuel feeding, air fans, air distribution, and flue gas removal should be improved. Also combustion control, feed water control and steam temperature control should be to a high enough level to be able to keep the process variables, emissions and operation efficiency in the required ranges.

3.2.3 Minimum load limitations

For the water/steam system at ultra-low loads there are two essential issues:

- to maintain sufficient flow conditions at every part of the heat exchangers to avoid overheating and material damages.
- to maintain steam temperature at acceptable level.

In natural circulation drum boilers, the steam fraction in riser tubes may turn to be very small causing a problem to maintain a sufficient pressure difference between the downcomers and risers for an adequate flow rate. A solution for the problem is to use additional circulation pump to guarantee a sufficient circulation flow in the vaporizer. This type of forced circulation water-tube boilers are called LaMont boilers. In circulating fluidized bed boilers, ultra-low loads may raise problems in the water circulation of cooled cyclones separating flue gases from solids, and also the heat exchangers located inside the sand bed in the return leg may be exposed to overheating problems.

In once through boilers ultra-low loads with low flow rates may also give rise to problems in furnace walls. Typically, riser tubes in once through boilers are dimensioned for high flow velocities (small diameter, high pressure losses), which should guarantee suitable flow distribution in different parts of the wall tubes. However, at ultra-low loads water distribution to different wall panels may be disturbed causing uneven temperature distribution and thermal stresses to wall elements.

In the superheating section of the drum boilers the expected problem is that at ultra-low loads steam temperature will remain far below the nominal value. In the boiler side this is not a problem but this will cause severe problems on the turbine side. If this type of operation is really needed in future, one solution may be some kind of additional superheating e.g. with electric heaters. In Benson type once through boilers it is easier to maintain the accepted steam temperature also at low loads because of the floating share of evaporation and superheating areas in the furnace. Thus at ultra-low loads the effective heat transfer surface for superheating can be increased to raise steam temperature to accepted level.

3.2.4 Startup limitations

Startup time of the water/steam system of the boiler is constrained by temperature gradients and differentials in thick-walled structures such as the steam drum and headers of the superheaters. Also T-shaped pipe joints are sensitive for thermal stresses. During a startup, temperature gradients are typically restricted to 50-80°C/h. This results that rising time from cold start point to 16 MPa pressure is app. 5 hours including that drum differential and superheters / reheaters metal temperarture limits are not exceeded. The saturated steam temperature rise is fixed not only to ensure the integrity of thick-walled components such as drum or headers, but also to limit the expansion rate of boiler pressure parts as a whole, so that differentials between those parts and headers, skin casings or support arrangements cannot result in torn welds or similar damage. During pressure rising the reheaters are particularly vulnerable, as they see no steam flow at all until the turbine is running. /29/ Figure 8 illustrates typical startup curves during a hot start of a drum boiler.

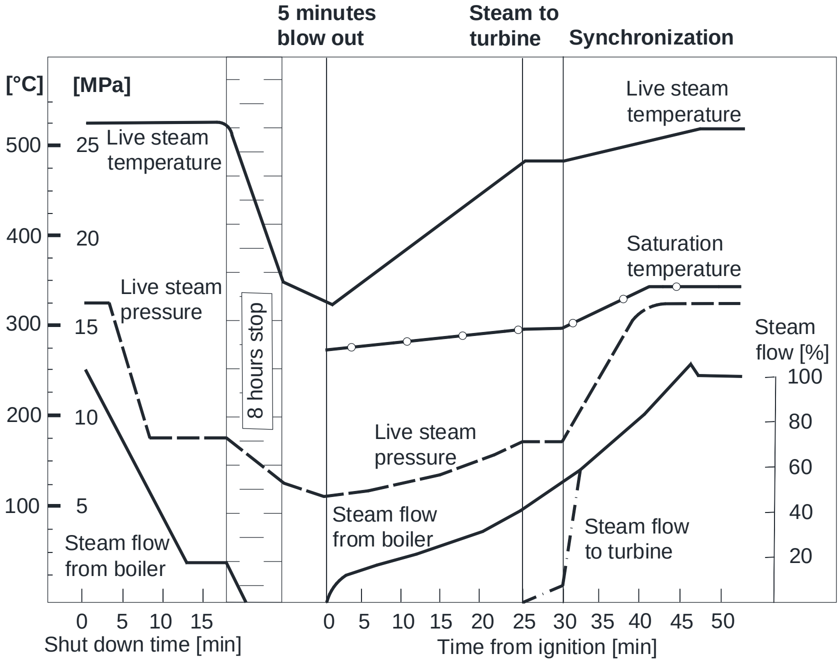


Figure 8. A typical start up diagram for a hot start of a drum boiler. Modified from /19/

Another issue possibly extending the startup time of the boiler is the operation of the condensate return system. It is common that during the startups purity requirements set for the returned condensate are not fulfilled due to impurities disengaged from the condensate lines and collector tanks. Makeup water production and reservoir tanks are not designed to produce enough makeup water for the boiler if the condensate return is not workin properly. Thus boiler loading will be delayed due to lack of boiler water until the condensate system is working properly.

3.3 CFB boiler

3.3.1 Overview of the technology

Process description

Circulating fluidized bed (CFB) boilers are used for conversion of a wide range of solid fuels into heat, process steam and/or power. Combustion takes place in the combustion chamber that in a CFB is also called a riser. Primary air is fed through air distributor nozzles located at the riser bottom. To reduce NOx emissions, the oxygen content in the lowermost part of the furnace is kept low and secondary air streams are fed through nozzles at furnace walls at higher elevations to complete combustion. Solid particles are carried up by the gas flow. Major part of the lifted particles move, as they rise, towards the walls and fall down along the walls while a smaller solids fraction follows the flue gas out from the combustion chamber into a cyclone. In the cyclone, the bulk of the particles are separated from the gas flow and returned to the bottom of the riser through a loop seal. The amount of circulating solid material is strongly dependent on the boiler load. At low load only a small amount of solids reach the cyclone. A small fraction of fine ash particles that also contain small amounts of unburned carbon follow the flue gas from the cyclone and need to be separated into a fly ash fraction in electrostatic precipitators or bag house filters. The solid mass in the furnace constitutes of ash, sorbents for SO2 removal (if a fuel with high S content is burned), solid fuel at different stages of conversion and makeup sand. Sand can be added if the ash content of the fuel is low and not enough solid particles otherwise enter the furnace with the fuel. Also if the fuel has high tendency to form agglomerates, bed material can be diluted by adding makeup sand to prevent this.

The vigorous turbulent movement of gas and solids enhance mixing and heat transfer and produce a fairly uniform temperature distribution in the furnace. The temperature is typically in the range 750 C to 950 C which guarantees that no NOx is formed from the nitrogen of the combustion air. At full load, uniform temperature is easy to maintain while at lower load the temperature at furnace bottom tends to increase and the temperature at riser top decrease if the fuel is coal. Temperature distribution can be influenced by recirculating cooled flue gases into the furnace. The bottom section of the furnace is refractory lined while the walls elsewhere in the combustion chamber are covered by water/steam pipes that collect the heat produced in the furnace. Additional heat transfer elements can be placed inside the central and upper parts of the furnace, in the loop seal and in the second pass through which the air exiting from the cyclone travels downwards. An air preheater (luvo) is typically located in the second pass. Schematic of Valmet CFB is shown in Fig. 9.

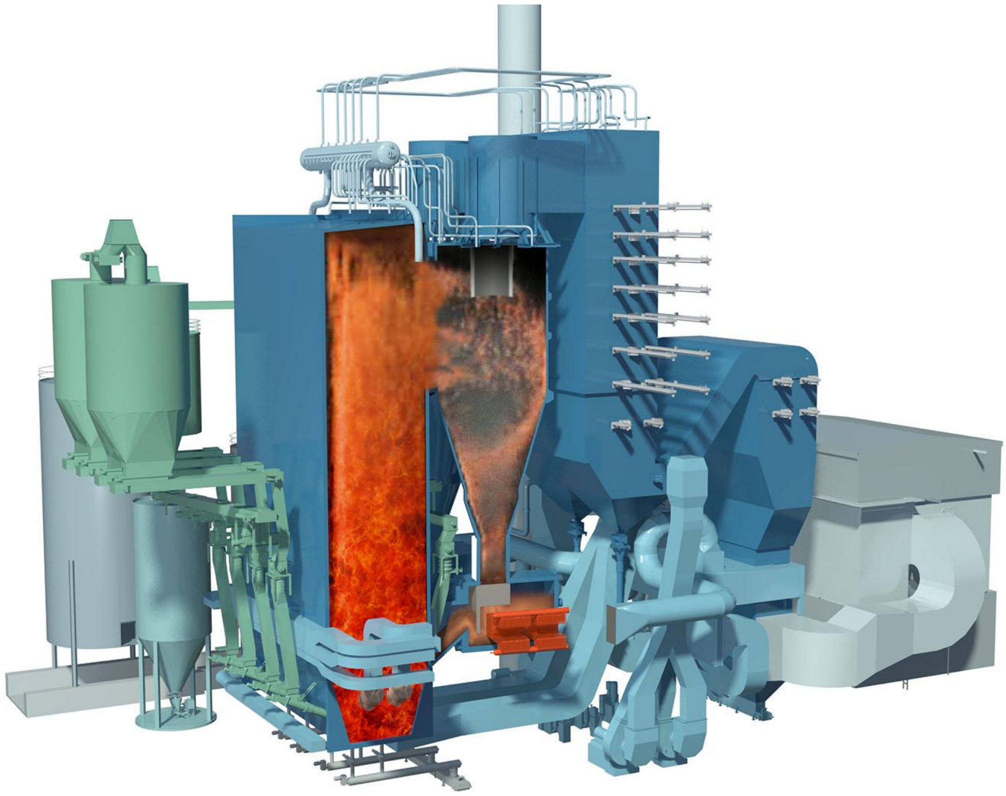


Figure 9. Schematic of Valmet CFB boiler.

Fuel flexibility

The big advantage of fluidized bed combustion is the large range of fuels that can be burned. Coals of even very low quality (ash content up to 60%) can be burned without major problems. In the case of biofuels, waste derived fuels and slurries, the limiting factor is the composition of the ash. Occurrence of corrosion, fouling and agglomeration of the bed material can prevent utilization of otherwise usable solid fuel fractions. In many cases problematic fuels can be burned mixed with another fuel or by using suitable additives that react with the harmful components or otherwise reduce their effects/20/.

CFB boilers are always designed for a specific fuel or for a range of fuels. Widening the range can increase the required investment costs. During the life time of a boiler it can be necessary to change the fuel mix several times due to changes in fuel prices or availability. The main fuel properties affecting boiler design and operation are the heating value and the moisture content of the fuel. In a boiler designed for combustion of a wide variety of fuels, boiler size is determined by the fuel forming the maximum amount of flue gases, i.e. the fuel with the highest moisture content and lowest heating value. This is to limit the maximum velocities in the boiler to prevent erosion and erosion-corrosion. Another alternative is to limit the maximum boiler load to a lower level when moist fuel with low heating value is burned. The fuel with lowest heating value also determines the fan capacity and the design of fuel feeding systems while the fuel with highest heating value determines the required flue gas recirculation system. Different fuels may also require separate feeding arrangements which increases costs. The selection of fuels also affects emissions but usually the effects can be handled.

Market potential

CFB’s have increased their popularity in the utility market. The increased maximum boiler size, up to 800 MWe, and increased efficiency of the modern supercritical boilers have contributed to this trend. The possibility to use lower quality coals and to add a significant share of other fuels in the fuel mix is a significant benefit of CFB technology over pulverized coal boilers. The investment cost for a CFB is often slightly higher (of the order of less than 5-10% higher) than for a PC boiler but over the life time the lower fuel prizes renders CFBs competitive. Moreover, separate flue gas treatment units required by PC plants but not by CFB boilers can increase the total prize of a PC boiler so that it exceeds that of a CFB. A large fraction of CFB boilers are smaller units used in industries to produce electricity and steam and in energy companies for district heating and power production. Investment cost of CFB units is higher than for grate and bubbling fluidized bed (BFB) furnaces which somewhat limits using CFBs in small power plants. Below the power range 20-30 MWth CFBs are uncommon while above 100 MWth BFBs are more rare. In BFBs, coal cannot be used as the fuel which also reduces the use of BFBs in boilers in the higher power range.

3.3.2 Operation at reduced load

Technology

At lowered loads, the fuel feed rate is reduced to match the load demand. The amount of combustion gas is also reduced simultaneously, but usually not exactly in the same proportion. The minimum allowable gas flow rate through the air distributor at the bottom of the fluidized bed is limited by the fact that the solid materials in the furnace need to be kept in fluidized state by blowing a large enough gas stream through the air distributor at riser bottom. Moreover, in a CFB it is desirable that significant amounts of solids rise to upper parts of the riser and enter the cyclone. To achieve this even higher gas velocities are necessary.

A further demand on the gas flow rate is set by the air distributor. Uniform distribution of fluidizing gas across furnace cross-section is obtained only when the pressure drop across the air distributor is of the same order as the weight of the bed mass. At lower gas fluxes channeling and unstable operation would occur. The pressure drop over the air distributor increases in proportion to the air flow rate in power of two. Simultaneously increases also the power consumption of the fan that provides the air to the boiler. Thus the air distributor needs to be designed for a range of gas velocities such that the power consumption of the fan stays reasonably low at full load and the bed is sufficiently uniformly fluidized at minimum load.

Typically, at low loads the amount of primary air is kept more or less unchanged from full load while the amount of secondary air is reduced. To achieve even lower loads, flue gas re-circulation needs to be used to increase the gas flow rate through the air distributor. This will increase investment and operation costs. Increased flue gas recirculation can lead to lower bed temperature and to prevent that a thicker layer of refractory can be necessary. Furthermore, a large load range of a boiler needs to be taken into account in the dimensioning of the steam pipes. At low loads the operation can become unstable and improved control may be necessary. If the superheater is located in the solids circulation loop (typical for biomass and waste firing CFBs), steam temperature will reduce drastically at low loads which sets its requirements on the turbine. Commonly in old CFB boilers the minimum load was of the order 40% of full load while in new boilers it can be of the order of 30%. Today there would be a need to reduce the minimum load further to 15-20%.

Effects on emissions

NOx emission typically increases at low and high end of boiler loads. Use of SNCR (selective non-catalytic reduction of NOx) system evens out the variation /21/. At low load the possibilities for staged combustion are reduced and commonly the oxygen content in the bottom region is much higher than at full load. This leads to significantly higher NOx emissions unless SNCR is used. SNCR is based on injecting an ammonia solution into the furnace. NH_3 reacts with NOx and produces N_2. The location for the spray needs to be selected such that the temperature is in a suitable range. In addition, oxygen is needed for the reaction to take place. In a boiler with a wide load range, finding a location for SNCR injection such that reduction of NOx is achieved at all loads can be difficult. Different locations can be used at different loads which slightly increases investment costs.

Temperatures in the furnace need to be high even at low load to ensure complete combustion. As a result, the boiler may need to be dimensioned for low loads. At high load, furnace temperature is then controlled by larger amounts of recirculation gas affecting not only the investment cost but increasing also the auxiliary power consumption in the higher load operation points. /22/

3.3.3 Ramp rate limitations

Typical maximum ramp rates for CFBs are in the range 2-6%/min (Table 2). Older CFB boilers achieve lower ramp rates than newer ones. The maximum ramp rate is affected by the amount of refractory covered area in the furnace, by the thickness of the refractory, by the amount of bed material and by fuel reactivity. Only slight increase in ramp rate can be achieved by thinner refractories but then the risk for tube leaks increases. Also an increase in combustion temperature can help to increase the ramp rate but then the risk of bed sintering increases.

Table 2. Typical ramp rates in a CFB for different fuels.

	coal	bio	waste
	ramp rate, % of MCR/min	ramp rate, % of MCR/min	ramp rate, % of MCR/min
30% to 50% load	2-4	4-8	-
50% to 90% load	3-5	4-6	1
90% to 100% load	3	3	1

3.3.4 Start-up limitations

Start-up times of CFB boilers are limited by the maximum allowable temperature change, typically 50-80°C/hour, of the refractories. A cold start takes typically 10-12 hours. Auxiliary fuel is needed during start-up which increases operation costs. Increasing number of shut-downs and start-ups will also increase the wear in the boiler. /23/

The boiler can be shut down for about 8 hours without significantly cooling down. A hot start-up after short breaks is therefore faster since heating of the boiler mass is not necessary. The steam cycle will set its limitations on the start-up time. Typical start-up times are listed in Table 3.

Table 3. Typical start-up times for CFB boilers.

Start-up type	Start-up time
Very Hot start (<2 hours shutdown)	< 1,5 h
Hot start (2 to 8 hours shutdown)	< 2,5 h
Warm start (8 to 48 hours shutdown)	3 – 6 h
Cold start (>72 hours shutdown)	10 – 12 h

3.4 BFB boiler

3.4.1 Overview of the technology

In a bubbling fluidized bed (BFB) gas velocities are lower than in CFBs. Since in a combustor, the gas flow rate is relative to the amount of burned fuel, the produced energy per cross-section is smaller in a BFB. Compared to grate furnaces, the size however is smaller, which reduces investment costs. Similar to CFBs, fuel particles are suspended in a hot bubbling bed of ash and other particulate materials (sand, limestone etc.) through which primary air is blown. The difference to CFBs is that in a BFB solid material stays to a large extent in the bottom bed and only small fuel and ash particles are entrained into to gas space above the bed, the freeboard. Temperatures in the freeboard are significantly higher than in the bed. To reduce NOx emissions, part of the air is fed higher up in the freeboard as secondary air.

Similar to CFBs, BFBs are flexible as regards to selection of fuels. BFBs are not suitable for coal type fuels with low share of volatile components since most of combustion would then take place in the bed where lateral mixing is poor and high local temperatures can occur. Typically the fuels are woody biomasses or refuge derived fuels. A benefit over grate furnaces is the possibility to reduce NOx emissions by air staging and SO2 emissions by using limestone as bed material thus avoiding separate flue gas treatment steps. Schematic of Valmet BFB is shown in Fig. 10.

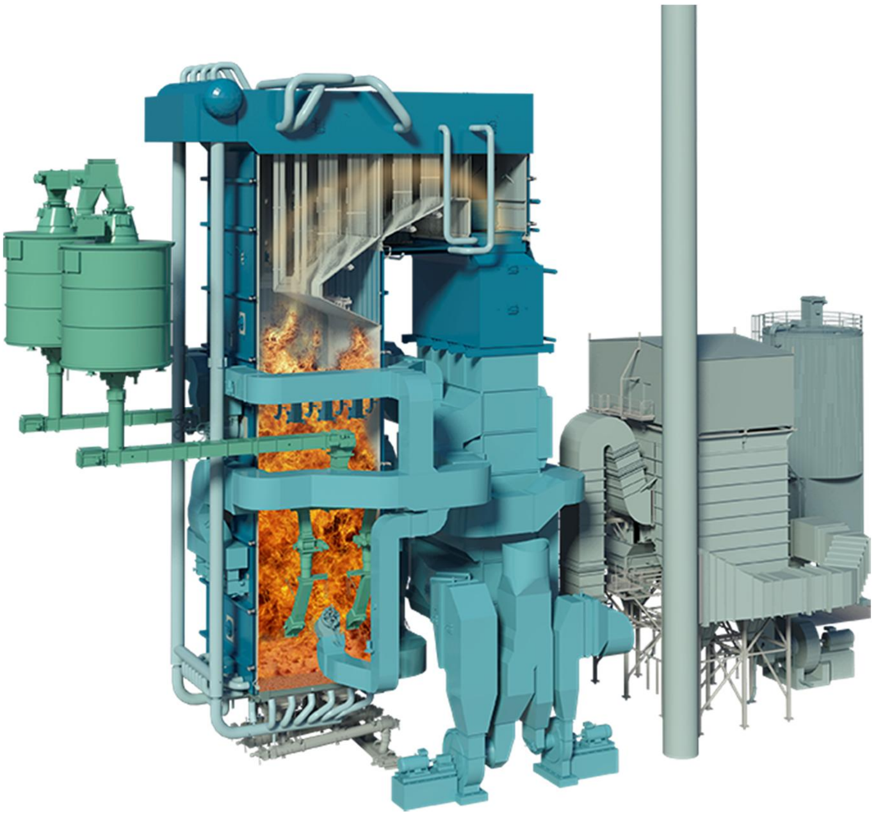


Figure 10. Schematic of Valmet BFB boiler.

3.4.2 Operation at reduced load

Some BFB boilers produce steam and electricity for industries and for such boilers operation at reduced load is not required. Most BFB boilers in industry and all BFB boilers used for district heating and electricity production, however, need to follow the load demand. In summer time when heat and power demands are small, the units can be used to produce only the required small amount of heat without generating power.

As for CFB boilers, the minimum load of a BFB boiler is limited by the need to fluidize the bed materials. Minimum loads of 30% can be quite easily achieved. In specific cases, emission limitations might limit further decreasing of the minimum load but in some cases 15% load could be achieved. Boiler efficiency reduces at reduced loads.

NOx emissions are dependent on the load and the dependency somewhat varies between different BFB units /21/. In general, a significant increase in NOx emissions is expected at very low loads.

3.4.3 Ramp rate limitations

The ramp rate is limited by the large fuel inventory in the bed, the large heat capacity of the furnace and the bed mass, and also by emission limits. Typical ramp rates in current BFBs are presented in Table 4. As shown by the table, typically BFBs allow faster ramp rates than CFBs. This is due to the smaller heat capacity of BFBs and partly also due to typically burning more reactive fuels in BFBs.

Table 4. Typical ramp rates in BFBs.

Load range	**Ramp rates,** % of MCR / min
30% to 50% load	2 – 5
50% to 90% load	4 – 8
90% to 100% load	5 – 10

3.4.4 Start-up limitations

As for CFB's the refractories of BFB boilers need to be heated up slowly. Hot start-up is consequently faster than cold start-up. Additional fuel is burned in start-up burners to warm up the boiler during start-up. The additional fuel is the main contributor to start-up costs, in addition to increased wear due to temperature changes. The start-up times are given in Table 5.

Table 5. The start-up times for BFB boilers.

Start-up type	**Start-up time**
Very hot start (<2 hours shutdown)	< 1 h
Cold start (>72 hours shutdown)	8 – 10 h

3.5 PC boiler

3.5.1 Overview of the technology

A pulverized coal (PC) boiler generates thermal energy by burning pulverized coal. Power plants based on subcritical PC boilers operate at about 35-37% efficiency, supercritical at about 40% and ultra-supercritical in the 42-45% range. The maximum efficiencies achievable with lower grade and lower rank coals will be somewhat less in all cases. Net efficiencies of 45-47% are achievable with supercritical steam using bituminous coals and currently developed furnace materials.

The ash content of the coal burned in PC furnaces needs to be low which increases the fuel price. Also peat is used as a fuel and powdered woody biomass has been used as part of the fuel mix. PC boilers dominate the electric power industry although CFB boilers have started to increase their share especially in the range under 500 MWe. PC boilers can also be used for combined heat and power production. PC boilers have been built for the range between 50 and 1300 MWe. /24/

In a PC boiler, pulverized coal is blown with part of the combustion air into the furnace through several burner nozzles. Part of combustion air can be added as secondary and tertiary air in the burner and as over-fire air higher above to reduce NOx emissions. The positioning of the burners varies. Wall-mounted burners can be located on one side, on opposite walls or tangentially in the corners or on the walls. Residence time of fuel can be affected by changing the inclination of the burners. Changing the inclination also affects the division into evaporator and super heater loads. Combustion takes place at temperatures from 1300-1700°C. The temperature correlates with coal rank. /24/

In the typical heat transfer surface layout, the furnace chamber is topped by heat transfer tubing. After passing the tubing, flue gases then turn 180°, and pass downwards through the main heat transfer and economizer sections. Another less common design is a tower boiler, where the heat transfer sections are located vertically above each other, over the combustion chamber. /24/

Furnace dimensions are selected such that the flue gas temperature before the first superheater is below the melting point of the fuel ash. The amount of flue gases also affects and thus a larger furnace size is required for low quality fuels.

3.5.2 Operation at reduced load

Minimum load of a PC boiler depends on the boiler type and burner arrangement. Fairly low, e.g. in the order of 25%, loads can be achieved by shutting down part of the fuel mills and burners. The minimum load of one coal mill is limited by the coal content in the carrier (primary) air. The coal content must be high enough to guarantee safe burning. For ultra low loads (ULL), sublemtary firing (fuel oil or gas) may be needed.

3.5.3 Ramp rate limitations

In a PC boiler, the fuel particles are small and burn fast, with particle residence time in the boiler typically in the range 2-5 seconds /24/. Thus fuel inventory does not limit the ramp rate. . A fast load response of 5% to 15% of the power output can be provided in few seconds by using the energy storage capacity of the steam/water cycle /25/. Quick controlling measures can be used for a limited time until the normal operating conditions are restored /25/. The most important factor that limits the ramp rates is the dynamics of the coal mills i.e. how fast the fuel can be grinded to coal pulver. Ball mills are the most common mill type. They have slower dynamics than ring rolling mills. Also the wear of the mill reduces grinding speed and old mills are slower than new ones. Typically mill dynamics can be improved by utilizing the fuel strorage capacity of the mill. When increased power output is needed, a temporary increase in the primary air flow will transport additional fuel to the furnace. The opposite procedure can be used to reduce the load. laskussa. These measures are still strictly limited since addition of too much primary air will increase the amount of unburned carbon in the ash. In addition, it can disturbe the operation state of the mill after which balancing the process can take tens of minutes.

Typical load change rates for a supercritical PC boiler power plant are reported in Table 6.

Table 6. Ramp rates for supercritical PC boiler power plant /25/

Load range	**Ramp rates** % power / min
30% to 50% load	2 – 3
50% to 90% load	4 – 8
90% to 100% load	3 – 5

3.5.4 Start-up limitations

Start-up times of PC boilers are shorter than for fluidized bed boilers since no refractory walls are used and there is no bed mass. After a nightly shutdown, a medium-large scale pulverized coal power plant can reach the minimum load (about 30%) in about 30-40 minutes, after boiler ignition, and then reach full load capacity in about 70-90 minutes /25/. Typical start-up times for a supercritical PC boiler power plant to reach full load operation are reported in Table 7.

Table 7. Start-up times for supercritical PC boiler power plant /25/

Start-up type	**Start-up time**
Very Hot start (<2 hours shutdown)	< 1 h
Hot start (2 to 8 hours shutdown)	1.5 – 2.5 h
Warm start (8 to 48 hours shutdown)	3 – 5 h
Cold start (>72 hours shutdown)	6 – 7 h

3.6 Other boilers

3.6.1 Grate furnace

Grate furnaces are typically used for combustion of biomass and waste with a thermal capacity of roughly 0.1−100 MW. In last decades, BFB boilers have taken over the market in the larger power range while in the lower range below 5 MWth and for waste incineration grate furnaces still dominate \22\. Large variations in fuel particle size and moisture content are possible which is a benefit especially in waste incineration. A low melting point of fuel ash can however be an obstacle for burning specific biofuels. Similarly fine fuel particles are not optimal for grate combustion.

In the combustion chamber, the solid fuel is fed to a grate. A primary air flow is supplied from the bottom of the fuel layer. While the grate transports the fuel through the furnace the fuel burns and only ash remains at the end of the grate. Typical fuel-feeding systems used in biomass-fired grate boilers are mechanical stokers. For biomass fuels that are heterogeneous in size and contain a relatively big mass fraction of fine particles, a spreader is used to reduce the tendency for fuel size segregation. The grate, which is at the bottom of the combustion chamber has two main functions: lengthwise transport of the fuel, and distribution of the primary air entering from beneath the grate. The grates are mainly classified into stationary sloping grates, travelling grates, reciprocating grates, and vibrating grates. In KPA Unicon's BioGrate system, the fuel is fed onto the center of a grate from below through a stoker screw. The grate consists of alternate rotating and stationary concentric rings with the rotating rings alternately rotated clockwise and counter-clockwise by hydraulics. This design distributes the fuel evenly over the entire grate.

The primary air for combustion and possibly added recirculated flue gas are fed from underneath the grate and they penetrate the fuel through the slots. In grate furnaces, primary air feed is typically divided into sections, which improves controllability of the process allowing even stable operation at low loads up to 25%. Above the fuel layer, burnout of the gases takes place and the heat is transferred to a heat exchanger. Preheated primary air can be used to enhance ignition and flue gas recirculation to control the temperature in the furnace. Similar to the other furnace types, staged combustion is used to reduce NOx emissions /26/. The secondary air is fed directly into the flame above the grate and the air distribution is controlled by dampers and speed-controlled fans. The gases released from biomass conversion on the grate and a small number of entrained fuel particles continue to combust in the freeboard, in which the secondary air supply plays a significant role in the mixing, burnout, and the formation of emissions. Tangential arrangement of secondary air jets may also be a good option for grate-fired boilers: The air jets form a strong rotating flow on the horizontal cross-sections, which results in a good burnout but also mitigates the deposit formation and corrosion on the furnace walls.

Due to the large fuel particle size, combustion is slow and the movement rate of the grate needs to be slow to guarantee complete burnout. Thus fuel residence time can go up to tens of minutes. Consequently also major load changes are relatively slow. Start-up time from cold to full load can go up to hours. To improve the load following capacity of existing grate boilers, model based control strategies have been developed. For a CHP plant, the model based control strategy configuration can be e.g. as follows: the primary air flow rate and the stoker speed are the manipulated variables and the moisture content in the fuel feed and the steam demand are the measured disturbances. In this approach, the mass of dry fuel in the thermal decomposition zone, the combustion power, and the steam pressure are the controlled variables. For steam boiler, the residual oxygen content of the flue gas, the feed temperature, as well as the mass of dry fuel in the thermal decomposition zone are typically chosen as variables to be controlled. Considering the water in the evaporation zone and the dry fuel in the thermal decomposition zone and the combustion power mass balances improve the load-following capacity of the boiler. In addition, the amount of the fuel on the grate needs to be held very close the set point to avoid primary air and secondary air saturations and their unfavorable effects /27, 28/

3.6.2 Recovery boiler

Recovery boilers are used in the pulping industry to recover white liquor from black liquor chemicals. In the recovery boiler furnace, black liquor is burned. The generated heat is usually used in the pulping process or for making electricity for use in the plant and partly also for sale.

Recovery boilers are normally run at full load and only in exceptional cases they are run at a lower load. Minimum allowable load is about 70% of the full load. The boiler is shut down if the pulping mill operation is interrupted. For short disturbances in the pulping process, the recovery boilers have a buffer in the form of a black liquor tank that contains black liquor for a few hours combustion. /29/

Recovery boilers produce electricity for the pulp mill. During startups electricity needs to be purchased from the electricity network. Newer plants produce a significant surplus of electricity that is sold out. On the contrary, old plants typically need to have an additional boiler (e.g. BFB or CFB) burning bark and wood waste to produce enough steam for the pulp mill. Due to the special role of the recovery boilers, their role in the energy networks differs significantly from that of other boilers since they produce a fixed power output. The amount of energy produced in recovery boilers is large and thus the amount of electricity delivered to the network needs to be taken into account in countries with significant pulp industry. However, since their operation is directly dependent on the operation and shutdowns of the pulp mills, recovery boilers can be considered rather as a disturbance than a resource for the flexible operation of the power grid. /29/

3.7 Steam turbine

Steam turbine is the prime mover of the steam power plant process. Superheated high pressure steam produced in the boiler flows through the turbine giving out thermal energy to be converted to mechanical energy rotating a turbine rotor and a generator. Steam turbine is one of the most expensive components of a steam power plant, and that is why it is very important to operate it in a proper way. Under normal operation conditions steam turbines are very reliable and long-lived components. However, in case of cyclic operation with fast load ramping and exceptional operation points a special attention should be paid to maintain the applicable operation conditions in order to avoid unexpected and very expensive damages.

3.7.1 General description

Steam turbines in power plants are typically axial turbines constructed from separate cylinders for high pressure (HP), intermediate pressure (IP), and low pressure (LP) sections. There are different cylinder arrangements depending on the turbine size, Fig. 11, /30/.

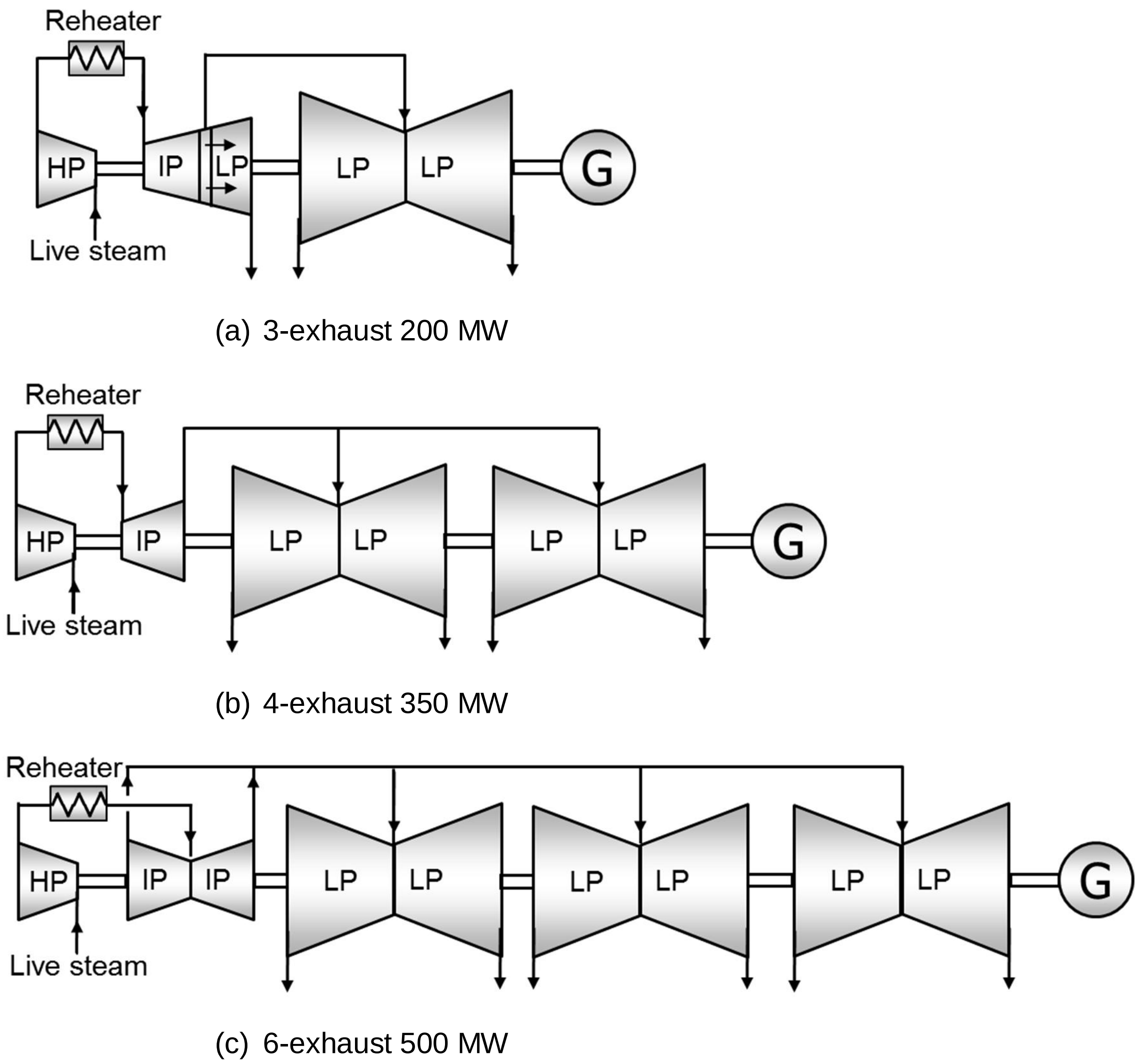


Figure 11. Steam turbine cylinder arrangements for different size tandem-compound machines.

Energy conversion inside the turbine takes place in two ways depending on the structure of the steam path. The path may consist of nozzles and blades (buckets) which can be fixed in a turbine shell or moving with the rotor. The nozzles with decreasing cross-sectional flow area increase the flow speed and drop the pressure of steam while blades change the flow direction and drop the flow speed while steam pressure remains constant. In a nozzle the increase of the flow speed exposes the nozzle to a reactive power while in a bucket the kinetic energy of steam flow is absorbed by the moving blade. A turbine composed of moving blades alternating with fixed nozzles is called an impulse turbine and a turbine composed of moving nozzles alternating with fixed nozzles is called a reaction turbine. The operation principles of different turbine types are illustrated in Fig. 12.

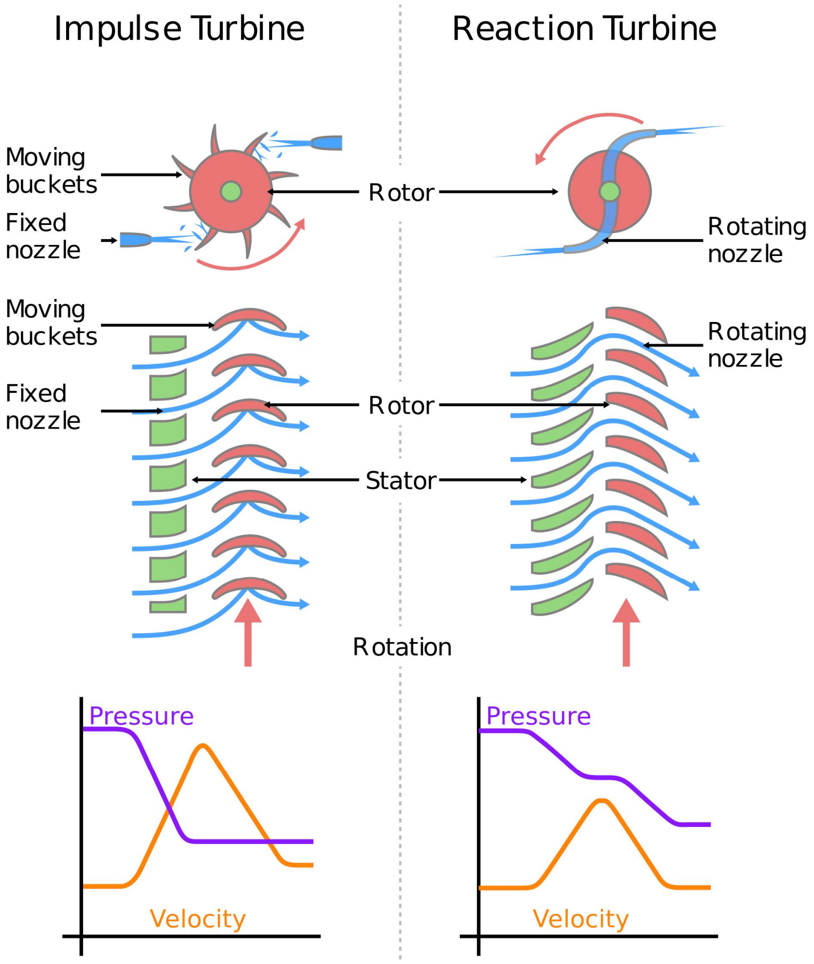


Figure 12. Operation principles of impulse and reaction turbines. /31/

In real machines, turbine blades are arranged in multiple stages in series forming a compound-system, where steam is expanded in several stages. This compounding greatly improves turbine efficiency especially at low flow speeds. A reaction stage is a row of fixed nozzles followed by a row of moving nozzles. Multiple reaction stages in series divide the pressure drop between the steam inlet and exhaust into numerous small steps, resulting a pressure-compounded turbine. Impulse stages may be either pressure-compounded, velocity-compounded, or pressure-velocity compounded (see Fig. 13). A velocity-compounded impulse stage (invented by Curtis and also called a "Curtis wheel") is a row of fixed nozzles followed by two or more rows of moving blades alternating with rows of fixed blades. This divides the velocity drop across the stage into several smaller drops. A series of velocity-compounded impulse stages is called a pressure-velocity compounded turbine.

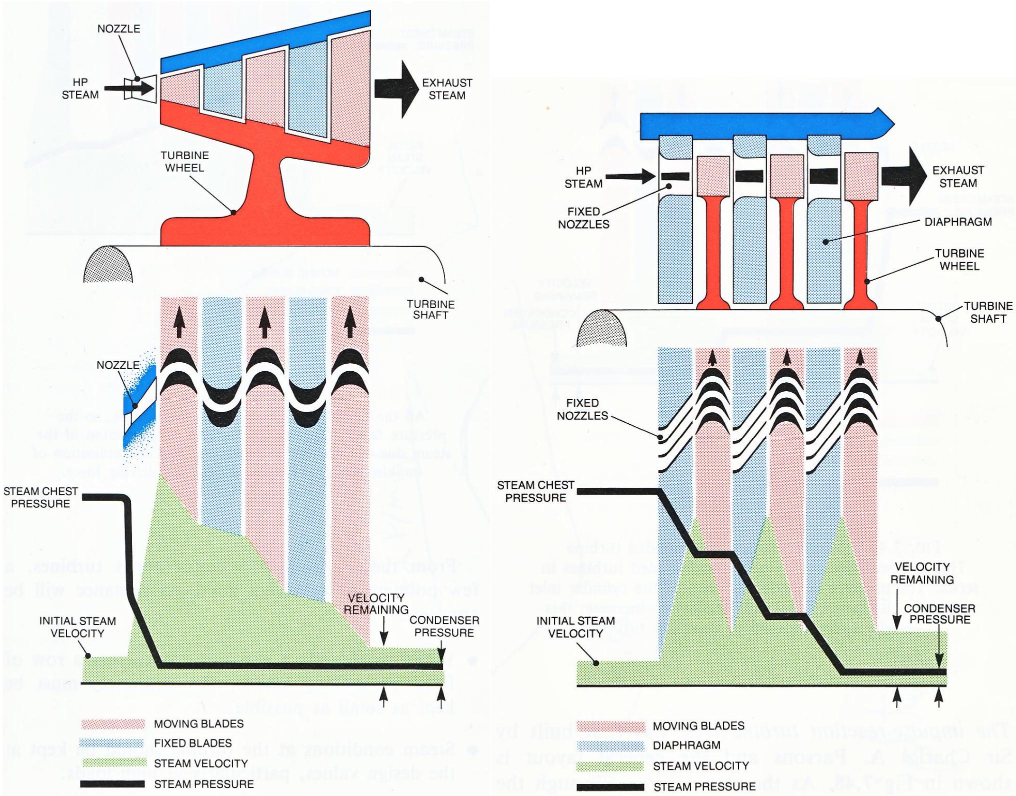
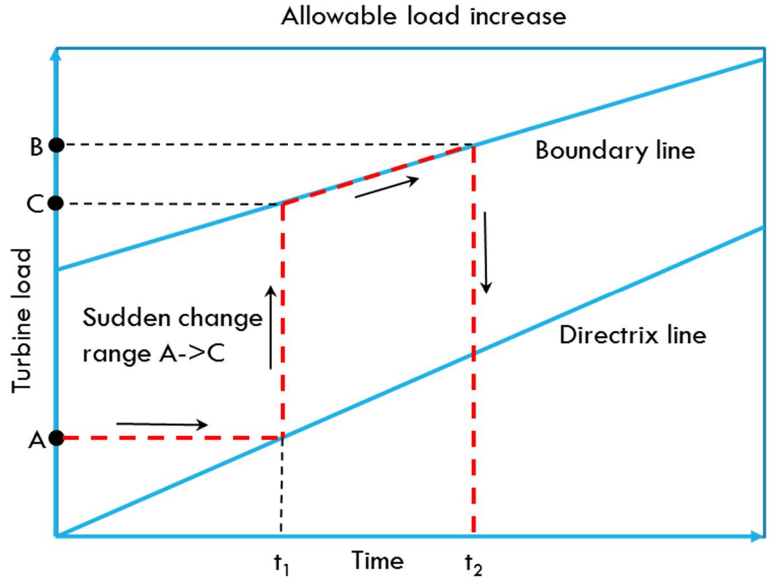


Figure 13. Velocity-compounded (left) and pressure-compounded (right) turbine structures. /30/

3.7.2 Ramp rate limitations

For a steam turbine, the startup phase, especially cold startup, is the most critical phase. The turbine must be warmed up and accelerated to the nominal speed strictly according to the instructions by the manufacturer. Due to very small clearances between rotor and stator blades, a thermal balance inside the turbine must be controlled very accurately in order to avoid seizing. Turbine rotor has characteristic resonance frequencies, which should be passed quickly during the acceleration phase. Turbine is typically accelerated to resonant free speeds step by step, let the temperatures to be stabilized after the step and then quickly skip over the resonance speed to the next safe speed. In this report the focus is in the operation and load change rate of already running and synchronized turbine, thus more detailed startup related issues will be skipped.

The maximum allowed change rates are defined for load change, inlet steam and material temperatures, and temperature differentials inside high pressure steam chest and cylinder walls, and different positions in turbine structures. Maximum load change rate in a thermal equilibrium for an individual turbine construction is defined by allowable load increase and decrease curves, Fig. 14. If the load change is inside a sudden change range, in Fig. 14 A-C for increasing load and B-D for decreasing load, the load change rate is not constrained by the turbine structure. If the load change is outside the sudden change range, the change rate is defined by the boundary line and the time needed for load change is defined by $t_2 - t_1$ for increasing load and $t_4 - t_3$ for decreasing load.



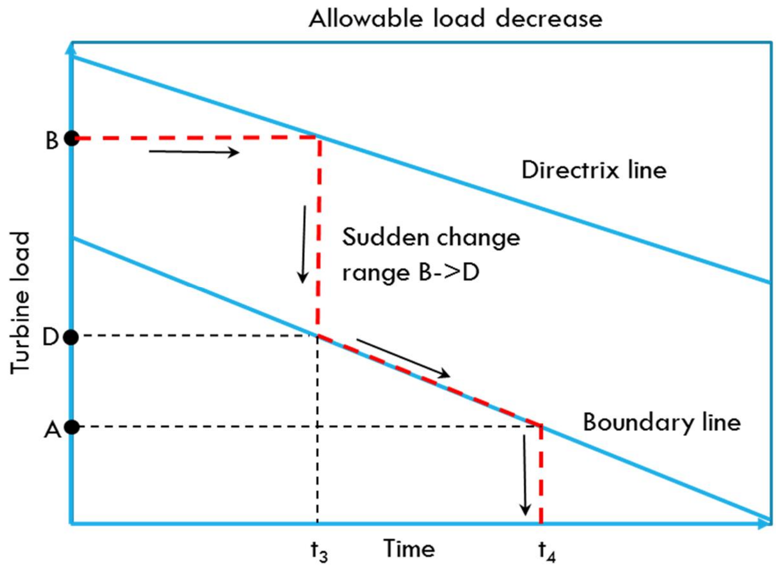


Figure 14. Structural load change rates for increasing and decreasing loads of a steam turbine operating at a thermal equilibrium state.

Temperatures inside turbine cylinders and casing materials depends on both live steam temperature and turbine load. Excessive temperature gradients in thick metal structures cause thermal stresses and thus should be avoided. Changing temperatures inside the turbine cylinder due to load change will cause uneven thermal expansion in turbine rotor and cylinder cases, which may cause seizing, i.e. rotor blades striking with fixed nozzles. For these reasons, the load change rate of the turbine is also constrained by temperature in different parts of the turbine. Figure 15 illustrates a typical temperature monitoring assembly in the high pressure steam chest of a turbine.

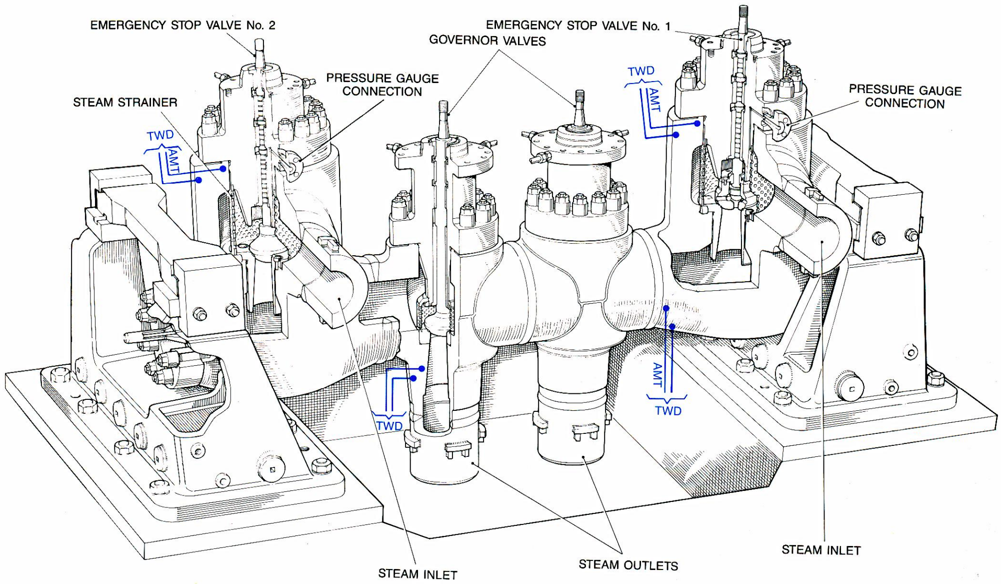


Figure 15. Steam chest temperature monitoring scheme for a 500 MW turbine. TWD is a through wall differential measurement with thermocouples (AMT). /30/

Maximum allowed variations and dynamics for live steam temperature are defined by the turbine manufacturer. The allowed temperature range can be described by a diagram presented in Fig. 16. When steam temperature is decreasing, the limit for the lowest allowed temperature is defined according to the temperature at the starting point subtracted by the maximum allowed negative temperature change (-)ΔT. During the temperature drop the limit for the lowest allowed steam temperature is changed according to a change rate coefficient (-)dT/dt. For the increasing steam temperature, the logic is analogous. The starting point for the maximum allowed steam temperature is defined by adding the maximum allowed positive temperature change (+)ΔT to the momentary steam temperature and during the temperature increase, the highest allowed steam temperature is changed according to a change rate coefficient (+) dT/dt. Typical values for maximum temperature change ΔT are 20 - 25°C and for maximum change rate dT/dt 1.5 – 2 °C/min.

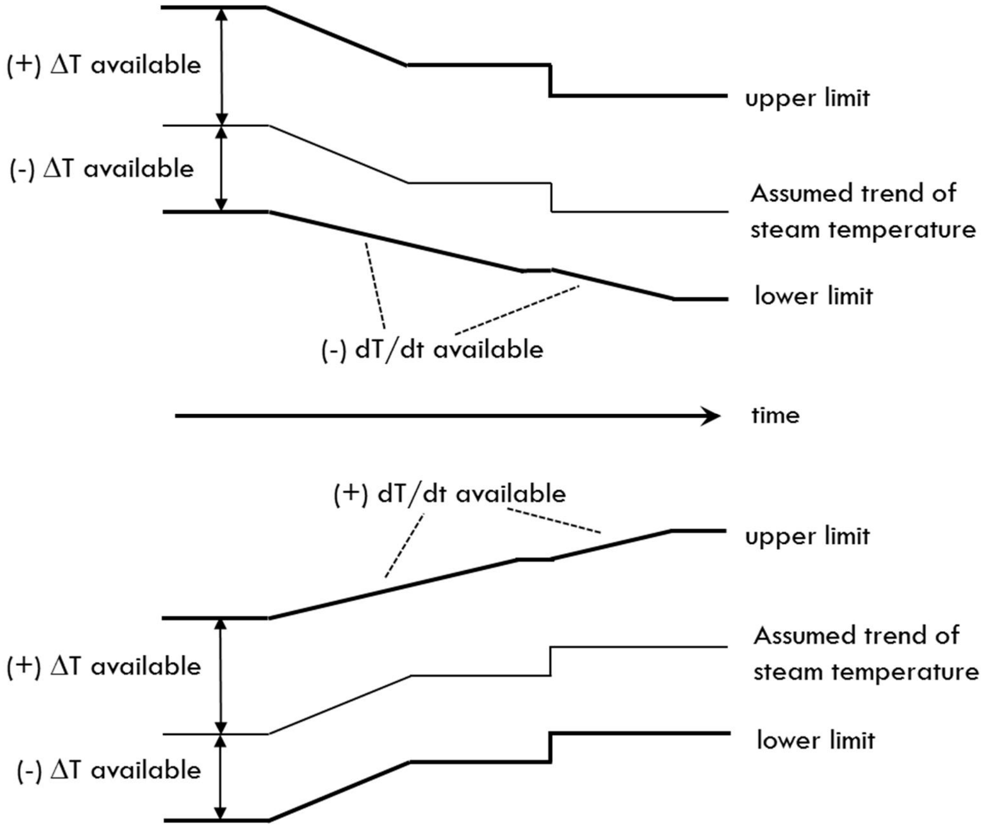


Figure 16. Limits for live steam temperature variations during a transient. Values for temperature change ΔT and change rate dT/dt are defined by the turbine manufacturer. Typical values for them are: ΔT ≈ 20 - 25°C, dT/dt ≈ 1,5 – 2 °C/min.

In modern turbines both structural load change rates and temperatures are measured or calculated with a turbine model in cases where direct measuring is impossible. This information is connected to a turbine controller, which automatically constrains the load ramping if maximum allowed limits are violated. Allowed load rate changes for individual turbines are defined by turbine manufacturers.

According to these turbine side limitations for the load change rate, it can be concluded, that in running power plants the steam turbine is typically not the bottleneck for load change rate performance but the limitations come from boiler side.

3.7.3 Minimum load limitations

There is no actual minimum load for steam turbines. If needed, they can be connected in the power system without generating any electric power. In this state the system is set to be a spinning reserve. However, there is some minimum steam load needed to idle the turbine to keep the turbine in operation temperature. However, when loading a turbine, a typical minimum load is something 10 – 15% from nominal load.

Temperatures of outlet steams from HP, IP and LP stages tend to rise along with the decreased load. In practice this may reduce the minimum allowable turbine load. Another minimum load limiting factor is decreased live steam temperature at low loads of the boiler. When a steam turbine is running at its nominal temperature the safety function related with minimum live steam temperature is activated to protect the machine from excessive temperature changes (see Fig. 16)

3.8 Process control

Process control has two separate tasks; to move a process from one operation point to another and to compensate effects of disturbances at a constant operation point. The first task is so called tracking control problem and the second task is so called stabilizing control or output regulation problem. In power plant control both of these tasks are topical, but the importance of the load tracking control will increase due to the increased penetration of renewable energy production in our power system.

Power balance of the power system is maintained by Transmission System Operators (TSO). They make contracts with power producer to maintain the required control power capacity to stabilize the system. The control concept consists of three steps: Primary Control Reserve (PCR), Secondary Control Reserve (SCR), and Tertiary Control Reserve (TCR).

A complete deployment of activated primary control reserve has to be realized within 30 seconds. Primary control is connected to power system frequency and control is activated automatically when network frequency deviates from its nominal range (49.9 – 50.1 Hz). In thermal power plants, especially the steam storage capacity of the boilers realizes this fast change of power and thus an increase of power is limited. Due to the limited steam storage capacity, this increase is only available only a restricted time period and this PCR should be substituted by other reserves as fast as possible.

Secondary control reserve (SCR) is – as primary control reserve (PCR) – an automatically activated control reserve, but with slower response dynamics. In the Continental system (UCTE) the activated control reserve has to be realized within 15 minutes. In the Nordic countries the SCR must be activated within three minutes and it must least up till 15 minutes, if the frequency stays out of normal range so long. SCR is activated only in those control areas where system imbalances occur.

SCR is not the best way to manage longer lasting system balance failures, e.g. due to forecast errors or power plant failures. This reserve demand is provided by tertiary control reserve (TCR). Requirements to this tertiary control reserve are lower compared with SCR. Thus, technical units having less flexibility than those required for SCR may be used for TCR.

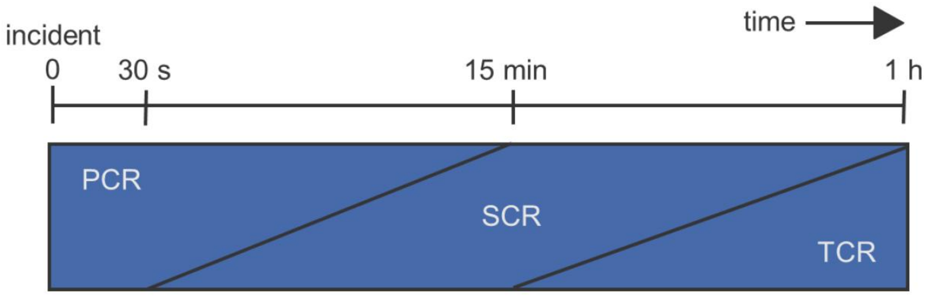


Figure 17. Three step control concept in the European interconnected system. /35/

3.8.1 Control approaches

Unit control

The conventional approach to load tracking control in power plants is the unit control of boiler – turbine system. The unit control consists of so-called master controls, namely power output control and live steam pressure control. The primary task of the master controls is to maintain the required power output of the power plant. The power related output can be electric power (condensing power plants), steam pressure in back pressure turbine outlet (industrial CHP power plants), or outlet temperature of district heating water (district heating CHP power plant).

In the conventional unit control scheme the operation point is controlled by two manipulated variables; fuel power (fuel feed into the boiler) and live steam admission to the turbine (turbine main control valve). Power output can be controlled either by turbine inlet control valve or by fuel power control. The manipulated variable left free from power output control is used for live steam pressure control. Thus, if the power output is controlled by turbine main control valve, steam pressure will be controlled by fuel feed (so called “boiler follows” scheme). In the other option, the power output is controlled by fuel feed and live steam pressure by turbine control valve (so called “turbine follows” or turbine front pressure control). These different options are illustrated in Figs. 18 and 19.

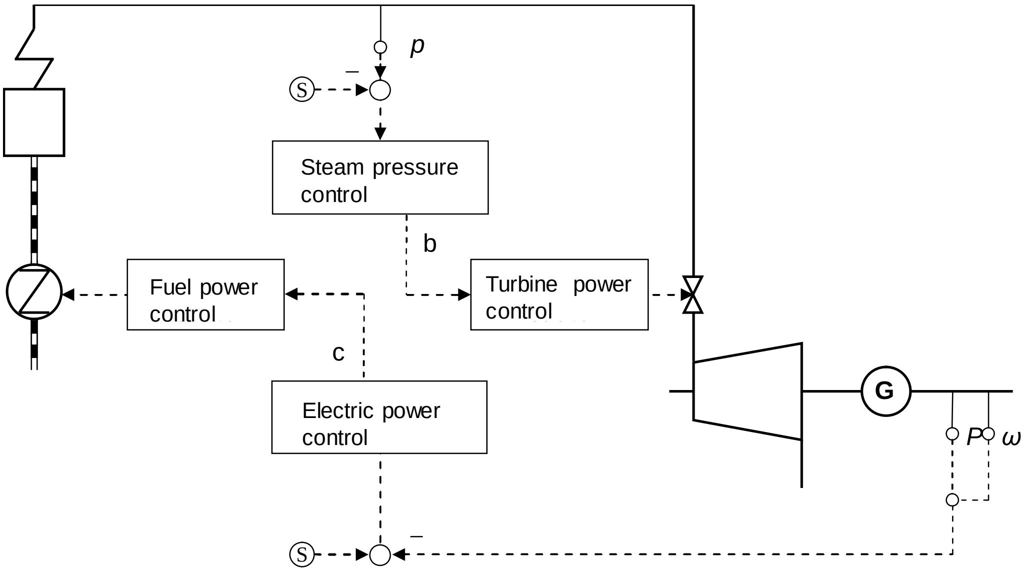


Figure 18. Turbine follows control structure for unit control. Modified from /32/.

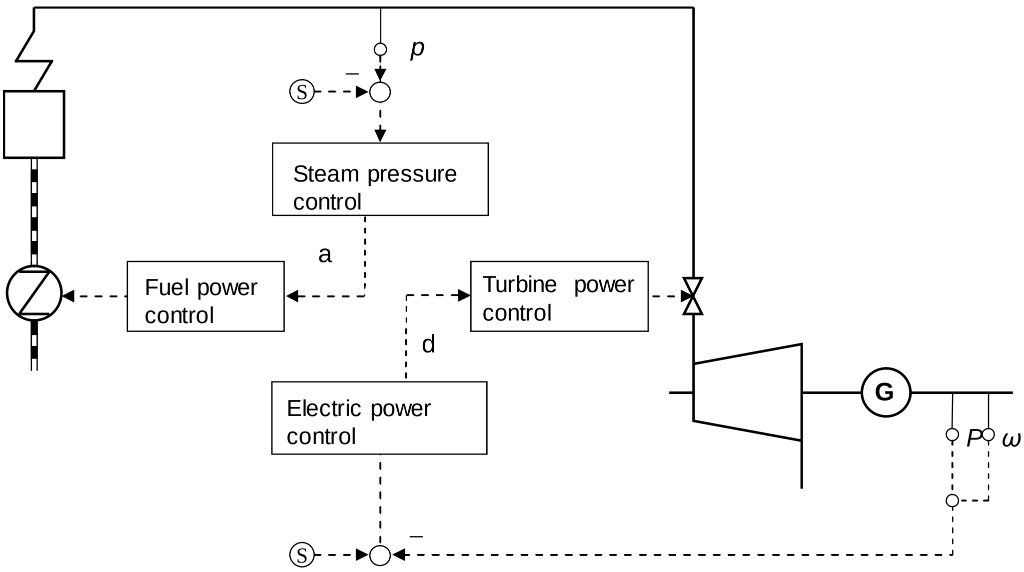


Figure 19. Boiler follows control structure for unit control. Modified from /32/.

In “turbine follows” control, load change is produced by first changing the fuel feed into the boiler. Changed fuel feed responses to steam pressure which will be regulated by the turbine valve, increasing or decreasing steam flow to turbine depending on if the steam pressure is risen or dropped. In this structure, the load change rate depends only on the boiler dynamics from fuel feed to steam pressure. Power output will not be changed until the boiler has changed its steam generation. Live steam pressure will remain very stable during the load change.

In “boiler follows” control, load change is produced by first changing the position of the turbine control valve. Changed steam flow will disturb the live steam pressure, which will be regulated by manipulated fuel feed. In this structure, power output will react immediately after the manipulation of turbine control valve position. Increased steam flow from the boiler will drop drum pressure and release steam capacity stored in the vaporizer and drum. Thus the boiler can produce extra steam even if the combustion power in the furnace is not changed yet, but the boiler is utilizing thermal energy stored in the boiler structures. The drawback of this structure is a worsen live steam pressure stability causing thermal stresses and possible control problems in other parts of the boiler process.

In the conventional unit control scheme, power and pressure controllers are working independently without any information about each other and an overall state of the whole system. Control is based on feedback regulation of disturbances caused by load change operations. Steam pressure is disturbed by changed steam flow (boiler follows control) or changed combustion power (turbine follows control). In other words, boiler turbine system is disturbed from one corner and feedback control eliminates the disturbance by moving the process to a new operation point. This type of unit control is not very effective, because it is slow and typically it causes fluctuations to system outputs. It is suitable only for small load changes, not e.g. for ramping over a wide operation range. One reason for this is that especially boiler process has a nonlinear characteristics resulting that conventional controllers can be tuned to work satisfactorily only inside a limited operation range. Because of this, it is typical that bigger load changes must be operated either manually or slowly in small steps.

Coordinated control

Coordinated control is based on the co-operation between different sub-processes so that the control system is aware of the resources available for system control. Calculation of control actions takes into consideration margins and constraints related with the process operation. Typical constraints considered are capacities of feed water pumps, combustion equipment, fans, operation of preheaters, position of turbine control valve, and thermal stresses in the boiler and turbine.

The basic idea of the coordinated control is instead of feedback control to utilize parallel feed forward control to put the process in to a new position. During a load change, the unit coordinator calculates new set points for all subsystems taking into account states and constraints of every process component. Feed forward control puts the process as near as possible to a new operation point, and the final adjustments are carried out by feedback controls.

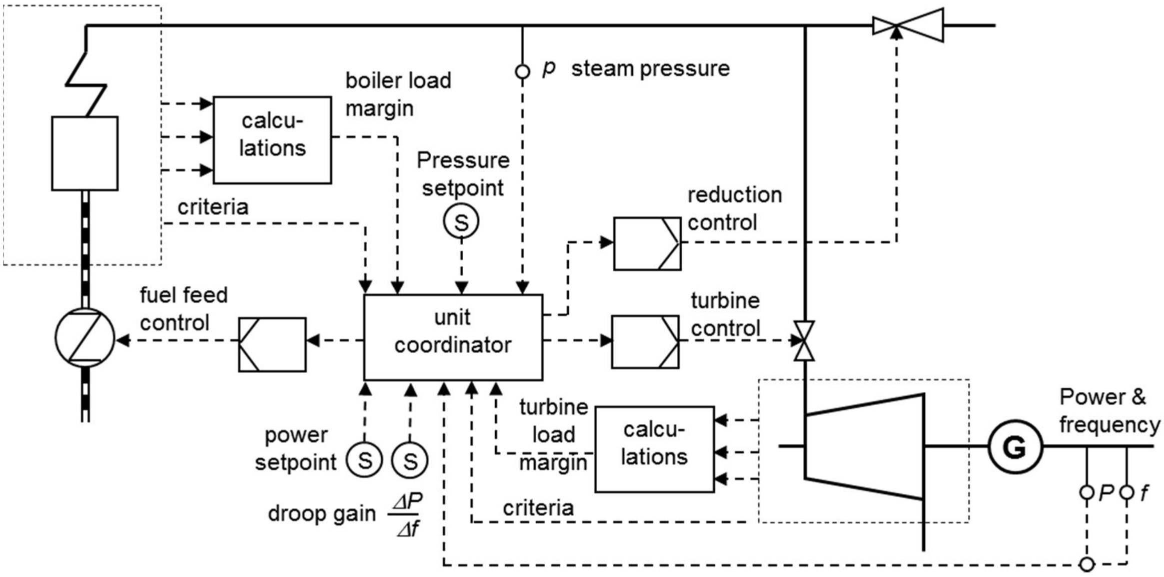


Figure 20. Principled structure of the coordinating unit control. Modified from /32/

Model predictive control structure (MPC) is a common way to apply coordinating control. MPC is an optimal multivariate control system, which can include system constraints when calculating new control actions for the process. /33/

Stabilizing controls

The role of stabilizing controls is to keep the process in the state defined by the unit control. The most important stabilizing controls in a fixed pressure drum boiler are illustrated in Fig. 21.

- Feed water control including drum level in drum boilers (C1)
- Steam temperature (C2)
- Steam pressure (C3)
- Power output (C4)
- Furnace pressure (C5).

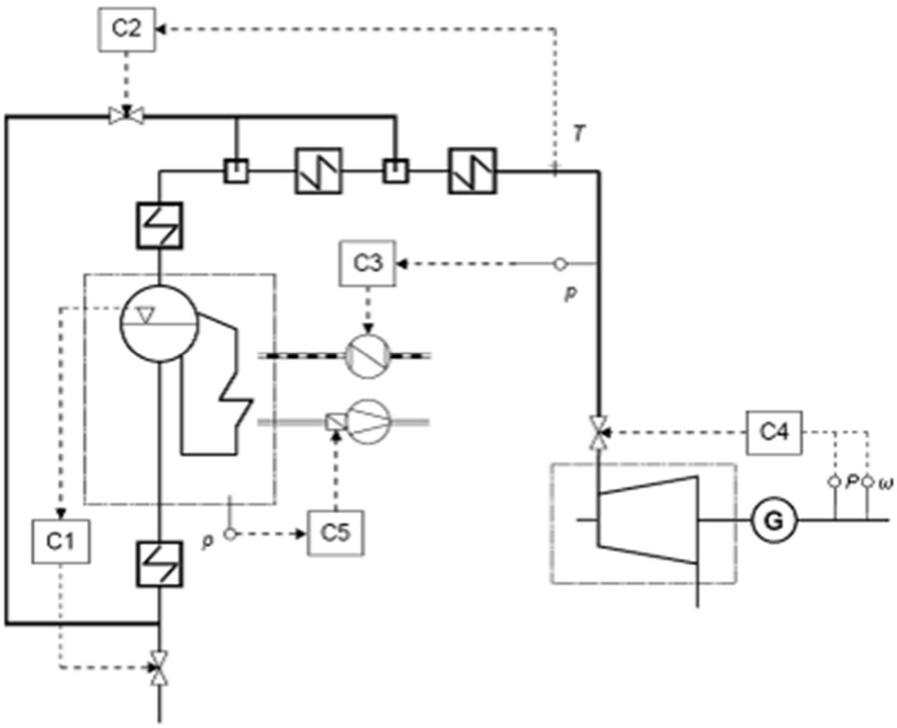


Figure 21. Main stabilizing control loops of a fixed pressure drum boiler. Modified from /32/.

Feed water control

Water balance in the boiler is controlled by feed water flow. In general, when feed water flow into the boiler equals with steam flow out from the boiler, the amount of stored water/steam in the boiler remains constant. In modern power plants during normal operation feed water flow is controlled by rotation speed controlled feed water pumps. Control structure depends on the boiler type; drum boilers and Benson and Sulzer type once through boilers have different type of control structures.

In drum boilers, feed water control is based on drum level. In a long run, stabilization of water level in a drum results a balance between feed water inflow and steam outflow from the boiler. However, drum level behavior during load changes is complicated as explained in Section 3.6.2, which makes the level control complicated as well. Because of this swell and shrink effect, known as non-minimum phase transient response, water balance in the drum boiler cannot be controlled just as a water level control in the drum. The most commonly used control structure is a so-called three element control. The three elements applied in the control are drum level, steam flow, and feed water flow. The basic idea is that the control signal manipulating the actuator (pump or valve) is made up mainly from a difference between feed water and steam flows. However, flow measurements contain errors and there are some side flows, which are not measured, thus the final balancing of incoming water and outgoing steam flows is controlled according to drum level. The three element control is tuned so that the controller output reacts rapidly to the flow differences but slowly to changes in drum level. In this way, the control signal is not disturbed by this shrink and swell behavior in the drum. For trouble free operation of the boiler, this control structure must be tuned carefully. This is very important also because the drum level is connected to boiler safety automation. If there is not enough water in the drum or boiler water may flow to superheaters, the boiler will be shut down. The structure of three element drum level control is depicted in Fig. 22.

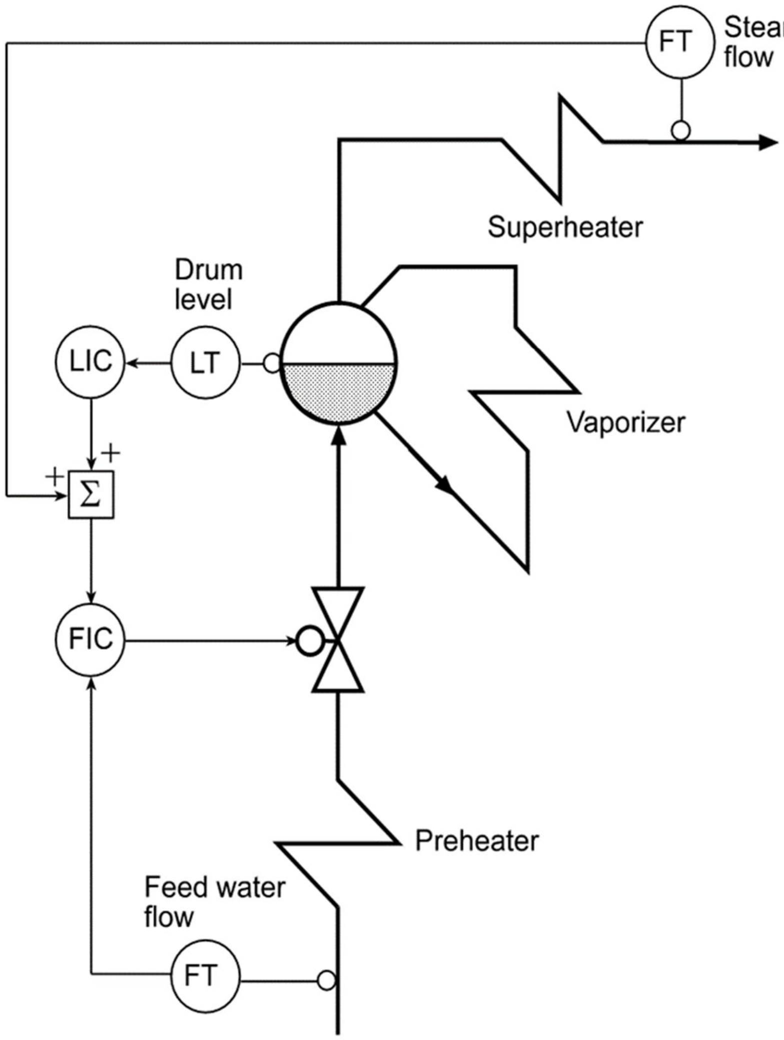


Figure 22. Three element control of drum boiler feed water control.

In once through boilers this kind of control structure cannot be utilized, because in those boilers there is no drum. In once through boilers feed water amount should be related with the released fuel power in the boiler so that feed water amount matches with the amount of thermal energy available to evaporize and superheat the water. This cannot be measured directly, but the feed water control in Benson boilers is based on steam temperature. The superheating section consists of superheaters and attemperators located between superheaters. The attemperators are used for steam temperature control. They drop temperature by spraying feed water in the steam. Benson boiler operation is designed to work so that there is a certain temperature drop across the attemperator. This means that thermal powers for evaporation and superheating are distributed as designed. Thermal power (fuel feed) of the boiler is controlled according to the power output demand and feed water flow is controlled to produce steam at the desired temperature. The control structure is depicted in Fig. 23.

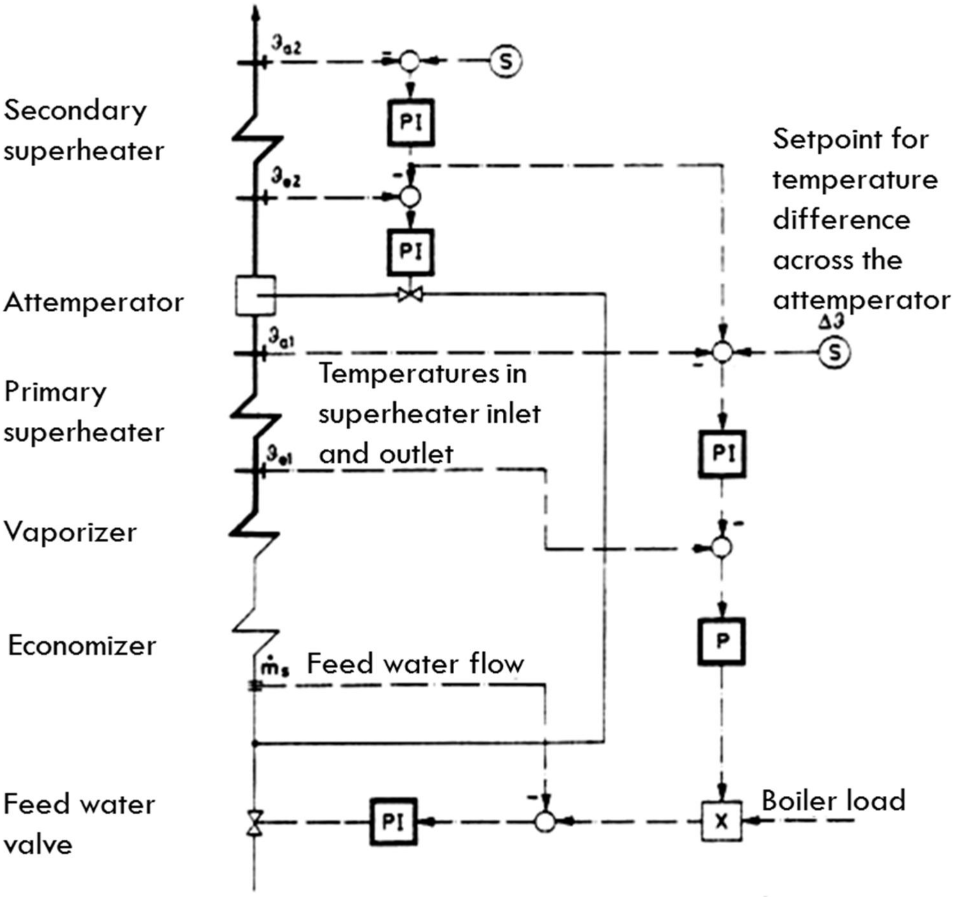


Figure 23. Structure of the feed water control in Benson boiler. Modified from /34/

<u>Steam pressure control</u>

Steam boilers can be operated either with fixed or sliding live steam pressure. In general, drum boilers are operated with fixed pressure and once through boilers with sliding pressure. However, this is not an absolute truth, because in some cases also drum boilers may be operated with sliding pressure, but this is not very common due to control problems and thermal stresses caused by pressure changes in the drum. Fixed pressure drum boilers have better load change rate compared with sliding pressure once through boilers due to utilization of boiler's energy storages. With sliding pressure operation, the efficiency of electricity production is better due to smaller internal power consumption (feed water pumping power) and there are no throttling related losses in the turbine (reduced enthalpy drop due to increased steam entropy).

In fixed pressure operation where steam pressure is maintained by fuel feed (boiler follows structure), the control structure consists of two layers (a cascade control); the primary (or master) controller defines the fuel power required for maintaining the desired steam pressure, and the secondary level controllers (slave controllers) are connected with the fuel feeding system to deliver the required amount of fuel into the furnace. There is usually feed forward connection from steam flow to fuel feed to improve the dynamic performance of the control structure. The master level pressure control is depicted in Fig. 24.

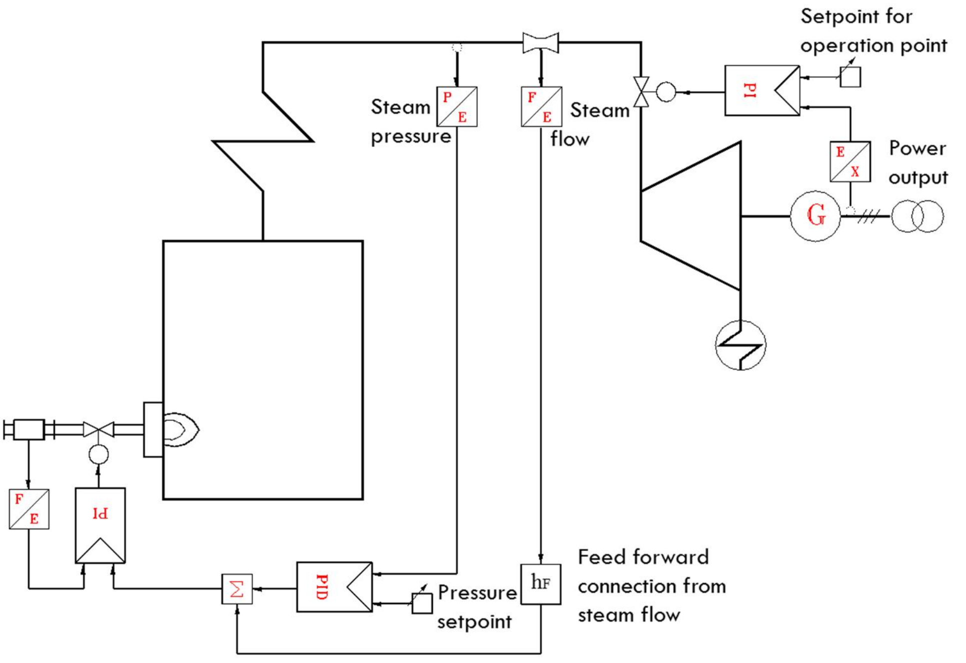


Figure 24. Live steam pressure control, where master controller computes required fuel power for combustion control. Control loop includes feed forward connection from steam flow to fuel feed.

In turbine follows structure live steam pressure is maintained by turbine control valve. This structure is presented in Fig. 25.

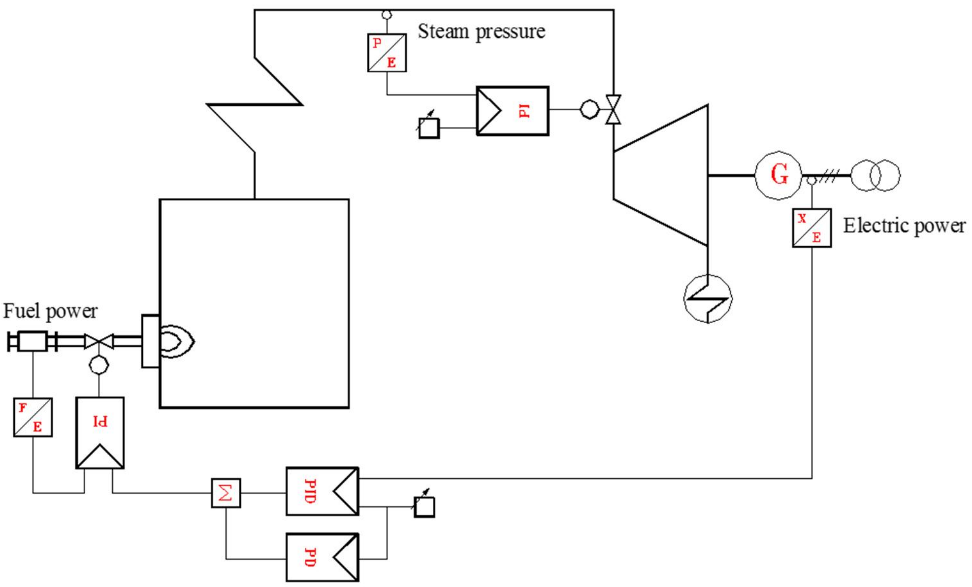


Figure 25. Live steam pressure control by turbine control valve (turbine follows structure, known also as turbine front pressure control).

In sliding pressure operation there is actually no pressure control, but the power plant is operated with turbine control valve 100% open and steam pressure is floating as a function of turbine load. However, there is also so called supervised sliding pressure mode, where steam pressure is floating as a function of turbine load, but turbine control valve is not fully open, but it is choked app. 10%. With this supervised mode the load tracking capacity of the boiler-turbine unit can be improved compared with pure sliding pressure operation, because by opening the turbine control valve fully open in case of rapid load increase, it is possible to utilize the energy storage of the boiler and increase temporary load change rate. The principle of supervised sliding pressure operation is depicted in Fig. 26.

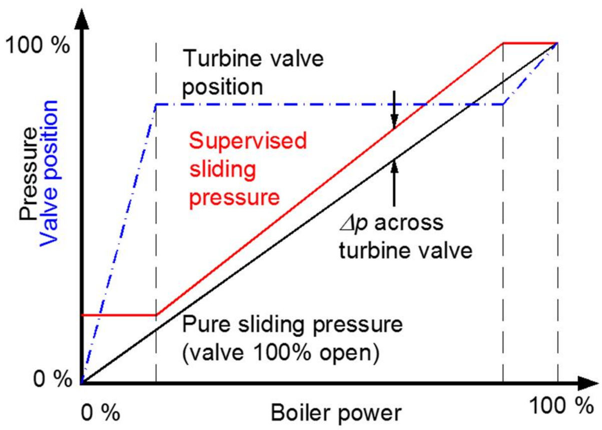


Figure 26. Operation principle of the supervised sliding pressure control, where during a normal operation the turbine control valve is app. 90% open providing some boosting capacity during rapid load increase by fully opening the valve. Modified from /32/

Combustion control

Combustion control in thermal power plants consists of fuel control and combustion air control. A corner stone of efficient combustion is a functional control of fuel/air ratio in the furnace. Lack of combustion air will result to imperfect combustion, fuel losses, and corrosion in the furnace, while excess in combustion air will result to increased flue gas losses and increased NO_X emissions. Fuel control is connected either to the live steam pressure control (boiler follows unit control) or to power output control of the power plant (turbine follows unit control). Different combustion methods have different type of control solutions depending on the structure of the fuel handling, e.g. crushing, milling, drying, etc. More information about these different applications can be found e.g. from /32/..

Combustion air control can be either series-connected or parallel-connected with the fuel feed. In the series-connected control structure a change in load demand changes at first either fuel feed (decreasing load) or combustion air feed (increasing load) and the second variable is following the first one according to the measured change of the first variable. A strategy of this control structure is to guarantee a sufficient amount of combustion air during load transients. In case of increasing load, amount of combustion air is increased at first, and fuel feed is increased just after the change of combustion air is detected by the measurement. In case of decreasing load fuel feed is reduced at first and combustion air follows according to the measured change of fuel flow. This series-connected structure is depicted in Fig. 27.

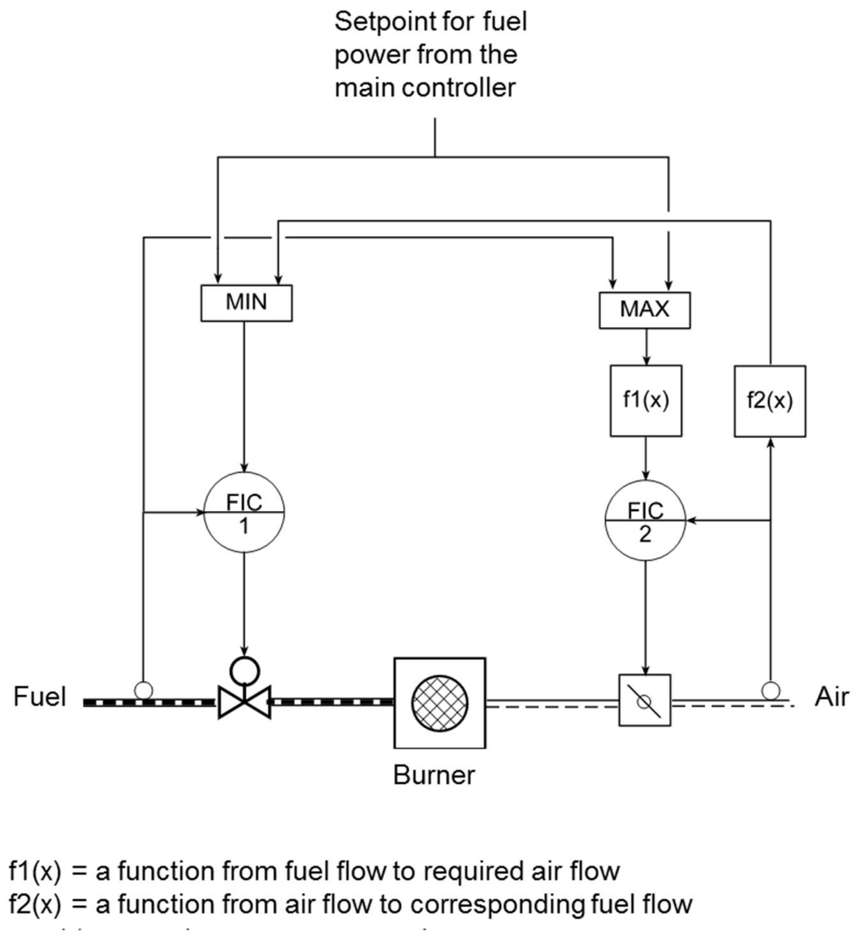


Figure 27. Series-connected control of fuel and combustion air feeds.

In the parallel-connected system both fuel feed and combustion air feed are manipulated at same time according to master controller power demand. This is a typical structure for gas and liquid fuel fired boilers, because their fuel feeding dynamics is fast and deterministic compared e.g. with solid fuel fired systems.

The amount and distribution of combustion air is defined by a boiler manufacturer. Air distribution diagram defines how much combustion air is needed as a function of boiler load and how this air shall be distributed to primary, secondary, tertiary, etc. feeds depending on the air distribution structure of the boiler. Fig. 28 depicts typical air distribution curves for combustion air feed for an air distribution system consisting of primary and secondary airs.

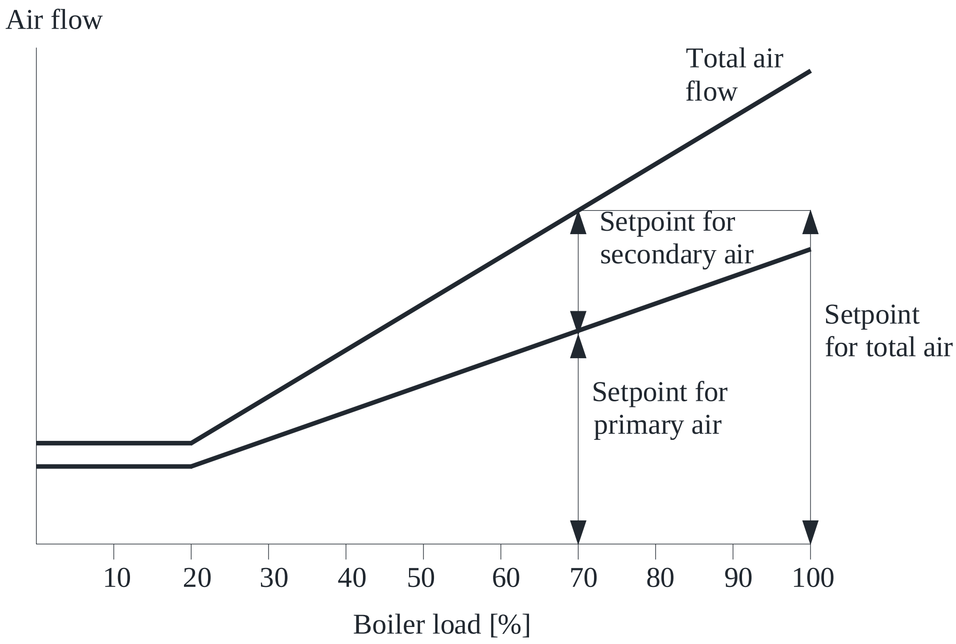


Figure 28. A typical air distribution curve to define set points for primary and secondary air flows and total air flow (primary + secondary) as a function of boiler load.

Initial fuel/air ratio of combustion is defined by total air curve. However, the air curve design is based on properties of a so called reference fuel. Especially in solid fuel fired boilers fuel properties may vary a lot, and air distribution designed according to reference fuel may lead to very inefficient combustion. The fine tuning of fuel/air ratio is carried out by controlling O_2 concentration in flue gases after the combustion zone. O_2 concentration is measured from flue gases and O_2 controller adjusts typically secondary air flow to get to the desired after combustion concentration. Figure 29 depicts how the optimal fuel/air ration defines the optimal operation point and Fig. 30 illustrates the control loop. Optimal set point for O_2 concentration depends on the boiler load, condition of fuel preparation system (e.g. coal mills in pulverized coal combustion), fuel properties and numerous other process variables. Thus the locations of loss curves are not constant but change as a function of those aforementioned variables. One applied method to define the optimal operation point is to include CO – measurement to the control scheme to indicate the limit of imperfect combustion, i.e. carbon is oxidized to CO instead of CO_2 due to a lack of oxygen.

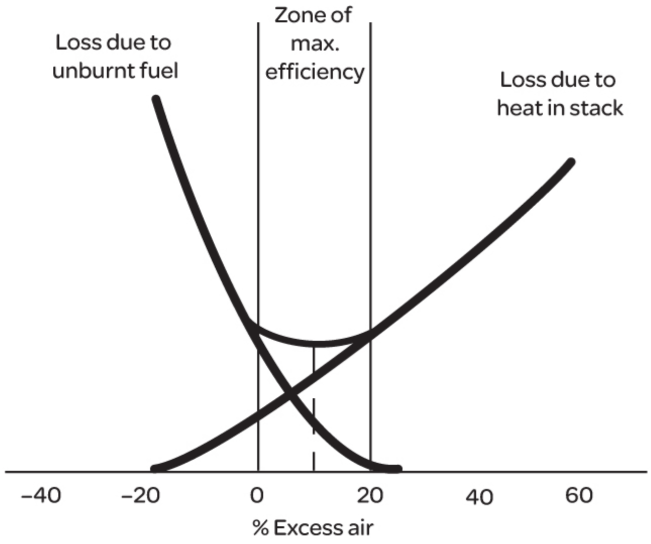


Figure 29. Optimal set point for fuel/air ratio.

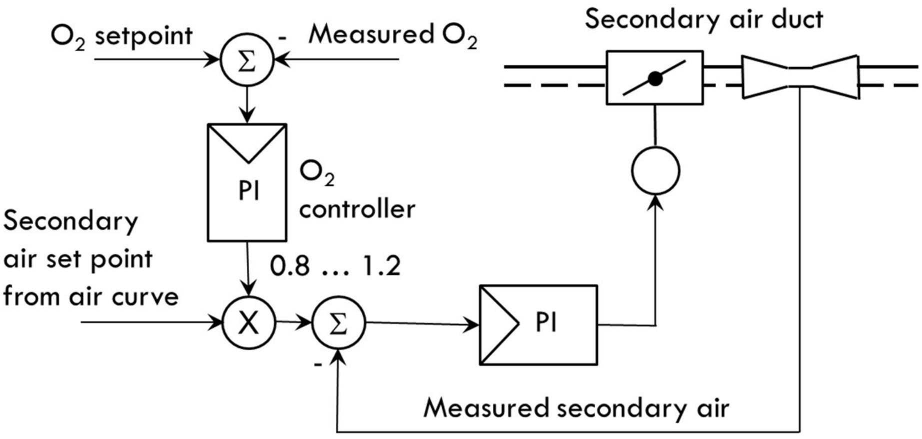


Figure 30. O_2 control loop for fuel/air ratio control. Modified from /32/.

<u>Steam temperature control</u>

Live steam temperature has an important role in the operation of the boiler – turbine system. The higher the temperature in a steam turbine inlet, the higher the conversion efficiency. Also steam moisture at the low pressure stage of the turbine is reduced with increased steam temperature, which in turn reduces water drops caused erosion in rotor blades in low pressure section. However, the maximum allowed steam temperature is constrained by material properties of high pressure steam line and high pressure steam chest of the turbine. The main task of steam temperature control is to stabilize the temperature as near the maximum allowed value as possible. How near to that limit the temperature set point can be adjusted depends on the performance of the control loop. The difficult thing in steam temperature control is the slow dynamics of the superheating process. Superheaters are massive steel structures resulting to time constants app. 100 … 150 seconds from input temperature to output temperature. Because temperature overshoots are not desired, the control loops must be tuned to operate slow.

The basic structure in steam temperature control is a cascade control consisting of primary controller connected with superheater output temperature and secondary controller, controlling superheater input temperature by an attemperator spraying cooling water in steam. The secondary controller gets its set point from the primary controller, i.e. the primary controller defines the input temperature for the superheater which will result to the desired output temperature. The basic structure of temperature cascade is presented in Fig. 31.

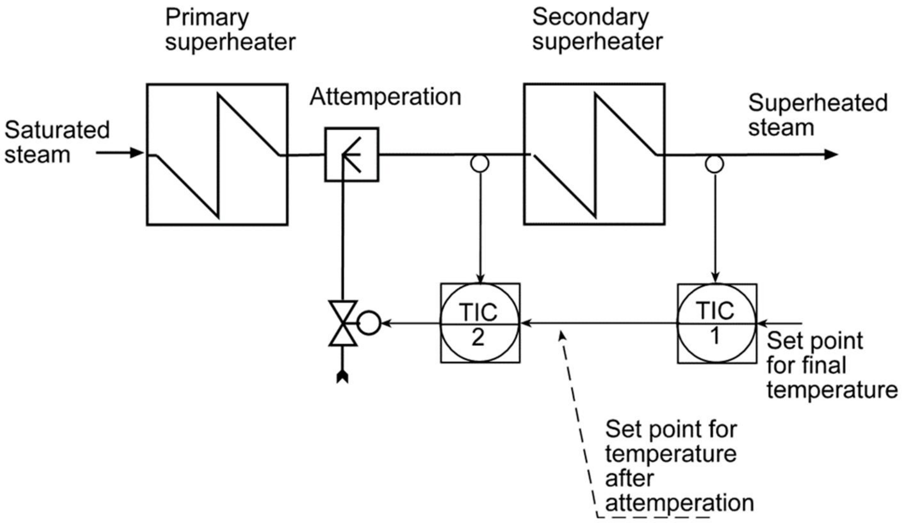


Figure 31. The principled structure of the steam temperature control of one superheating stage.

Live steam superheating system of medium and large size power plants typically consists of three successive superheating stages. In large boilers there are also parallel superheating lines. Fig. 32 shows one way to configure the temperature control of a superheater chain. Temperature control of secondary and tertiary superheaters' outlet temperature is based on cascade control shown in Fig. 31. The interconnection of the stages is done by manipulating the set points of secondary and tertiary superheaters' inlet temperatures and secondary superheater's outlet temperature as a function of boiler load. By this way the operation of attemperators is shared in a proper way to both spraying units. There is also a derivative type feed forward connection from fuel feed to inlet temperature control of tertiary superheater to speed up the control system response to a change in released fuel power in the furnace.

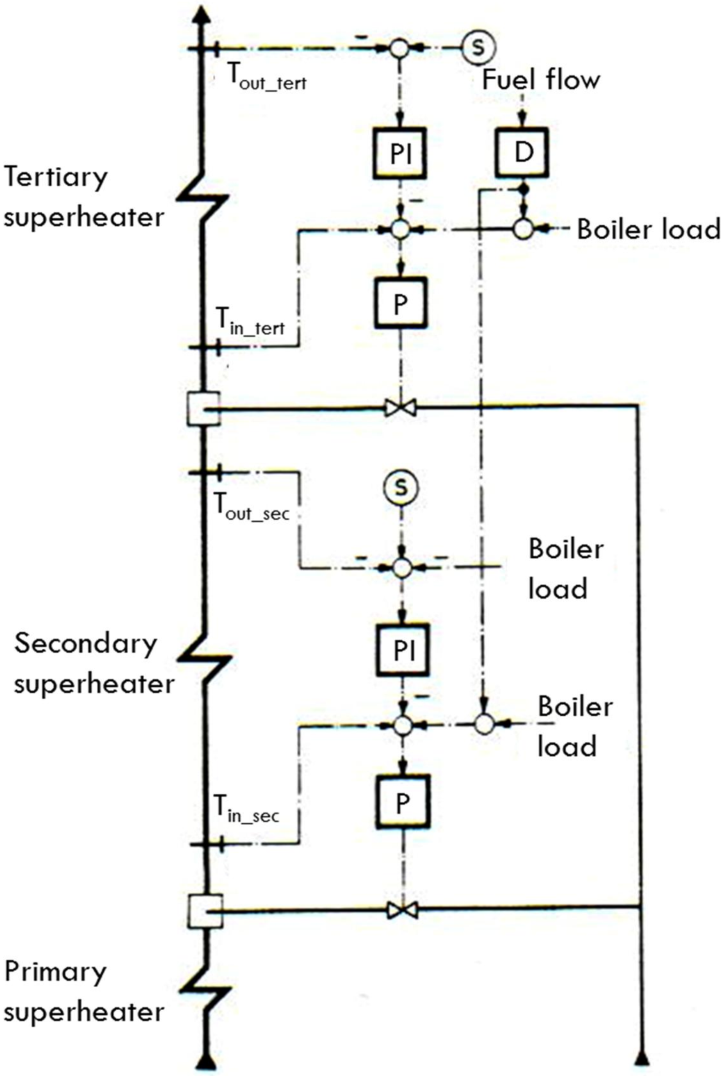


Figure 32. Temperature control of a three stage superheater chain. Modified from /34/.

Furnace pressure control

Flue gases from combustion process are removed from the furnace by flue gas fans. The fans are controlled by the measured pressure from the furnace. In big power plant boilers, the set point of the furnace pressure is app. 10 – 20 kPa under the atmospheric pressure. By this way all the possible gas side leakages will flow from a boiler house to a furnace. Thus boiler house will be free of flue gases. Furnace pressure stability is important, because fluctuating pressure will stress furnace structures and it affects combustion air control as well. That is because the furnace pressure is the outlet pressure of combustion air feed, and fluctuating pressure will result to fluctuating combustion air flow, which will disturb the combustion process. Furnace pressure is also connected to boiler safety system, so that if the pressure cannot be controlled to a defined range, the boiler will be shut down. For a good dynamic performance of the furnace pressure control, there is usually a feed forward connection from combustion air control to flue gas fan control. The structure of the furnace pressure control is depicted in Fig. 33.

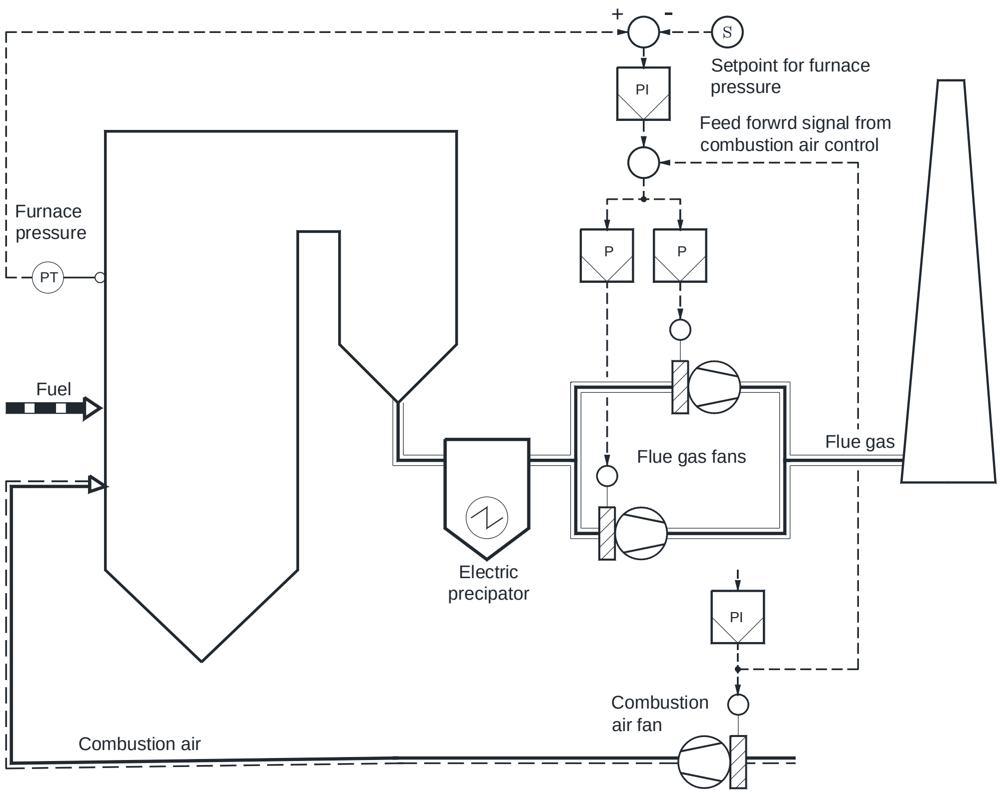


Figure 33. Furnace pressure control with feed forward connection from combustion air control. Modified from /32/.

3.8.2 Ways to cope with ramp rate limitations

According to dynamic analysis of temperature changes in boiler and turbine structures and discussions with experts on strength calculations, the conclusion is that for rapid but pretty small load changes (5 – 10% from the maximum rated capacity) the limiting factor for the dynamic performance is process control. Fast ramping from minimum load to maximum load has not been analyzed yet, so there are no results about that. For fast load changes carried out by utilizing energy storages of the boiler, temperature changes are so small that they have actually no effect on the expected life time of the boiler or the turbine.

Rapid load changes boosted by pressure change in the evaporator are challenging for drum level control and steam temperature control. Along with a rapid change of combustion power also combustion air and furnace pressure controls are challenged besides the feed water and temperature controls. For drum level control the shrink and swell effect caused by drum pressure change must be tackled in some way. In a ”normal” operation the three element controller can be tuned so that the flow controller maintains the mass balance and the level controller slowly compensates the effects of measurement errors and unmeasured side flows from the balance calculation without being disturbed too much by the shrink and swell effect. However, it may be possible that the extended utilization of the storage capacity of the boiler will increase the magnitude of the shrink and swell effect so much that the standard control structure is not in working order any more. In order to avoid the disturbance caused by this extended non-minimum phase transient response, the level controller may be needed to tune so slow that it is not any more able to compensate the errors. In this case a novel control design is required to be able to cope with this problem.

For live steam temperature control the pressure boosted load change is problematic. In the first phase the additional steam flow is produced from saturated water in the evaporator. However, in this phase there is no additional fuel power yet for superheating the additional saturated steam. Also the temperature of the saturated steam is dropped due to pressure drop in the drum. Thus live steam temperature will drop. Depending on the operation point of the boiler the superheated steam temperature control is activated or not, i.e. the superheated steam is cooled by attemperator sprays or not. If the boiler is operated at high load with steam cooling, steam temperature drop can be reduced by closing spray water valves. In order to improve the dynamic performance of the disturbance attenuation, the temperature control loop needs at minimum a new feed forward connection indicating temperature disturbance caused by the utilization of the boiler energy storage. In the existing control structure there is usually a feed forward compensation from fuel feed, but in this first phase of this type of load change there is no change in fuel feed but steam flow is increased. If the boiler is operated at part load and spray water valves are closed, there is no way to compensate the temperature disturbance. In this case the pressure drop in the boiler just have to be constrained according to the maximum allowed temperature change. Another thing which must be taken care is that when the thermal power starts to increase in the furnace, the temperature control system must be ready for increase steam cooling rapidly. In this kind of rapid load ramping the boiler is typically overfired (fuel power exceeds the steam power) which is also challenging for steam temperature control.

In rapid changes of fuel power also combustion air and furnace pressure controls are challenged. The conventional solution of series connected or parallel connected fuel-air feed is probably not capable to maintain thermally and environmentally efficient combustion environment in the furnace. Depending on the combustion system, management of flue gas flows and mixing of air and fuel in the furnace during transient operation is very difficult. Especially in circulating fluidized bed combustion this is very important and difficult. To maintain an optimal combustion environment in the furnace air distribution and flow controls must be probably redesigned.

Rapid changes in fuel and combustion air feed may cause severe disturbs to furnace pressure. This is problematic, because pressure changes in the furnace disturbs also combustion air feed and furnace pressure is connected to boiler safety system. If the furnace pressure cannot be maintained in the acceptable range, the boiler will be shut down.

In wide range load ramping the conventional unit control is not capable for moving the operation point from minimum level to maximum level fast and efficiently, but a coordinated high level control system is needed. A prerequisite for wide range ramping is that the boiler – turbine system is controlled as a whole, all subsystems parallel, not only by a feed back system which eliminates disturbances spreading from one sub-process to another during a load change.

As a conclusion of the control chapter is that the current status of steam power plants control is designed and optimized primarily for disturbance rejection, to keep the process in the designed operation point as effectively as possible. New requirements set for flexibility in steam power plant operation require a novel approach to power plant control. Constraints related to thermal stresses and emission control do not make the task any simpler. Besides of pure power management, the operation environment must be controlled to be able to make this kind of operation as beneficial and environmentally friendly as possible.

4 Discussion

The present report focuses on engine- and combustion-based energy technologies that are commonly available and in use at present. The situation in the energy market changes fast, which makes it difficult to predict, what the role of engine- and combustion-based power production will be in the future. Simultaneously, tightening emission limits will affect the cost-effictiveness of the plants and retrofitting existing plants will often become necessary. New energy technologies for energy storage, for heat generation (e.g. deep heat), and for hybrid energy and chemical industry processes also continuously emerge and they can change the market situation and the need for flexible capacity. Similarly, better integration of the consumers and their energy consumption and storage possibilities in the energy network will in the future help to reduce the need of flexible energy production. At present, however, no such fast drastic changes are visible that would remove the need for flexible combustion-based energy. Engine-based production provides a good option for very fast load changes. Although boiler-based units allow clearly slower ramp rates they have their market due to excelent fuel flexibility.

5 Summary

The report summarizes flexibility limitations of engine– and boiler-based energy production. To allow the reader to easier understand the limitations, the operation principles of the engines and the boiler-based plants are also shortly described.

Engine-based units offer excellend load following capability and can easily respond to very fast power change demands. Alhough small amount of addidional power can be obtained fast from the heat reserves of boilers, boiler-based units react in general slower to load changes. Fluctuations in production of solar and wind power are however not in second scale but more in minute and hour scale and boilers can react in this time scale. The biggest challenge for boilers comes from the need for low load operation or temporary shutdowns in situations when the amount of solar and wind power is large. Startup of a boiler is a slow operation and to reduce the need for shutdowns, the goal of energy producers is today to run the boiler at as low load as possible in these situations. The report discusses the problems related to low load operation and fast load changes.

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