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Biomass Energy for Heating and Hot Water Supply in Belarus

Best Practice Guidelines

Part A: Biomass Combustion

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Part A: Biomass Combustion

Colophon

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1 INTRODUCTION

In the frame of the UNDP/GEF project Biomass Energy for Heating and Hot Water Supply in Belarus a range of informative documents on bio-energy will be produced, including:

(a) A set of Fact Sheets for different target audiences, presenting various aspects of biomass fuel supply and bioenergy conversion
(b) A Wood Energy Brochure, presenting technological, environmental and financial aspects of biomass energy systems for persons without formal technical education
(c) A Best Practice Guidebook, describing selected issues discussed in the Wood Energy Brochure more comprehensively for persons with formal technical education (university and polytechnic student and their teachers).

This document constitutes the first draft of the "biomass combustion" section of the Best Practice Guidebook. The section on "biofuel supply" remains to be completed.

The focus of this publication is on the combustion of wood. Limited attention is given to the combustion of other types of biomass. The focus is further on combustion plants with automatic system operation and capacities between 0.1 and 10 MW_{th}.

Subjects covered in this document include:
- Section 2: Properties and combustion characteristics of wood
- Section 3: Industrial biomass combustion concepts
- Section 4: Power generation and co-generation by biomass combustion
- Section 5: Emissions from biomass combustion
- Section 6: Case Studies

2 PROPERTIES AND COMBUSTION CHARACTERISTICS OF WOOD

2.1 Introduction

The characteristics and quality of wood as a fuel vary widely, depending mainly on the type of wood and the pre-treatment applied. For example, the moisture content of the fuel as fed into the furnace may vary from 25-55% wt% (w.b.) (bark, sawmill side-products) or drop below 10% wt% (w.b.) (pellets, briquettes, dry wood processing residues).

There are several characteristics affecting the properties of wood as a fuel. These include heating value, chemical composition (e.g. content of such elements as chlorine Cl, carbon C, hydrogen H, nitrogen N and sulphur S), moisture content, density, hardness, the amount of volatile matters, the amount of solid carbon, ash content and composition, the melting behaviour of ash, the slagging behaviour of ash, the amount of impurities, dust and fungi spores. Wood fuel chips, for instance, are often made of various tree species with various proportions of wood, bark, foliage, twigs (branches), buds and even cones. This causes variation in the fuel properties.

2.2 Wood composition

The main components of wood cells are cellulose[1], hemicellulose[2] and lignin[3], forming some 99% of the weight of the wood material. Cellulose and hemicellulose are formed by long chains of carbohydrates (such as glucose), whereas lignin is a complicated component of polymeric phenolics. Lignin has a close relationship with hemicellulose, as it acts as a glue fixing the bunches of cellulose chains and planting tissues together. Thus it gives mechanical strength to the plant. Lignin is rich in carbon and hydrogen, which are the main heat producing elements. Hence lignin has a higher heating value than carbohydrates. Wood and bark also contain so-called extractives, such as terpenes, fats and phenols. Many of them are soluble in organic solvents (hexane, acetone, ethanol) and in hot water. The amount of wood extractives is relatively small when compared to the amount of extractives from bark and foliage.

Approximately one half of fresh, just fallen tree is water. The other half consists of dry matter of wood, approx. 85% of which consists of volatile matters, 14.5% of solid carbon and 0.5% of ash (see Figure 2.1). In water-free wood, the total content of the carbon element is about 50%. When wood is combusted, its components will change into steam of water (H_2O), carbon dioxide (CO_2), nitric oxides (NO_x), sulphur oxide (SO_2) and ash. Wood has practically no sulphur at all, as its share in wood is 0.05% at the highest.

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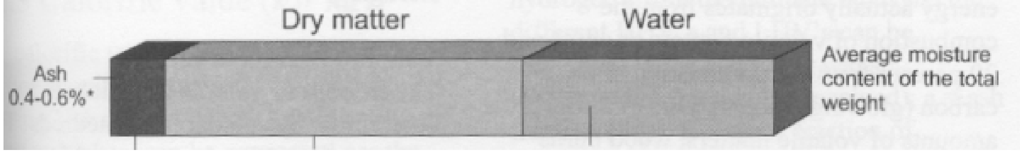
[1] Cellulose ($C_6H_{10}O_5$) is a condensed polymer of glucose (C6H,00r,). The fibre walls consist mainly of cellulose and represent 40 to 45% of the dry weight of wood.

[2] Hemicellulose consists of various sugars other than glucose that encase the cellulose fibres and represent 20 to 35% of the dry weight of wood.

[3] Lignin ($C_{40}H_{44}O_6$) is a nonsugar polymer that gives strength to the wood fibre, accounting for 15 to 30% of the dry weight.

Figure 2.1 Average chemical contents of wood fuels



SOLID	VOLATILE MATERIALS		BARK	ca. 60 %
CARBON (C)	84-88%*		SAWDUST	ca. 55 %
11.4-15.6%*	Carbon (C)	ca. 35.5 % → CO, CO_2,	FOREST	
→ CO_2	Hydrogen (H)	6.0 - 6.5% → H,O	CHIPS	ca. 40 %
	Oxygen (O)	38 - 42% →	CHOPPED	
	Nitrogen (N)	0.1 - 0.5% → NO_x	FIREWOOD	ca. 25 %
	Sulphur (S)	max. 0.05% → SO_2	BRIQUETTES	ca. 5 %

* Share % of dry matter weight

The nitrogen content of wood is in average about 0.75%, varying somewhat from one tree species to another. The wood chips, for instance, made from so-called nitrogen-fixing trees, such as alder (*Ainus sp.*), contain more than twice as much nitrogen as chips made of coniferous trees like pine (*Pinus sp.*) and spruce (*Picea sp.*). The bark of wood also contains more nitrogen than wood material.

The heating properties of different fuels depend on the proportions of the elements they contain. Carbon and hydrogen increase the heating value, whereas a high share of oxygen in wood decreases it. Compared to many other fuels, wood has a fairly low carbon content (some 50% of dry weight) and high oxygen content (some 40%), which leads to a fairly low heating value per dry weight. Dry wood and bark also have quite low ash contents, as one solid cubic metre of wood fuels produces in average only 3-5 kg of clean ash. In practice, however, there is often some sand and unburned carbon in the ash.

The combustibles of solid fuels can be shared into two groups: volatile matters and components combusting as solid carbon. The share of volatile matters in wood is typically high, whereas the share of solid carbon is low. Eighty percent of wood energy actually originates from the combustion of volatile matters or gases and twenty percent from the combustion of solid carbon (glowing embers). Due to the large amounts of volatile matters, wood burns with long flames and therefore needs a lot of space for combusting. The bark of wood is similar to peat when combustion properties are considered.

2.3 Proximate and ultimate analysis

For the determination of wood fuel properties two types of analysis are used. *Proximate analysis* is the determination, by given prescribed methods, of the moisture content (ISO 331), volatile matter content (ISO 562), ash content (ISO 1171) and fixed carbon content (ISO 609) of a fuel. *Ultimate analysis* is the determination, by given prescribed methods, of the elemental composition of a fuel.

2.3.1 Proximate analysis

In the proximate analysis of wood fuels, such properties as fixed carbon, volatile materials, ash and moisture contents are defined in the following ways.

Ash

Ash content expressed in weight% (wt%) of dry base (dry) and of as received (ar) material. Through the moisture content the different types of ash contents are related to each others:

Ash content (wt% dry) = ash content (wt% ar) * 100 / (100 - moisture content (wt%))

Moisture content

Moisture content in weight% of wet base (as received). It is important to note that there can be a large difference in the moisture contents of the material between the time it is actually available and the time it is analysed. Also, natural drying during storage can lower the moisture content.

Volatiles and fixed carbon

The amount of volatile materials is determined by standardised methods. The amount of volatiles is expressed in weight% of dry material, as received material (ar) or dry and ash free material (daf). The amount of fixed (solid) carbon is calculated according to the following formulas as the remaining part:

dry fixed C = 100 - ash (dry) - volatiles (dry)
daf fixed C = 100 - volatiles (daf)
ar fixed C = 100 - ash (ar) - water content - volatiles (ar)

2.3.2 Ultimate analysis

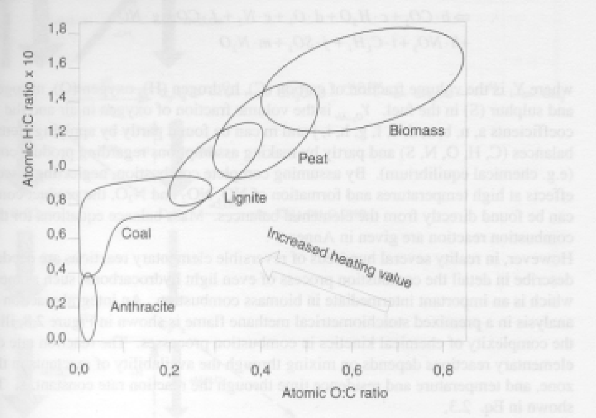
In ultimate analysis, the share of different elements of dry material is defined in the following manner: Carbon (C), hydrogen (H), oxygen (O), nitrogen (N), sulphur (S), chlorine (Cl), fluorine (F) and bromine (Br) content in weight % of dry material (wt% dry), dry and ash free material (wt% daf) and on as received material (wt% ar).

dry C + H + 0 + N + S + Cl + F + Br + ash = 100
daf C + H + 0 + N + S + Cl + F + Br = 100
ar C + H + 0 + N + S + Cl + F + Br + ash + water content = 100

In many cases the oxygen content is not measured but calculated as the difference between 100 and the measured components. When the oxygen content is measured, the total sum can exceed 100% due to experimental errors in the analysis. For each component it is indicated whether it has been measured or calculated.

Compared to other solid fuels, biomass contains relatively much hydrogen and oxygen, as is illustrated in Figure 2.2.

Figure 2.2 Chemical composition of various sold fuels



2.4 Moisture content and calorific value (MJ/kg)

In general, the *moisture content* of wood fuels varies usually from 20 to 65 percent and is influenced, among other things, by the climatic conditions, the time of year, tree species, the part of stem in question and by storage phase. The *calorific value* is expressed either as Gross Calorific Value[4] (GCV) or Net Calorific Value[5] (NCV). The heating value, gross or net, can be expressed per the dry fuel unit (normally kg or m^3) or per the fuel unit including the moisture. In addition to the moisture content of fuel the moisture is born in the combustion of hydrogen. The state of moisture makes the difference between gross and net calorific value. The GCV is calculated with assumption that moisture is condensed to water and NCV with assumption that moisture is in form of saturated steam. The unit is usually MJ/kg.

The GCV of biomass fuels usually varies between 18 and 21 MJ/kg (d.b.), which is similar to the GCV of peat but significantly lower than the GCV of oil (see Table 2.1).

Table 2.1 Thermal values for selected fuels

Fuel	**H_i[6] (MJ/kg)**
Wood (dry)	18.5-21.0
Peat (dry)	20.0-21.0
Carbon	23.3-24.9
Oil	40.0-42.3

The GCV of biomass can be calculated reasonably well by using the following empirical formula:

$$GCV = 0.3491 \cdot X_C + 1.1783 \cdot X_H + 0.1005 \cdot X_S - 0.0151 \cdot X_N - 0.1034 \cdot X_O - 0.0211 \cdot X_{ash} \text{ [MJ/kg d.b.]}$$

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[4] The gross calorific value (GCV) is also referred to as higher heating value (HHV) or combustion heat

[5] The net calorific value (NCV) is also referred to as lower heating value (LHV) or effective heating value

[6] *Effective thermal value*, H_i, is used when the existing water after the combustion is in the form of steam. This is the most common thermal value in technical combustion processes as the exhaust gas seldom is cooled to a level that makes the steam condense into water. *Calorimetric heating value* is used when the existing water after the combustion is in liquid form.

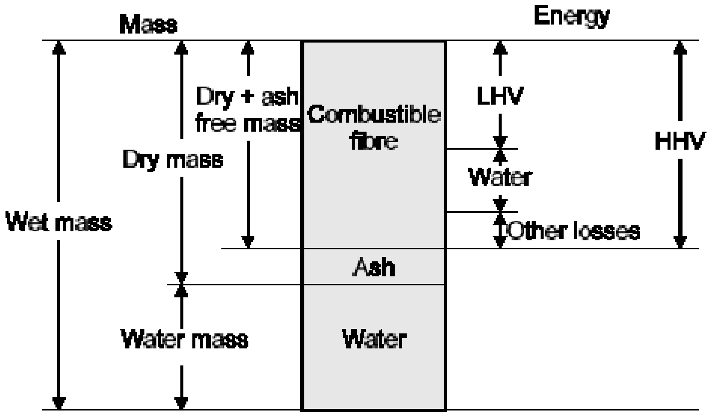
where X_I is the content of carbon (C), hydrogen (H), sulphur (S), nitrogen (N), oxygen (0) and ash in wt% (d.b.). As can be seen from the formula, the content of C, H and S contributes positively to GCV, while the content of N, O and ash contributes negatively to GCV. The net calorific value (NCV, MJ/kg, w.b.) can be calculated from GCV taking into account the moisture and hydrogen content of the fuel, as follows

$$GCV_{ar} = GCV_{dry} * (1\text{-}w/100)$$
$$GCV_{dry} = GCV_{daf} * (1\text{-}a/100)$$
$$NCV_{dry} = GCV_{dry} - 2.442 * 8.396 * H/100$$
$$NCV_{ar} = NCV_{dry} * (1\text{-}w/100) - 2.442 * w/100$$
$$NCV_{ar} = GCV_{dry} - 2.442 * \{8.396 * H/100 * (1\text{-}w/100) + w/100\}$$

where w = moisture fraction (as received); a = ash fraction (dry); H = mass fraction of hydrogen in sample (dry).

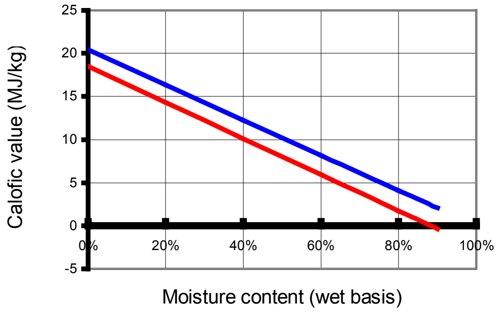
Figure 2.3 illustrates the relationship between the several heating values

Figure 2.3 Relationship between several heating value definitions



As Figure 2.3 shows the moisture content significantly influences the calorific value since vaporising water requires energy. The effect of moisture content on the heating value of wood is further illustrated in Figure 2.4.

Figure 2.4 The effect of moisture content on the heating value of wood



Explanation: net and gross calorific values (NCV & GCV) of barkless wood as a function of the moisture content in percentage of total weight. Red line: NCV. Blue line: GCV.

2.5 Average properties of wood chips

This section explores the most important fuel properties of wood (chips), which include:

- Moisture content
- Density
- Heating value
- Particle size distribution
- Ash content and properties
- Chemical composition
- Amount of volatiles
- Results of proximate and ultimate analysis.

The *moisture content* of fresh wood chips depends on the production method. The moisture content (m.c.) of wood chips produced from green trees is approx. 50-60% of the total weight, but after summer drying of the trees for 3-6 months the moisture content is reduced to approx. 35-45% of the total weight.

The *solid volume content* of chips indicates the relationship between the masses of so-called bulk measure and solid measure, that is, how many solid m^3 one bulk m^3 will yield. The solid volume content of chips is influenced mostly by the technical specifications of the chipper, such as particle size distribution, blowing power and loading method. The drying time of chips and the compacting that occurs during long-distance transport, however, have no decisive effect on the solid volume content value. Solid volume content (the portion of solid measure) is needed for converting bulk measure into solid measure. The *bulk density* of Austrian beech (40% mc_{db}) is some 327 kg/loose m^3, for Austrian spruce 221 kg/loose m^3.

The *particle size* and moisture content of direct wood fuels or forest fuels are often very heterogeneous. The particle size varies from sawdust, needle and bark material to sticks of wood and branch pieces. The size of the wood particles is influenced both by the original raw material being chipped and by the chipper types. The more stemwood the raw material contains, the more even the particle size distribution will be. The condition of chipper knives as well as the aperture size of the screen in the chipper also influence the particle size. Chips produced with crushers have typically coarser particles compared to the chips produced with chippers.

The *calorific heating value* of wood chips does not vary a great deal from one tree species to another, but it is slightly higher in coniferous species than in broad-leaved or deciduous tree species.

The structural elements (*ultimate analysis*) of the organic portion of wood are carbon (45-50%), oxygen (40-45%), hydrogen (4.5-6%) and nitrogen (0.3-3.5%). The ash composition of tree species is usually less than a few percent (0.3% in spruce or birch without bark, 1.6% in birch bark and 3.4% in spruce bark).

The distinct advantage of woody biomass over fossil fuels is the small amount of sulphur. The ultimate analysis of some tree species shows that carbon and hydrogen contents are rather uniform among species. Bark has a higher percentage of carbon and hydrogen than wood. This is most visibly the case with birch and alder. In the *proximate analysis* the amount of volatiles is 65-95%, fixed carbon 17-25% and ash content 0.08 - 2.3%.

Many sources present details on the properties of wood fuels. The most comprehensive database of wood fuel properties, Phyllis, is maintained by ECN, the energy research centre of the Netherlands, and can be accessed via URL: http://www.ecn.nl/phyllis/

Table 2.2 presents the moisture content, heating values, bulk density and energy density of various woodfuels.

Table 2.2 Physical characteristics of selected woodfuels

woody material	**moisture content [wt% wb]**	**GCV [(kWh/kg (db.)]**	**NCV [(kWh/kg (d.b.)]**	**Bulk density [kg (wb) /m^3]**	**Energy density [kWh/m^3]**
wood pellets	10.0	5.5	4.6	600	2,756
wood chips (hardwood, pre-dried)	30.0	5.5	3.4	320	1,094
wood chips (hardwood)	50.0	5.5	2.2	450	1,009
wood chips (softwood, pre-dried)	30.0	5.5	3.4	250	855
wood chips (softwood)	50.0	5.5	2.2	350	785
Bark	50.0	5.6	2.3	320	727
Sawdust	50.0	5.5	2.2	240	538

Abbreviations: GCV = gross calorific value, NCV = net calorific value, db = dry basis, wb= wet basis.

2.6 Theory of wood firing

Efficient and complete combustion is a prerequisite of utilising wood as an environmentally desirable fuel. In addition to a high rate of energy utilisation, the combustion process should therefore ensure the complete destruction of the wood and avoid the formation of environmentally undesirable compounds.

Emissions caused by incomplete combustion are mainly a result of either:

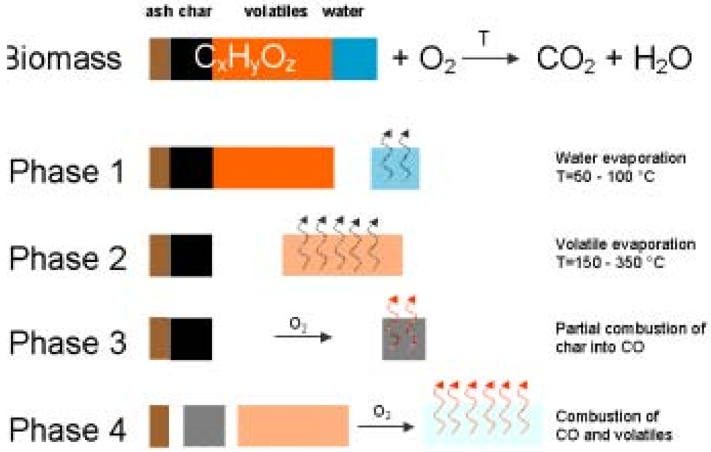
- inadequate mixing of combustion air and fuel in the combustion chamber, giving room for local fuel-rich combustion zones;
- an overall lack of available oxygen;
- too low combustion temperatures;
- too short residence times;
- too low radical concentrations, in special cases, for example in the final stage of the combustion process (the char combustion phase) in a batch combustion process.

These variables are all linked together. However, in cases in which oxygen is available in sufficient quantities, temperature is the most important variable due to its exponential influence on the reaction rates. An optimisation of these variables will in general contribute to reduced emission levels of all emissions from incomplete combustion.

2.6.1 Stages of wood combustion

Drying and pyrolysis/gasification are the first steps in a solid fuel combustion process. However, the relative importance of these steps varies, depending on the combustion technology implemented, the fuel properties and the combustion process conditions. A separation of drying/pyrolysis/gasification and gas and char combustion, as in staged-air combustion, may be utilised. In large-scale biomass combustion applications with continuous fuel feeding, such as moving grates, these processes will occur in various sections of the grate. However, in batch combustion applications there will be a distinct separation between a volatile and a char combustion phase, also with time. Figure 2.5 shows qualitatively the combustion process for a small wood particle.

Figure 2.5 The wood combustion route



For larger particles, there will be a certain degree of overlap between the phases, while in batch combustion processes, as in wood log combustion in wood-stoves and fireplaces, there will be a large degree of overlap between the phases.

Drying: Moisture will evaporate already at low temperatures (50-100°C). Since the vaporisation uses energy released from the combustion process, it lowers the temperature in the combustion chamber, which slows down the combustion process. In wood-fired boilers it has been found that the combustion process cannot be maintained if the wood moisture content exceeds 60% on a wet basis (w.b.). The wet wood requires so much energy to evaporate contained moisture, and subsequently to heat the water vapour, that temperatures are reduced below the minimum temperature required to sustain combustion. Consequently, moisture content is a very important fuel variable.

Pyrolysis can be defined as thermal degradation (devolatilisation) in absence of an externally supplied oxidising agent. The pyrolysis products are mainly tar and carbonaceous charcoal, and low molecular weight gases. Also CO and CO_2 can be formed in considerable quantities. Fuel type, temperature, pressure, heating rate and reaction time are all variables that affect the amounts and properties of the products formed.

Devolatilisation of wood starts at 200°C and the devolatilisation rate increases as the temperature is raised. First the hemicellulose of the wood decomposes at higher temperature the cellulose. At 400°C, most of the volatiles are gone and the devolatilisation rate decreases rapidly. However, a low devolatilisation rate can be observed in the temperature range of 400-500^{o}C. This is caused by lignin decomposition, which occurs throughout the whole temperature range, but the main area of weight loss occurs at higher temperatures.

Gasification can be defined as thermal degradation (devolatilisation) in the presence of an externally supplied oxidising agent. However, the term gasification is also used for char oxidation reactions with for example CO_2 or H_2O. While pyrolysis is usually optimised with respect to a maximum char or tar yield, gasification is optimised with respect to a maximum gas yield. Temperatures of 800-1100 ^{o}C are used. The gas contains mainly CO, CO_2, H_2O, H_2, CH_4 and other hydrocarbons. Gasification can be carried out with air, oxygen, steam or CO_2 as oxidising agents.

Combustion can ideally be defined as a complete oxidation of the fuel. The hot gases from the combustion may be used for direct heating purposes in small combustion units, for water heating in small central heating boilers, to heat water in a boiler for electricity generation in larger units, as a source of process heat, or for water heating in larger central heating systems. Drying and pyrolysis/gasification will always be the first steps in a solid-fuel combustion process.

2.6.2 Important variables in biomass combustion

Biomass combustion is a complex process, involving many variables that directly or indirectly influence emission levels and energy efficiency. Variables that are of importance (mainly in large-scale biomass combustion applications) are briefly described below.

Heat transfer mechanisms: Heat can be transferred by conduction, convection of radiation. To achieve low emission levels of emissions from incomplete combustion, it is necessary to minimise heat losses from the combustion chamber, which is done by optimising those variables that directly affect the heat transfer mechanisms. However, to achieve a high thermal efficiency, efficient heat exchange is necessary between the combustion chamber and the chimney inlet.

Heat storage: A significant amount of heat will accumulate in the walls of the combustion chamber, stealing heat from the combustion chamber in the start-up phase. This is of special importance in small-scale biomass combustion applications. The stored heat will be transferred to the surroundings with a significant time delay, which is positively utilised in heat storing stoves (heavy stoves). However, high emission levels of emissions from incomplete combustion may be found in the start-up phase.

Insulation: Heat is transferred by conduction through the walls of the combustion chamber. Consequently, by improving the insulation of the combustion chamber, a higher combustion chamber temperature can be achieved. The insulation can be improved by either increasing the thickness of the insulation or using a material that insulates better. However, insulation occupies space and is an additional expense, and should be utilised with care.

Air pre-heating: The combustion chamber temperature can be significantly increased by air pre-heating. The inlet air may be pre-heated through heat exchange with the flue gas, after the flue gas has left the combustion chamber. Stealing heat directly from the combustion chamber for air pre-heating will have no effect, unless the goal is to reduce the temperature in one part of the combustion chamber, by moving heat to another part. One example is pre-heating of secondary air at the expense of the fuel bed temperature.

Excess air ratio: A given fuel requires a given amount of air (oxygen) in order to be converted stoichiometrically, i.e. the amount of excess air λ (lambda) should be equal to 1. The fuel is converted stoichiometrically when the exact amount of oxygen that is required for the conversion of all of the fuel under ideal conditions is present. In biomass combustion applications, it is necessary to have an excess air ratio well above 1, to ensure a sufficient mixing of inlet air and fuel gas. In small-scale applications the excess air ratio usually has to be above 1.5. This means that there will be an overall excess of oxygen. The combustion temperature will be significantly reduced, compared to the stoichiometrical combustion temperature, mainly due to heating of inert nitrogen in the air. Hence, an optimal mixing of air and fuel is of the utmost importance, enabling operation at lower overall excess air ratios, with increased combustion temperatures. Optimal design of the air inlets and advanced process control optimisation is necessary to ensure sufficient mixing at very low excess air.

Fuel type: The fuel type influences the combustion process through various characteristics of different fuel types, mainly with respect to fuel composition, volatile/char content, thermal behaviour, density, porosity, size and active surface area. The **fuel composition** is important with respect to GCV and emissions, though mainly emissions from complete combustion (see Section 5.2), and ash-related problems. In batch combustion applications, the fuel composition will vary continuously as a function of degree of burnout. Biomass generally contains a high **volatile content** and low **char content** compared to coal, which makes biomass a highly reactive fuel. However, the volatile content varies for different biomass fuels, and influences the thermal behaviour of the fuel. The thermal behaviour of a fuel is also influenced by the different chemical structures and bonds present in different biomass fuels. This results in significantly different devolatilisation behaviour as a function of temperature. However, when wood is used similar thermal behaviour can be observed. The **density** of different biomass fuels is highly variable, and a significant difference can also be found between hardwoods and softwoods. Hardwoods, such as birch, have a higher density, which influences the combustion chamber volume to energy input ratio, and also the combustion characteristics of the fuel. The **porosity** of the fuel influences the reactivity (mass loss per time unit) of the fuel, and thereby its devolatilisation behaviour. **Fuel size** is an important variable in large-scale biomass combustion applications, especially where entrainment of fuel particles in the flue gas occurs, as in pulverised fuel combustion. Smaller fuel particles will need a shorter residence time in the combustion chamber. The homogeneity of the fuel is also of importance: increasing homogeneity, which improves with decreasing fuel size, enables better process control. Finally, the **active surface area** of the fuel influences the reactivity of the fuel.

Moisture content: The significance of the moisture content has already been described in Section 2.4. However, in batch combustion applications an additional complication is introduced: the moisture content will vary continuously as a function of burnout. The moisture will be released in the devolatilisation phase, and the moisture content decreases as a function of burnout. Hence, the moisture content and its negative effects on the combustion process may be substantial in the early stages of the devolatilisation phase, resulting in high emission levels of emissions from incomplete combustion.

Combustion temperature: The significance of sufficiently high combustion temperatures has already been described. However, in batch combustion applications an additional complication is introduced: the moisture content and fuel composition will vary continuously as a function of burnout. This will influence the adiabatic combustion temperature. The adiabatic combustion temperature will increase as a function of degree of burnout at a constant excess air ratio. However, as char is much less reactive than the volatile fraction of biomass fuels, the fuel consumption rate and the oxygen need will be much lower. Since it is usually difficult to effectively control the amount of air supplied in the char combustion phase, especially if natural draught is applied, the excess air ratio will be quite high. This, together with a much lower fuel consumption rate, may decrease the temperature in the combustion chamber below the level needed for complete combustion. However, the higher heating value of char will to some extent compensate for the much lower fuel consumption rate in the char combustion phase. The **residence time** needed for complete combustion is directly influenced by the combustion temperature and to some extent by the mixing time.

Design: From the variables described above it is clear that the combustion application design significantly influences the combustion process through the construction and operational principle of the combustion chamber, through the choice of materials, and through process control possibilities. The **materials** used, mainly their heat capacity, density, thickness, insulating effect and surface properties, influence the combustion chamber temperatures.

Heat exchange: Proper heat exchange is necessary to achieve high thermal efficiency. This can be arranged in many ways by using different kinds of heat exchangers before the flue gas reaches the chimney. To be able to control the heat exchange active process control systems using process control variables, for example the amount of water flowing through the boiler, must be applied.

Air staging: By applying staged-air combustion, a simultaneous reduction of both emissions from incomplete combustion and NO_x emissions is possible through a separation of devolatilisation and gas phase combustion. This results in improved mixing of fuel gas and secondary combustion air, which reduces the amount of air needed, providing a lower local and overall excess air ratio and higher combustion temperatures. Hence, emissions from incomplete combustion are reduced by a temperature increase, which speeds up the elementary reaction rates, and by an improved mixing which reduces the residence time needed for mixing of fuel and air. See Section 5.5.5 for further information.

Air distribution: An efficient air distribution is of the utmost importance to achieve an effective reduction of both emissions from incomplete combustion and NO_x emissions in staged-air systems. The distribution of primary and secondary air, within the combustion chamber and the flame zone, influences the mixing quality of air and fuel, and thus the residence time, and subsequently the combustion temperature needed for complete combustion.

Fuel feeding: Any batch combustion application will benefit from a more continuous combustion process, in which the negative effects of the start-up phase and the char phase are reduced. This is partly achieved manually in wood log boilers, by semi-batch operation.

Fuel distribution: The distribution of fuel inside the combustion chamber, reducing or increasing the active surface area, will influence the combustion process through a decreasing or increasing reactivity, respectively.

Heat distribution: Heat distribution is closely related to heat exchange and fuel distribution, and in addition to several other variables influences the combustion chamber temperature in different sections of the combustion chamber, and the heat transfer after the combustion chamber.

Regulation: By applying efficient combustion process control, emission levels can be minimised and thermal efficiency can be optimised. Several different methods for combustion process control have been developed (see Section 5.5.4). These can be based on measurements of specific flue gas compounds or temperatures, which then will provide a combustion process controller with the necessary information needed to change the combustion process, for example by changing the amount and distribution of air fed to the combustion chamber.

A further aspect that is of great importance in large-scale biomass combustion applications are the problems caused by the utilisation of low-quality and cheap biofuels. This often results in depositions and corrosion on heat exchangers and superheaters, and additional emissions caused by a higher nitrogen, sulphur, chlorine, fluorine, potassium, and sodium content in the fuel, than there is in wood.

From the above it can be concluded that the combustion process, and therefore emission levels and energy efficiency, is influenced by a great many variables. These should be kept in mind when designing and operating any biomass combustion application.

3 INDUSTRIAL BIOMASS COMBUSTION CONCEPTS

3.1 Introduction

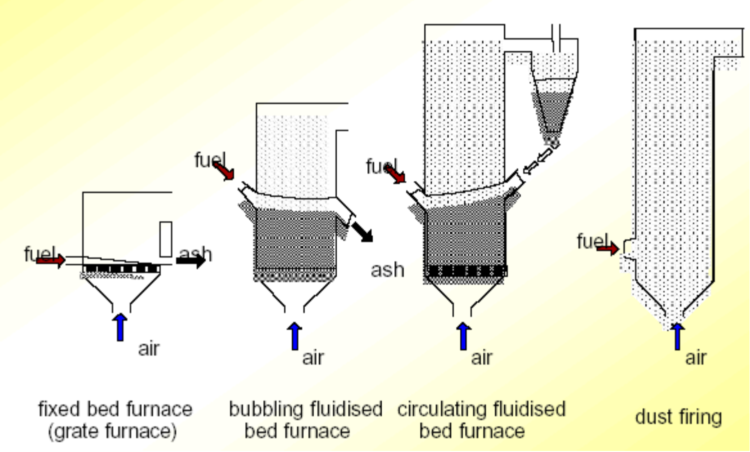
This chapter describes combustion systems of a nominal thermal capacity exceeding 100 kW. These furnaces are generally equipped with mechanic or pneumatic fuel-feeding. Manual fuel-feeding is no longer customary due to high personnel costs and strict emission limits. Moreover, modern industrial combustion plants are equipped with process control systems supporting fully automatic system operation.

In principle, the following combustion technologies can be distinguished:

- fixed-bed combustion
- (bubbling and circulating) fluidised bed combustion
- dust combustion.

The basic principles of these technologies are shown in Figure 3.1 and described below.

Figure 3.1 Diagram of principal industrial biomass combustion technologies



Fixed-bed combustion systems include grate furnaces and underfeed stokers. Primary air passes through a fixed bed, in which drying, gasification, and charcoal combustion takes place. The combustible gases produced are burned after secondary air addition has taken place, usually in a combustion zone separated from the fuel bed.

Within a **fluidised bed furnace**, biomass fuel is burned in a self-mixing suspension of gas and bed material into which combustion air enters from below. Depending on the fluidisation velocity, bubbling fluidised bed and circulating fluidised bed combustion can be distinguished.

Dust combustion is suitable for fuels available as small particles (average diameter smaller than 2 mm). A mixture of fuel and primary combustion air is injected into the combustion chamber. Combustion takes place while the fuel is in suspension, and gas burnout is achieved after secondary air addition. Variations of these technologies are available. Examples are combustion systems with spreader stokers and cyclone burners.

3.2 Fixed-bed combustion

3.2.1 Grate furnaces

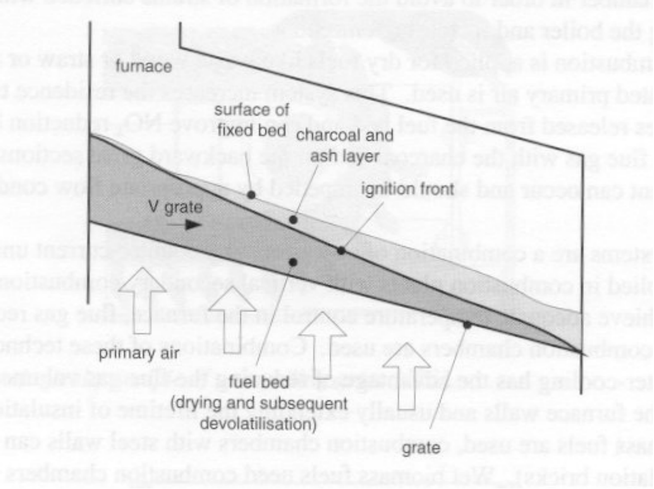
There are various grate furnace technologies available: fixed grates, moving grates, travelling grates, rotating grates, and vibrating grates. All of these technologies have specific advantages and disadvantages, depending on fuel properties, so that careful selection and planning is necessary.

Grate furnaces are appropriate for biomass fuels with a high moisture content, varying particle sizes (with a downward limitation concerning the amount of fine particles in the fuel mixture), and high ash content. Mixtures of wood fuels can be used, but current technology does not allow for mixtures of wood fuels and straw, cereals and grass, due to their different combustion behaviour, low moisture content, and low ash-melting point. A good and well-controlled grate is designed to guarantee a homogeneous distribution of the fuel and the bed of embers over the whole grate surface. This is very important in order to guarantee an equal primary air supply over the various grate areas. Inhomogeneous air supply may cause slagging, higher fly-ash amounts, and may increase the excess oxygen needed for a complete combustion. Furthermore, the transport of the fuel over the grate has to be as smooth and homogeneous as possible in order to keep the bed of embers calm and homogeneous, to avoid the formation of "holes" and to avoid the elutriation of fly ash and unburned particles as much as possible.

The technology needed to achieve these aims includes continuously moving grates, a height control system of the bed of embers (e.g. by infrared beams), and frequency-controlled primary air fans for the various grate sections. The primary air supply divided into sections is necessary to be able to adjust the specific air amounts to the requirements of the zones where drying, gasification, and charcoal combustion prevail (see Figure 3.2). This separately controllable primary air supply also allows smooth operation of grate furnaces at partial loads of up to a minimum of about 25% of the nominal furnace load and control of the primary air ratio needed (to secure a reducing atmosphere in the primary combustion chamber necessary for low NO_x operation). Moreover, grate systems can be water-cooled to avoid slagging and to extend the lifetime of the materials.

Another important aspect of grate furnaces is that a staged combustion should be obtained by separating the primary and the secondary combustion chambers in order to avoid back-mixing of the secondary air and to separate gasification and oxidation zones. Due to the fact that the mixing of air and flue gas in the primary combustion chamber is not optimal because of the low turbulence necessary for a calm bed of embers on the grate, the geometry of the secondary combustion chamber and the secondary air injection have to guarantee a mixture of flue gas and air that is as complete as possible. The better the mixing quality between flue gas and secondary combustion air, the lower the amount of excess oxygen that will be necessary for complete combustion and the higher the efficiency. The mixing effect can be improved with relatively small channels where the flue gas reaches high velocities and where the secondary air is injected at high speed via nozzles that are well distributed over the cross-section of this channel. Other means of achieving a good mixture of flue gas and secondary air are combustion chambers with a vortex flow or cyclone flow.

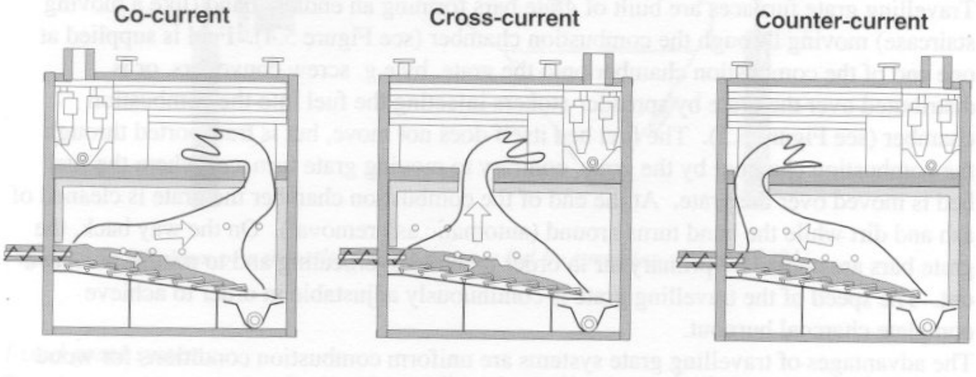
Figure 3.2 Diagram of the combustion process in fixed fuel beds



Based on the flow directions of fuel and the flue gas, there are various systems for grate combustion plants (Figure 3.3):

- counter-current flow (flame in the opposite direction as the fuel),
- co-current flow (flame in the same direction as the fuel),
- cross-flow (flue gas removal in the middle of the furnace).

Figure 3.3 Classification of grate combustion technologies: co-current, cross-current and counter-current



Counter-current combustion is most suitable for fuels with low heating values (wet bark, wood chips, or sawdust). Due to the fact that the hot flue gas passes over the fresh and wet biomass fuel entering the furnace, drying and water vapour transport from the fuel bed is increased by convection (in addition to the dominating radiant heat transfer to the fuel surface). This system requires a good mixing of flue gas and secondary air in the combustion chamber in order to avoid the formation of strains enriched with unburned gases entering the boiler and increasing emissions.

Co-current combustion is applied for dry fuels like waste wood or straw or in systems where pre-heated primary air is used. This system increases the residence time of unburned gases released from the fuel bed and can improve NO_x reduction by enhanced contact of the flue gas with the charcoal bed on the backward grate sections. Higher fly-ash entrainment can occur and should be impeded by appropriate flow conditions (furnace design).

Cross-flow systems are a combination of co-current and counter-current units and are also especially applied in combustion plants with vertical secondary combustion chambers. In order to achieve adequate temperature control in the furnace, flue gas recirculation and water-cooled combustion chambers are used. Combinations of these technologies are also possible. Water-cooling has the advantage of reducing the flue gas volume, impeding ash sintering on the furnace walls and usually extending the lifetime of insulation bricks. If only dry biomass fuels are used, combustion chambers with steel walls can also be applied (without insulation bricks). Wet biomass fuels need combustion chambers with insulation bricks operating as heat accumulators and buffering moisture content and combustion temperature fluctuations in order to ensure a good burnout of the flue gas. Flue gas recirculation can improve the mixing of combustible gases and air and can be regulated more accurately than water-cooled surfaces. However, it has the disadvantage of increasing the flue gas volume in the furnace and boiler section. Flue gas recirculation should be performed after fly-ash precipitation in order to avoid dust depositions in the recirculation channels. Moreover, flue gas recirculation should not be operated in stop-and-go mode, to avoid condensation and corrosion in the channels or on the fan blades.

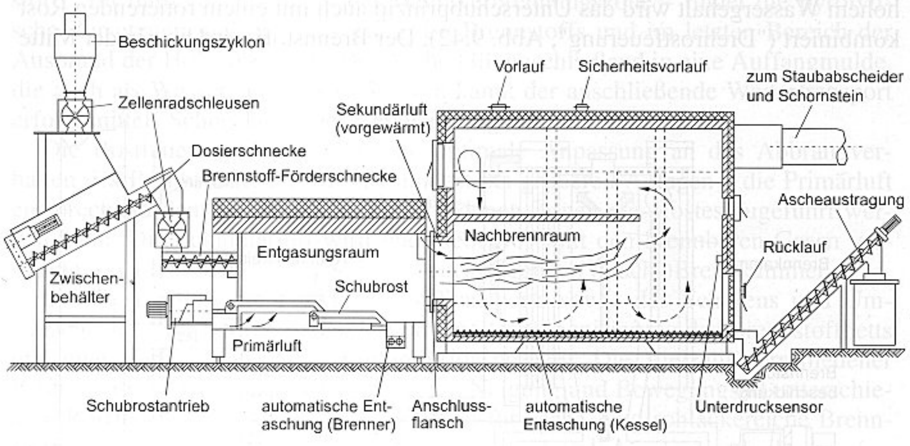
Dual-chamber furnace

In a dual-chamber furnace primary and secondary combustion is achieved in physically separated modules. Dual-chamber furnaces are used when relatively wet wood chips are used as fuel. Dual-chamber furnaces consist of a fuel supply screw with backfire protection, a well-insulated chamber with grate (usually of the co-current type) and a separate boiler module. In the well-insulated chamber (known as pre-furnace or pre-oven) primary air is fed with a fan and combustion or partial gasification takes place. Only a small amount of fuel is burned at a time. Using pre-heated secondary air the fuel gases and flue gases are let through a flange into the downstream boiler module. Depending on the configuration, oxidation takes place in the boiler module, before the hot gases are led into the heat exchanger. In dual-chamber furnaces the pyrolysis and gasification zones are thus more separated spatially from the oxidation zone than in other combustion concepts. The possibility to install turbulence zones in the flue pipe can additionally improve the mixing of the combustion air and thus the burnout. However, an insufficient thermally insulated and not water-cooled furnace can lead to increased radiation losses.

Pre-furnaces can be connected to an existing boiler, as a cost-effective solution for the conversion of an existing fossil fuel (mazut, light oil or natural gas) boiler. The pre-furnace is either installed in front of or, occasionally, under the existing boiler. Better results are obtained if the pre-furnace and the boiler closely match together. Compared with other types of heating plant the space requirement is relatively high. Other disadvantages are the partly insufficient heat removal from the first conversion step as well as slagging and sometimes high NO_x emissions.

The basic principle of a wood chip pre-furnace is shown in Figure 3.4.

Figure 3.4 Schematic diagram of a dual-chamber furnace



Beschickungszyklon	Fuel hopper
Zellenradschleusen	sluice
Zwischenbehälter	intermediate vessel
Dosierschnecke	metering screw
Brennstoff-Förderschnecke	fuel feed
Entgasungsraum	degasification
Schubrostantrieb	grate drive
Primärluft	primary air
Schubrost	grate
automatische Entaschung (Brenner)	automatic de-ashing (burner)
Sekundärluft (vorgewärmt)	secondary air (pre-heated)
Anschlussflansch	flange
Nachbrennraum	combustion chamber
Vorlauf	advance
Sicherheitsvorlauf	safety advance
zum Staubabscheider und Schornstein	to dust remover and chimney
automatische Entaschung (Kessel)	automatic de-ashing (boiler)
Unterdrucksensor	negative pressure sensor
Rücklauf	return
Ascheaustragung	ash removal

Travelling grate

Travelling grate furnaces are built of grate bars forming an endless band (like a moving staircase) moving through the combustion chamber (see Figure 3.5). Fuel is supplied at one end of the combustion chamber onto the grate, by e.g. screw conveyors, or is distributed over the grate by spreader-stokers injecting the fuel into the combustion chamber (see Figure 3.6). The fuel bed itself does not move, but is transported through the combustion chamber by the grate, contrary to moving grate furnaces where the fuel bed is moved over the grate. At the end of the combustion chamber the grate is cleaned of ash and dirt while the band turns around (automatic ash removal). On the way back, the grate bars are cooled by primary air in order to avoid overheating and to minimise wear-out. The speed of the travelling grate is continuously adjustable in order to achieve complete charcoal burnout.

The advantages of travelling grate systems are uniform combustion conditions for wood chips and pellets and low dust emissions, due to the stable and almost unmoving bed of embers. Also the maintenance or replacement of grate bars is easy to handle.

In comparison to moving grate furnaces, however, the fact that the bed of embers is not stoked results in a longer burnout time. Higher primary air input is needed for complete combustion (which implies a lower NO_x reduction potential by primary measures). Moreover, non-homogeneous biomass fuels imply the danger of bridging and uneven distribution among the grate surface because no mixing occurs. This disadvantage can be avoided by spreader-stokers because they cause a mixing of the fuel bed by the fuel-feeding mechanism applied.

Figure 3.5 Operation of a travelling grate furnace

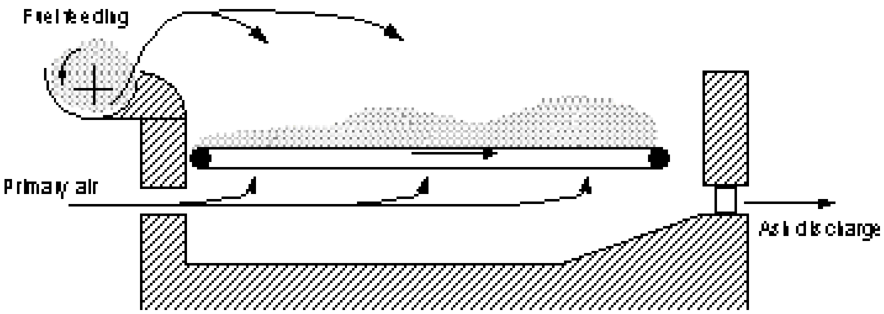
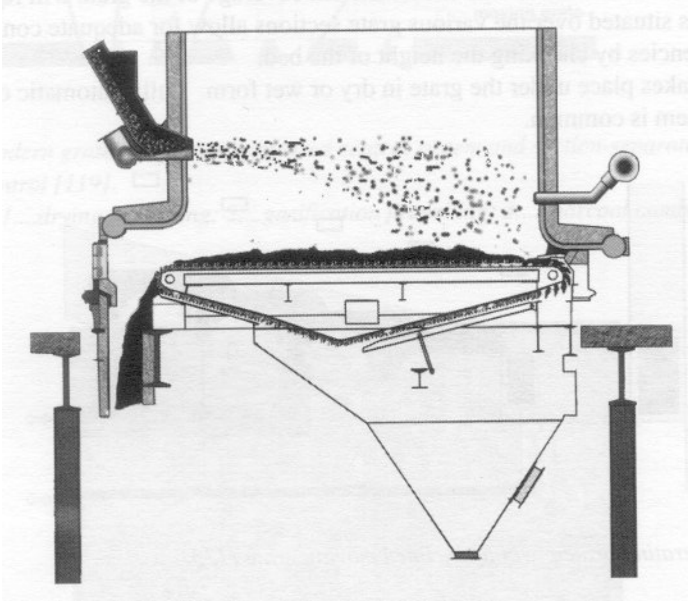


Figure 3.6 Diagram of a travelling grate furnace fed by spreader stokers



Fixed grate systems

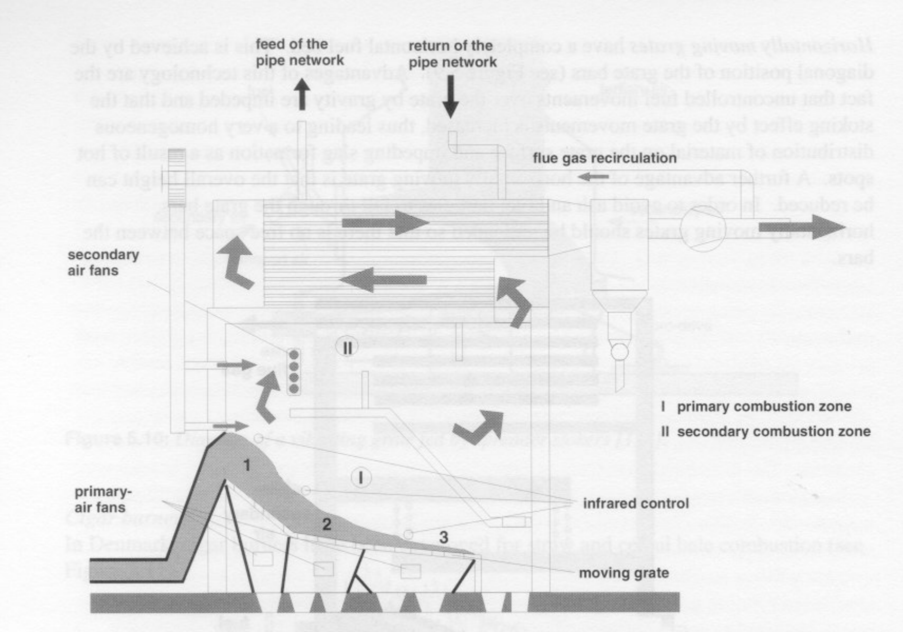
Fixed grate systems (see Figure 3.7) are only used in small-scale applications. In these systems, fuel transport is managed by fuel feeding and gravity (caused by the inclination of the grate). As fuel transport and fuel distribution among the grate cannot be controlled well this technology is no longer applied in modern combustion plants.

Figure 3.7 Picture of an inclined fixed grate

Inclined moving grates and horizontally moving grates

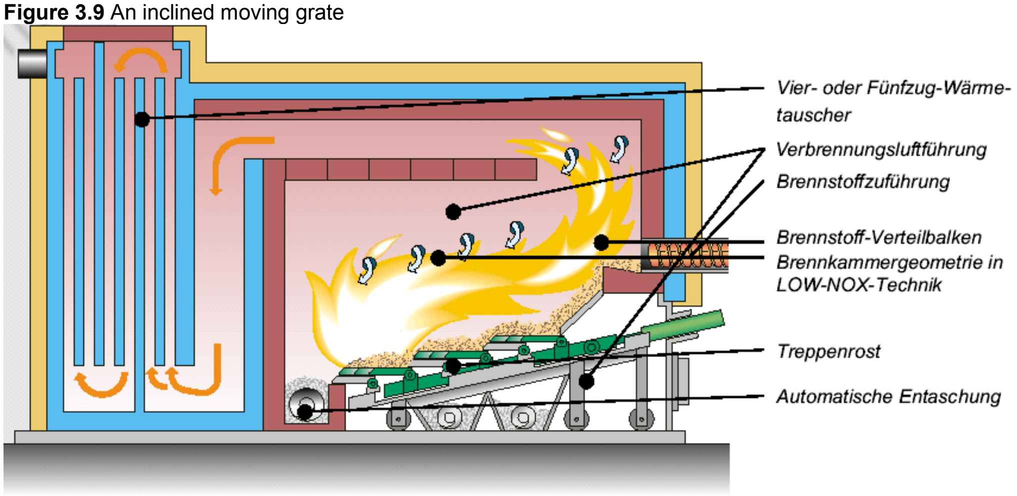
Moving grate furnaces usually have an inclined grate consisting of fixed and moveable rows of grate bars (see Figure 3.8 and Figure 3.9). By alternating horizontal forward and backward movements of the moveable sections, the fuel is transported along the grate. Thus unburned and burned fuel particles are mixed, the surfaces of the fuel bed are renewed, and a more even distribution of the fuel over the grate surface can be achieved (which is important for an equal primary air distribution across the fuel bed). Usually, the whole grate is divided into several grate sections, which can be moved at various speeds according to the different stages of combustion (see Figure 3.9). The movement of the grate bar is achieved by hydraulic cylinders. The grate bars themselves are made of heat-resistant steel alloys. They are equipped with small channels in their side-walls for primary air supply and should be as narrow as possible in order to distribute the primary air across the fuel bed as well as possible.

Figure 3.8 Modern grate furnace with infrared control system and section-separated grate and primary air control



Explanations: 1.... drying prevailing; 2.... gasification prevailing; 3.... charcoal combustion prevailing.

Figure 3.9 An inclined moving grate



Explanation: Wärmetauscher = heat exchanger, Verbrennungsluftführung = combustion air supply, Brennstoffzuführung = fuel supply, Brennstoff-Verteilbalken = fuel distribution beams, Treppenrost = moving grate, Automatische Entaschung = automatic ash removal

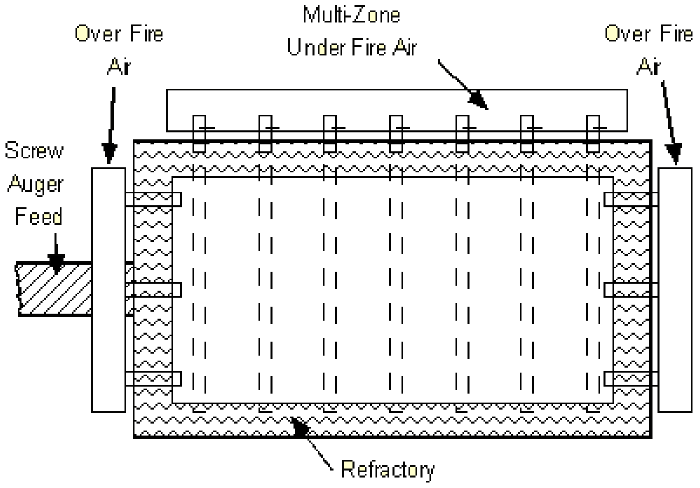
In moving grate furnaces a wide variety of biofuels can be burned. Air-cooled moving grate furnaces use primary air for cooling the grate and are suitable for wet bark, sawdust, and wood chips. For dry biofuels or biofuels with low ash-sintering temperatures, water-cooled moving grate systems are recommended. In contrast to travelling grate systems, the correct adjustment of the moving frequency of the grate bars is more complex. If the moving frequencies are too high, high concentrations of unburned carbon in the ash or insufficient coverage of the grate will result. Infrared beams situated over the various grate sections allow for adequate control of the moving frequencies by checking the height of the bed. Ash removal takes place under the grate in dry or wet form. Fully automatic operation of the whole system is common.

Figure 3.10 Picture of an inclined moving grate

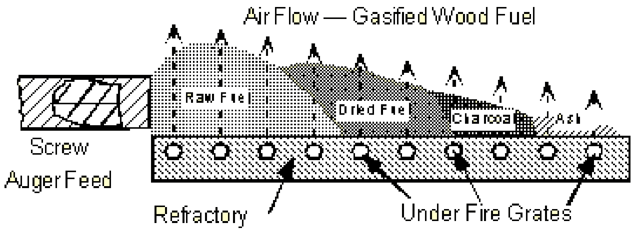
Horizontally moving grates have a completely horizontal fuel bed. This is achieved by the diagonal position of the grate bars (see **Error! Reference source not found.**). Advantages of this technology are the fact that uncontrolled fuel movements over the grate by gravity are impeded and that the stoking effect by the grate movements is increased, thus leading to a very homogeneous distribution of material on the grate surface and impeding slag formation as a result of hot spots. A further advantage of the horizontally moving grate is that the overall height can be reduced. In order to avoid ash and fuel particles to fall through the grate bars, horizontally moving grates should be preloaded so that there is no free space between the bars.

Figure 3.11 Top and side views of the combustion base of a horizontally moving grate furnace

Top View of Combustion Base



Side View of Combustion Base

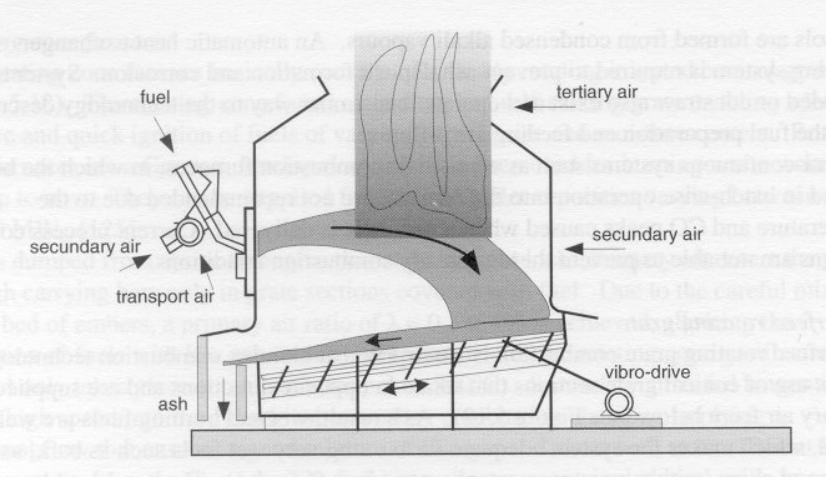


Vibrating grates

Vibrating grate furnaces consist of a declined finned tube wall placed on springs (see Figure 3.12). Fuel is fed into the combustion chamber by spreaders, screw conveyors, or hydraulic feeders. Depending on the combustion process, two or more vibrators transport fuel and ash towards the ash removal. Primary air is fed through the fuel bed from below through holes located in the ribs of the finned tube walls. Due to the vibrating movement of the grate at short periodic intervals (5-10 seconds per every 15-20 minutes), the formation of larger slag particles is inhibited, which is the reason why this grate technology is especially applied with fuels showing sintering and slagging tendencies (e.g. straw, waste wood).

Vibrating grates can achieve very high boiler efficiencies, up to 92%, comparable with those of fluidised bed systems. Operational costs are low due to very small power consumption and very low wear of the grate combustors. Disadvantages of vibrating grates are the high fly-ash emissions caused by the vibrations, the higher CO emissions due to the periodic disturbances of the fuel bed, and an incomplete burnout of the bottom ash because fuel and ash transport are more difficult to control.

Figure 3.12 Diagram of a vibrating grate fed by spreader stokers

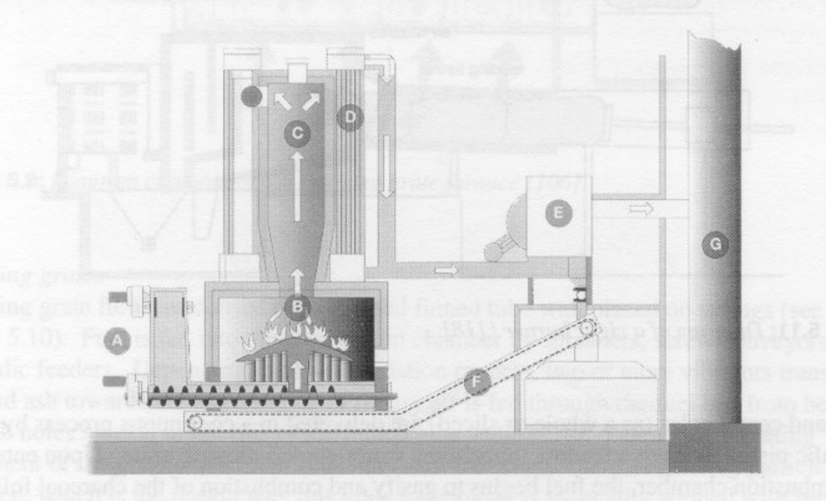


Underfeed rotating grate

Underfeed rotating grate combustion is a new Finnish biomass combustion technology that makes use of conical grate sections that rotate in opposite directions and are supplied with primary air from below (see Figure 3.13 and Figure 3.14). As a result, wet and burning fuels are well mixed, which makes the system adequate for burning very wet fuels such as bark, sawdust, and wood chips (with up to 65%$_{wb}$ moisture content). The combustible gases formed are burned out with secondary air in a separate horizontal or vertical combustion chamber. The horizontal version is suitable for generating hot water or steam in boilers with a nominal capacity between 1 and 10 MW$_{th}$. The vertical version is applied for hot water boilers with a capacity of 1-4 MW$_{th}$. The fuel is fed to the grate from below by screw conveyors (similar to underfeed stokers), which makes it necessary to keep the average particle size below 50 mm.

Underfeed rotating grate combustion plants are also capable of burning mixtures of solid wood fuels and biological sludge. The system is computer-controlled and allows fully automatic operation.

Figure 3.13 Underfeed rotating grate



Explanations:

A ... fuel feed,

B... primary combustion chamber,

C .. secondary combustion chamber,

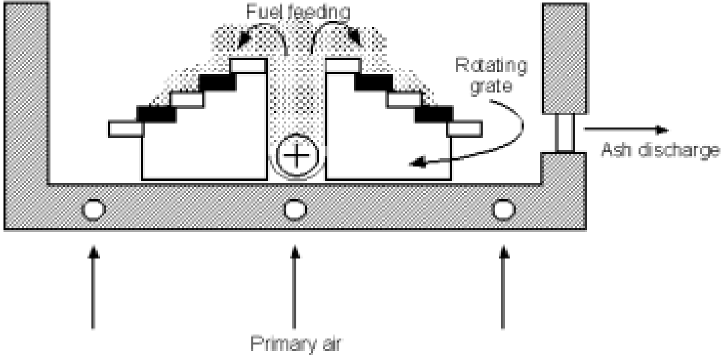
D ...boiler,

E... flue gas cleaner,

F... ash removal,

G...stack.

Figure 3.14 Diagram of an underfeed rotating grate

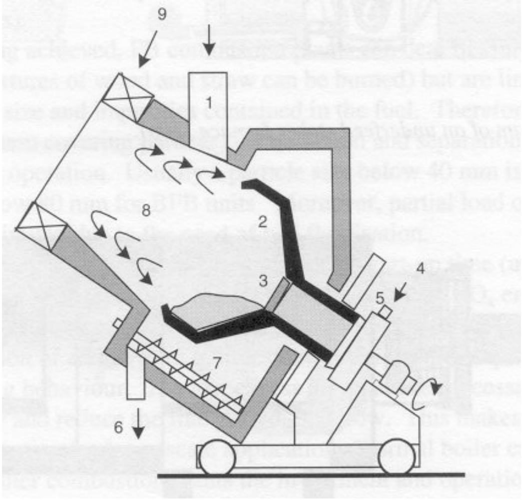


Rotating cone furnace

The rotating cone furnace basically consists of a slowly rotating inverted conical grate (see Figure 5.13). The rotating cone forms an endless and self-stoking grate enabling adequate mixture and quick ignition of fuels of varying particle size and moisture content. Rotating cone furnaces are a German development and have been used for burning waste wood and coal up to now. They can be supplied for nominal boiler capacities varying between 0.4 and 50 MW_{th}.

Fuel is dumped from above through a two-stage airtight lock. Primary air enters the grate through carrying bars only in grate sections covered with fuel. Due to the careful mixing of the bed of embers, a primary air ratio of $\lambda = 0.3$ to 0.6 is achieved, allowing the utilisation of fuels with low ash-melting temperatures (in the rotating cone gasification of the fuel only takes place at temperatures below $800^{o}C$). Secondary air is fed tangentially and at high speed into the cylindrical secondary combustion chamber, implying a rotational flow that ensures a good mixture of flue gas and air as well as an efficient fly-ash separation from the flue gas. The furnace walls are water-cooled steel walls in order to ensure adequate temperature control in the oxidising zone and to avoid ash deposit formations. Thus, the total combustion air ratio can be kept between $\lambda = 1.2$ and 1.4, which is a very low value for fixed-bed furnaces and ensures a high combustion efficiency.

Figure 3.15 Diagram of the rotating cone furnace



Explanations:

1 ... fuel feeding,

2... rotating grate,

3 ... bottom of the cone,

4 ...primary air,

5 ...air control,

6 ... ash disposal,

7…ash screw conveyor,

8 …burn out zone,

9 …secondary air.

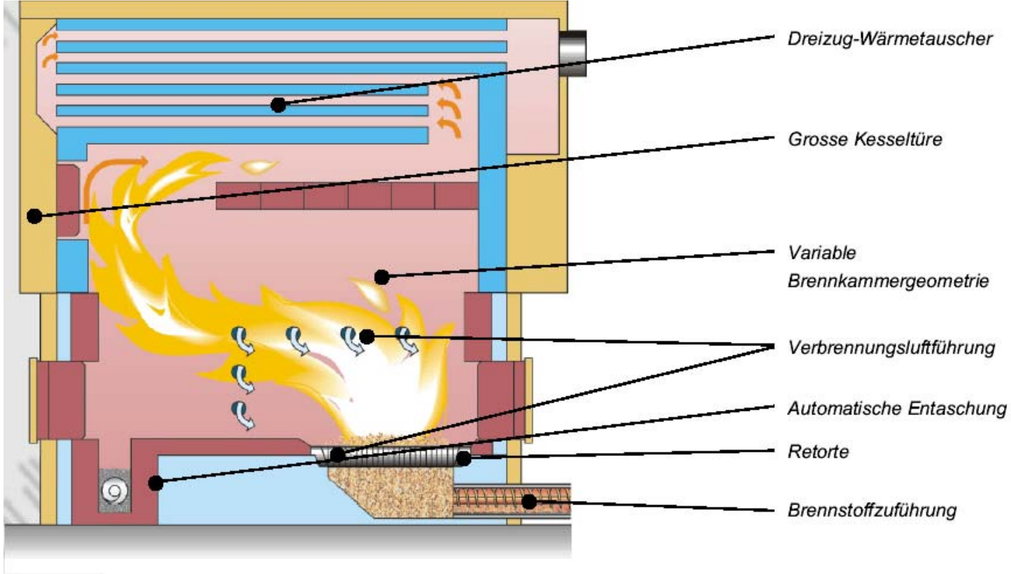
The weak points or disadvantages of this innovative combustion technology are:

- the limited experience with the utilisation of various biofuels at different loads as well as with the wear-out of the grate and the furnace;
- the necessary auxiliary burner needed for start-up due to the water-cooled walls;
- the necessity of shutting down periodically for removing large ash particles that accumulate in the furnace core (this operation is performed automatically by a grappler installed); the frequency depends on the amount of mineral impurities in the fuel.

3.2.2 Underfeed stokers

Underfeed stokers (Figure 3.16, Figure 3.17 and Figure 3.18) are suitable for biomass fuels with low ash content (wood chips, sawdust, pellets) and small particle sizes (particle dimension up to 50 mm). The maximum allowable moisture content is limited at 40% since all moisture moves through the fixed bed and, if too much, strongly influences oxygen concentration. Ash-rich biomass fuels like bark, straw, and cereals need more efficient ash removal systems. Moreover, sintered or melted ash particles covering the upper surface of the fuel bed can cause problems in underfeed stokers due to unstable combustion conditions when the fuel and the air are breaking through the ash-covered surface.

Figure 3.16 Overview of an underfeed stoker furnace



Explanation: Wärmetauscher = heat exchanger, Grosse Kesseltüre = large boiler doors, Verbrennungsluftführung = combustion air supply, Automatische Entaschung = automatic ash removal, Retorte = retort, Brennstoffzuführung = fuel supply.

Relatively high boiler efficiencies of 80-85% can be achieved with underfeed combustors. Both single screw and multiple screw underfeed systems are available. Single screw combustors exist for capacities up to 2 MW_{th}, multiple screw combustors up to 6 MW_{th}. An advantage of underfeed stokers is their good partial-load behaviour and their simple load control. Load changes can be achieved more easily and quickly than in grate combustion plants because the fuel supply can be controlled more easily.

A new Austrian development is an underfeed stoker with a rotational post-combustion, in which a strong vortex flow is achieved by a specially designed secondary air fan equipped with a rotating chain (see Figure 3.19).

Figure 3.17 Diagram of an underfeed stoker furnace

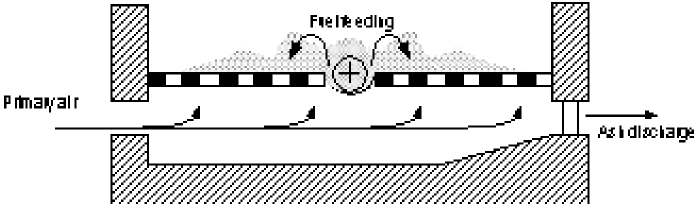
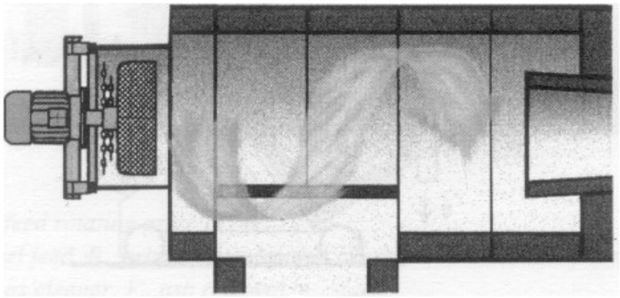


Figure 3.18 Picture of an underfeed stoker

Figure 3.19 Diagram of a post-combustion chamber with imposed vortex flow



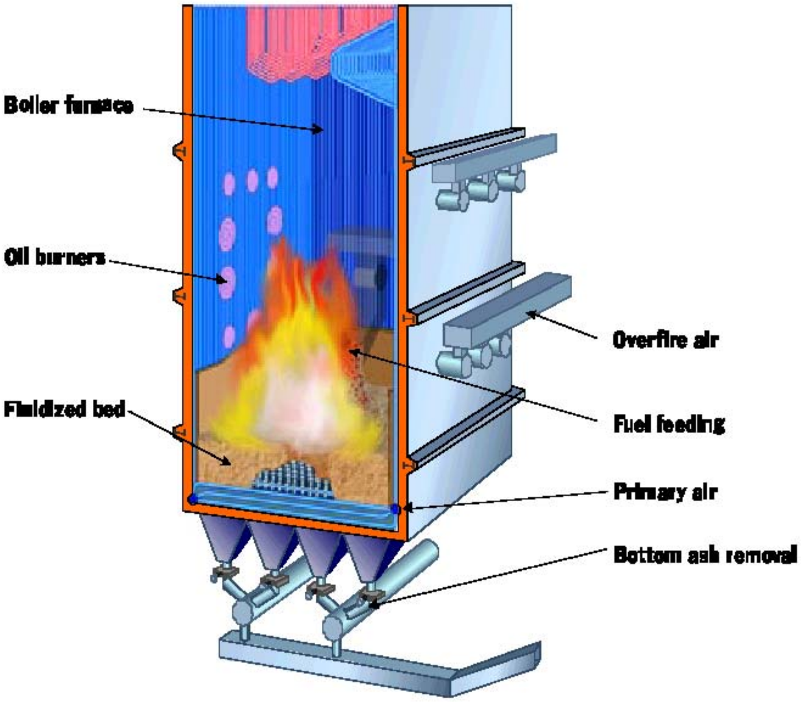
3.3 Fluidised bed combustion

Fluidised bed (FB) combustion systems have been applied since 1960 for combustion of municipal and industrial wastes. Since then, over 300 commercial installations have been built worldwide. Regarding technological applications, bubbling fluidised beds (BFB) and circulating fluidised beds (CFB) have to be distinguished. A fluidised bed consists of a cylindrical vessel with a perforated bottom plate filled with a suspension bed of hot, inert, and granular material. The common bed materials are silica sand and dolomite. The bed material represents 90-98% of the mixture of fuel and bed material. Primary combustion air enters the furnace from below through the air distribution plate and fluidises the bed so that it becomes a seething mass of particles and bubbles. The intense heat transfer and mixing provides good conditions for a complete combustion with low excess air demand (λ between 1.1 and 1.2 for CFB plants and between 1.3 and 1.4 for BFB plants). The combustion temperature has to be kept low (usually between 800-900^{o}C) in order to prevent ash sintering in the bed. This can be achieved by internal heat exchanger surfaces, by flue gas recirculation, or by water injection (in fixed-bed combustion plants combustion temperatures are usually 100-200^{o}C higher than in fluidised bed units).

Due to the good mixing achieved, FB combustion plants can deal flexibly with various fuel mixtures (e.g. mixtures of wood and straw can be burned) but are limited when it comes to fuel particle size and impurities contained in the fuel. Therefore, appropriate fuel pre-treatment system covering particle size reduction and separation of metals is necessary for fail-safe operation. Usually a particle size below 40 mm is recommended for CFB units and below 80 mm for BFB units. Moreover, partial load operation of FB combustion plants is limited due to the need of bed fluidisation.

Fluidised bed combustion systems need a relatively long start-up time (up to 15 hours) for which oil or gas burners are used. With regard to emissions, low NO_x emissions can be achieved owing to good air staging, good mixing, and a low requirement of excess air. Moreover, the utilisation of additives (e.g. limestone addition for sulphur capture) works well due to the good mixing behaviour. The low excess air quantities necessary increase combustion efficiency and reduce the flue gas volume flow. This makes FB combustion plants especially interesting for large-scale applications (normal boiler capacity above 30 MW_{th}). For smaller combustion plants the investment and operation costs are usually too high in comparison to fixed-bed systems. One disadvantage of FB combustion plants is posed by the high dust loads entrained with the flue gas, which make efficient dust precipitators and boiler cleaning systems necessary. Bed material is also lost with the ash, making it necessary to periodically add new material to the plant.

Figure 3.20 A BFB boiler of Kvaerner design



3.3.1 Bubbling fluidised bed combustion (BFB)

For plants with a nominal boiler capacity of over 20 MW_{th}, BFB furnaces start to be of interest. In BFB furnaces (see Figure 3.20), a bed material is located in the bottom part of the furnace. The primary air is supplied over a nozzle distributor plate and fluidises the bed. The bed material is usually silica sand of about 1.0 mm in diameter; the fluidisation velocity of the air varies between 1.0 and 2.5 m/s. The secondary air is introduced through several inlets in the form of groups of horizontally arranged nozzles at the beginning of the upper part of the furnace (called freeboard) to ensure a staged-air supply to reduce NO_x emissions. In contrast to coal-fired BFB furnaces, the biomass fuel should not be fed onto, but into, the bed by inclined chutes from fuel hoppers because of the higher reactivity of biomass in comparison to coal. The fuel amounts only to 1 to 2% of the bed material and the bed has to be heated (internally or externally) before the fuel is introduced. The advantage of BFB furnaces is their flexibility concerning particle size and moisture content of the biomass fuels. Furthermore, it is also possible to use mixtures of different kinds of biomass or to co-fire them with other fuels. One big disadvantage of BFB furnaces, the difficulties they have at partial load operation, is solved in modern furnaces by splitting or staging the bed.

3.3.2 Circulating fluidised bed (CFB) combustion

By increasing the fluidising velocity to 5 to 10 m/s and using smaller sand particles (0.2 to 0.4 mm in diameter) a CFB system is achieved. The sand particles will be carried with the flue gas, separated in a hot cyclone or a U-beam separator, and fed back into the combustion chamber (see Figure 3.21).

Figure 3.21 CFB process shown schematically

The bed temperature (800 to 900 ^{0}C) is controlled by external heat exchangers cooling the recycled sand, or by water-cooled walls. The higher turbulence in CFB furnaces leads to a better heat transfer and a very homogeneous temperature distribution in the bed. This is of advantage for stable combustion conditions, the control of air staging, and the placement of heating surfaces right in the upper part of the furnace. The disadvantages of CFB furnaces are their larger size and therefore higher price, the even greater dust load in the flue gas leaving the sand particle separator than in BFB systems, the higher loss of bed material in the ash, and the small fuel particle size required (between 0.1 and 40 mm in diameter), which often causes higher investments in fuel pre-treatment. Moreover, their operation at partial load is problematic. In view of their high specific heat transfer capacity, CFB furnaces start to be of interest for plants of more than 30 MW_{th}, due to their higher combustion efficiency and the lower flue gas flow produced (boiler and flue gas cleaning units can be designed smaller).

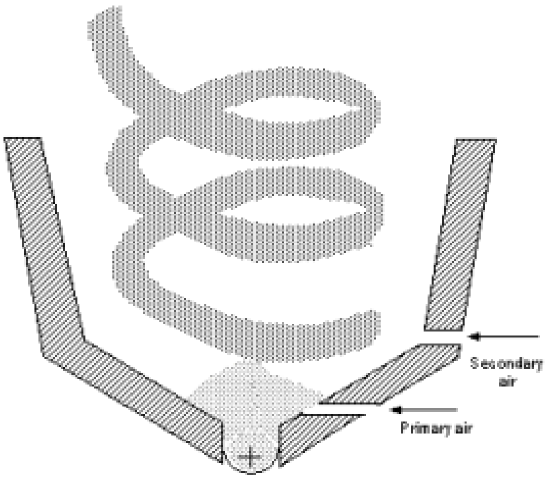
3.4 Dust combustion

In dust combustion systems most of the combustion takes place while the fuel is in suspension. The transportation air is used as primary air. Start-up of the furnace is achieved by an auxiliary burner. When the combustion temperature reaches a certain value, biomass injection starts and the auxiliary burner is shut down.

Fuel/air mixtures are usually injected tangentially into a cylindrical furnace to establish a rotational flow (usually a vortex flow). The rotational motion can be supported by flue gas recirculation in the combustion chamber. The tangential air supply leads to rotation and good mixing of air and fuel. The fuel is fed either mechanically or pneumatically. Due to the high energy density at the furnace walls and the high combustion temperature, the furnace should be water-cooled. Fuel gasification and charcoal combustion take place at the same time because of the small particle size. Therefore, quick load changes and an efficient load control can be achieved. Due to the explosion-like gasification of the fine and small biomass particles, the fuel feeding needs to be controlled very carefully and forms a key technological unit within the overall system.

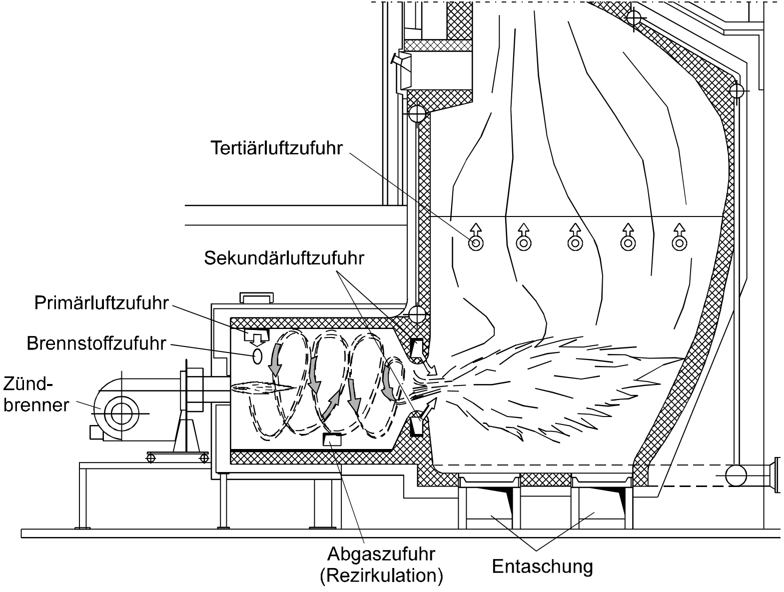
Cyclone burners (Figure 3.22) are commonly used dust combustors, suitable for fuels with a dust content of at least 50%, a particle size of 10-30 mm and a moisture content of up to 10%.

Figure 3.22 Cyclone suspension burner



Muffle dust furnaces (Figure 3.23) are being used increasingly for fine wood wastes originating from the chipboard industry. The outlet of the muffle forms a neck, where secondary air is added in order to achieve a good mixture with the combustible gases. Due to the high flue gas velocities, the ash is carried with the flue gas and is partly precipitated in the post-combustion chamber. Low excess air amounts (λ = 1.3-1.5) and low NO_x emissions can be achieved by proper air staging. This technology is available for thermal capacity between 2 and 8 MW. A maximum fuel particle size of 10- 20 mm has to be maintained and the fuel moisture content should normally not exceed 20 %

Figure 3.23 Diagram of a dust combustion plant (muffle furnace)



Explanations:
Zündbrenner =
Burner for ignition
Brennstoffzufuhr =
Fuel feed,
Primärluftzufuhr =
Primary air,
Sekundärluftzufuhr =
secondary air,
Tertiärluftzufuhr =
tertiary air,
Abgaszufuhr
(Rezirkulation) =
Exhaust gas
(Recirculation),
Entaschung =
removal of ash

3.5 Summary of combustion technologies

Table 3.1 gives a technological evaluation of the biomass combustion technologies discussed. Regarding gaseous and solid emissions, BFB and CFB furnaces normally show lower CO and NO_x emissions due to more homogeneous and therefore more controllable combustion conditions. Fixed-bed furnaces, in turn, usually emit fewer dust particles and show a better burnout of the fly. Table 3.2 summarises typical thermal capacities and required fuel properties for the wood combustion techniques discussed.

Table 3.1 Technological evaluation and fields of application of various biomass combustion technologies

Advantages	**Disadvantages**
Underfeed stokers • low investment costs for plants < 6 MW_{th} • simple and good load control due to continuous fuel feeding • low emissions at partial load operation due to good fuel dosing	• suitable only for biofuels with low ash content and high ash-melting point (wood fuels) • low flexibility in regard to particle size
Grate furnaces • low investment costs for plants < 20 MW_{th} • low operating costs • low dust load in the flue gas • less sensitive to slagging than fluidised bed furnaces	• no mixing of wood fuels and herbaceous fuels possible • efficient NO_x reduction requires special technologies • high excess oxygen (5-8 vol%) decreases efficiency • combustion conditions not as homogeneous as

	in fluidised bed furnaces • low emissions level at partial load operation is difficult to achieve
dual chamber furnaces • can be combined with an existing boiler • cost-effective solution for boiler conversion	• relatively high space requirement • partly insufficient heat removal from the pre-furnace • slagging and sometimes high NOx emissions
dust combustion • low excess oxygen (4-6 vol%) increases efficiency • high NO$_x$ reduction by efficient air staging and mixing possible if cyclone or vortex burners are used • very good load control and fast alternation of load possible	• particle size of biofuel is limited (< 10-20 mm) • high wear out of the insulation brickwork if cyclone or vortex burners are used • an extra start-up burner is necessary
BFB furnaces • no moving parts in the hot combustion chamber • NO$_x$ reduction by air staging works well • high flexibility concerning moisture content and kind of biomass fuels used • low excess oxygen (3-4 Vol%) raises efficiency and decreases flue gas flow	• high investment costs, interesting only for plants > 20 MW$_{th}$ • high operating costs • low flexibility with regard to particle size (< 80 mm) • high dust load in the flue gas • operation at partial load requires special technology • medium sensitivity concerning ash slagging • loss of bed material with the ash • medium erosion of heat exchanger tubes in the fluidised bed
CFB furnaces • no moving parts in the hot combustion chamber • NO$_x$ reduction by air staging works well • high flexibility concerning moisture content and kind of biomass fuels used • homogeneous combustion conditions in the furnace if several fuel injectors are used • high specific heat transfer capacity due to high turbulence • use of additives easy • very low excess oxygen (1-2 vol%) raises efficiency and decreases flue gas flow	• high investment costs, interesting only for plants > 30 MW$_{th}$ • high operating costs • low flexibility with regard to particle size (< 40 mm) • high dust load in the flue gas • partial-load operation requires a second bed • loss of bed material with the ash • high sensitivity concerning ash slagging • loss of bed material with the ash • medium erosion of heat exchanger tubes in the furnace

Table 3.2 Typical capacities and fuel properties for wood combustion techniques

Type	Typical size range	Fuels	Ash	Water content
understoker	20 kW - 2.5 MW	wood chips, wood residues	<2%	5%-50%
dual-chamber	20 kW - 1.5 MW	dry wood (residues)	<5%	5%-35%
moving grate	150 kW - 15 MW	all wood fuels; most biomass	<50%	5%-60%
understoker with rotating grate	2-5 MW	wood chips, high m.c.	<50%	40%-65%
BFB	5-15 MW	various biomass, d< 10mm	<50%	5%-60%
CFB	15-100 MW	various biomass, d< 10mm	<50%	5%-60%
dust combustor, entrained flow	5-10 MW	various biomass, d< 5mm	<5%	<20%

3.6 Heat recovery systems and possibilities for increasing plant efficiency

Table 3.3 gives an overview of the potential of various options for increasing the efficiency of biomass combustion plants. **Biomass drying** is one interesting method, though the efficiency increase and cost savings are usually moderate. Advantages that can be achieved are the prevention of auto-ignitions in wet bark piles, the reduction of dry-matter losses due to microbiological degradation processes during storage, and a reduction of the necessary storage volume at the plant. However, biomass drying processes must be carefully examined for their economic advantage, considering the additional investment costs as well as the operating costs in the form of electricity consumption, and man and machine hours necessary to run the process. In most cases, biomass drying is only economic if pre-heated air is available at low or no cost (examples are solar air collectors and the utilisation of pre-heated air from flue gas condensation units). Drying biomass piles for several months by natural convection is not economic in most cases because the dry-matter losses by biological degradation (1-2 wt% per month) are higher than the increase of efficiency obtained when outdoor storage is considered.

Table 3.3 Influence of various measures on the thermal efficiency of biomass combustion plants

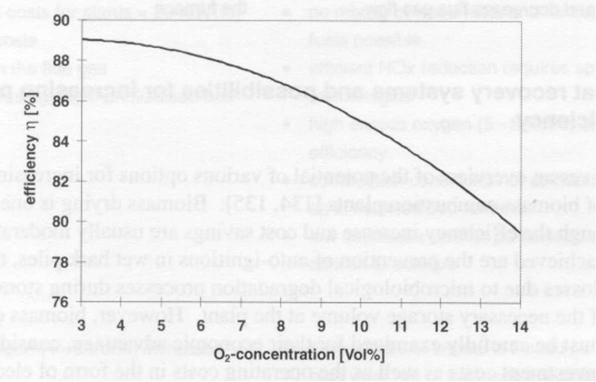
Measures	**Potential thermal efficiency improvement**
Drying from a moisture content of 50 wt% to 30 wt%	+8.7% (w.b.)
Decreasing the O_2-content in the flue gas by 1.0 vol%	about +0.9%
Bark combustion:	
reducing the C_{org} content in the ash from 10.0 to 5.0 wt.% (d.b.)	+0.3%
Decreasing the flue gas temperature at boiler outlet by 10 ^{0}C	+0.8%
Flue gas condensation	average +17%;
(compared to conventional combustion units)	maximum +30%

__Explanations__: Potential thermal efficiency improvement, as compared to the net calorific value of one dry tonne of fuel (%). Calculations performed for wood chips and bark as fuel; GCV = 20 MJ/kg (d.b.).

__Abbreviations__: d.b. ... dry basis; w.b. ... wet basis; NCV .. net calorific value; GCV .. gross calorific value; efficiency = heat output (boiler) / energy input fuel (NCV).

Reducing the excess oxygen content in the flue gas is an effective measure for increasing the efficiency of the combustion plant, as shown in Figure 3.24.

Figure 3.24 Influence of the oxygen content in the flue gas on the plant efficiency.

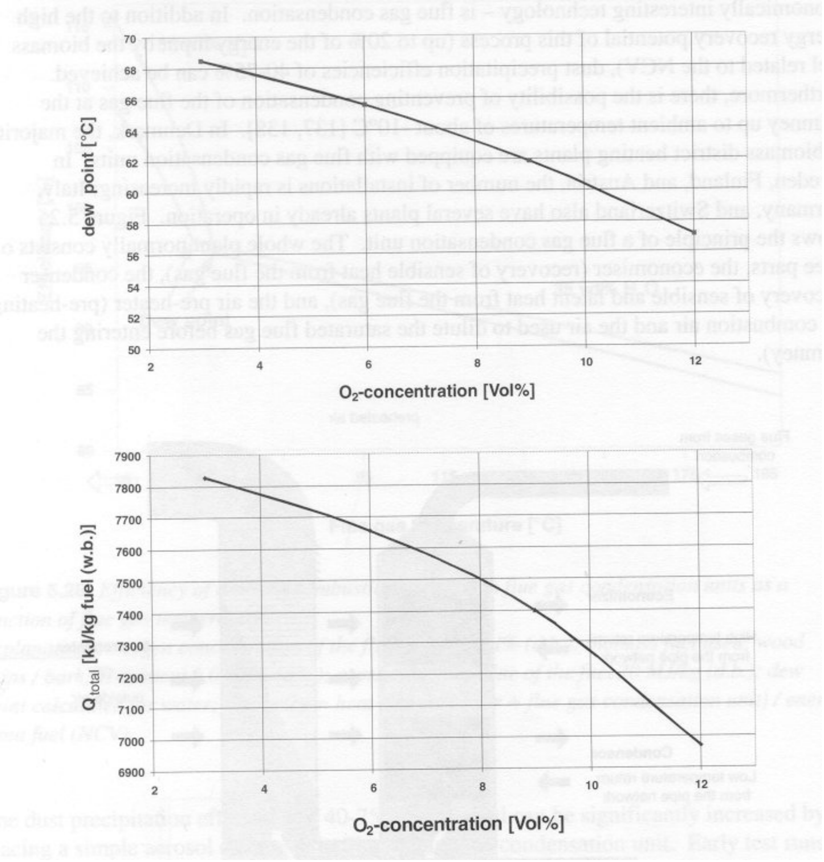


<u>Explanations</u>:

moisture content of the fuel 55 wt%$_{wb}$; biomass fuel used: wood chips / bark; H-content 6.0 wt%$_{db}$; GCV of the fuel 20.3 MJ/kg (d.b.); Flue gas temperature at boiler outlet 165^{0}C, efficiency related to the NCV of the fuel; O_2-concentration related to dry flue gas; efficiency = heat output from boiler / energy input with the fuel (NCV).

There are two technological possibilities for **decreasing the excess air ratio** and to ensure a complete combustion at the same time. On the one hand, an oxygen sensor is coupled with a CO sensor in the flue gas at the boiler outlet to optimise the secondary air supply (CO-λ-control); on the other hand, there are improvements of the mixing quality of flue gas and air in the furnace (as already explained). In addition, a lower excess oxygen concentration in the flue gas can also significantly improve the efficiency of flue gas condensation units, due to the fact that it increases the dew point and therefore raises the amount of recoverable latent heat of the condensing water at a certain temperature (see Figure 3.25).

Figure 3.25 Influence of the oxygen content in the flue gas on the heat recoverable in flue gas condensation plants.



<u>Explanations</u>:

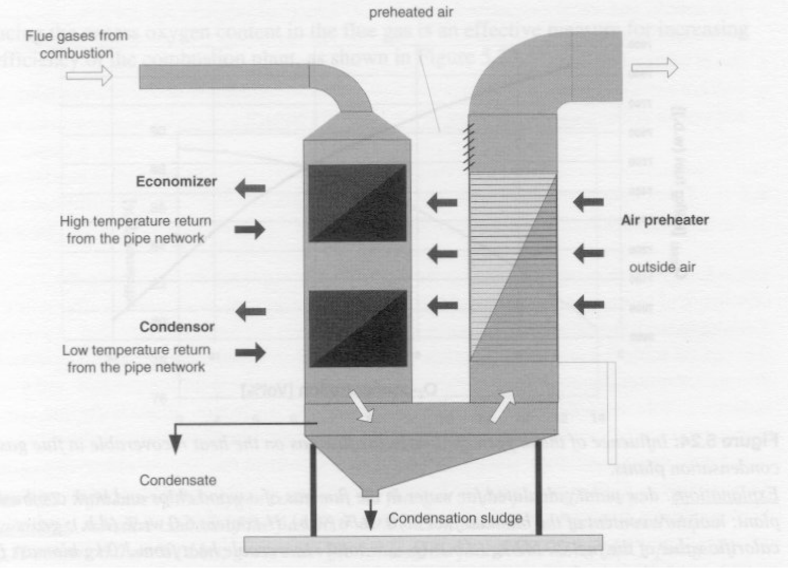
dew point calculated for water in the flue gas of a wood chips and bark combustion plant, moisture content of fuel 55 wt%$_{wb}$, H-content 6.0 wt%$_{db;}$ GCV of the fuel 20 MJ/kg (d.b.); Q_{total} = recoverable heat from 1.0 kg biomass fuel (w.b.) burned when the flue gas is cooled down to 55^{0}C.

Furthermore, reducing the excess oxygen concentration in the flue gas also decreases the flue gas volume flow, which limits pressure drops and reduces the sizes of boilers and flue gas cleaning units. Of course, care has to be taken that a reduction of the excess oxygen concentration in the flue gas also increases the combustion temperature, which underlines the importance of an appropriate furnace temperature control system.

Low carbon-in-ash values are of minor importance for the efficiency of the plant but of major importance for the possibility of ash re-utilisation, because the concentration of organic contaminants in biomass ashes normally increases with higher carbon concentrations.

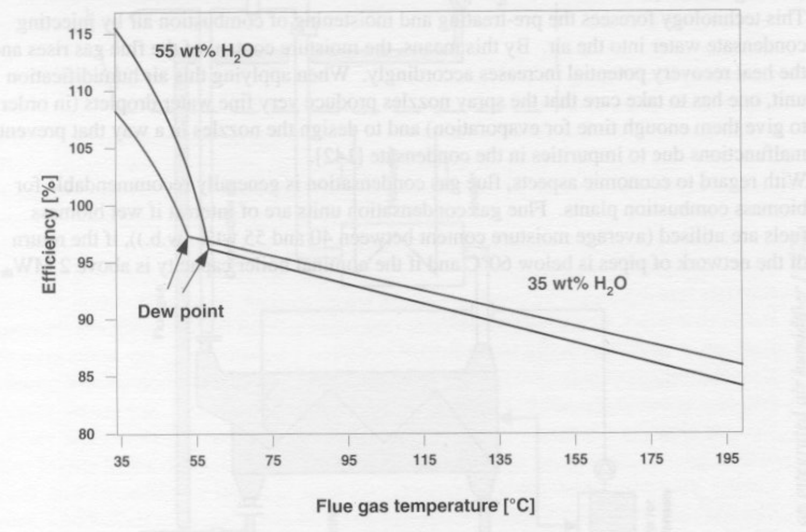
The most effective way to recover energy from the flue gas -and in many cases also an economically interesting technology- is **flue gas condensation**. In addition to the high energy recovery potential of this process (up to 20% of the energy input by the biomass fuel related to the NCV), dust precipitation efficiencies of 40-75% can be achieved. Furthermore, there is the possibility of preventing condensation of the flue gas at the chimney up to ambient temperatures of about -10^{0}C. In Denmark, the majority of biomass district heating plants are equipped with flue gas condensation units. In Sweden, Finland, and Austria, the number of installations is increasing rapidly; Italy, Germany, and Switzerland also have several plants already in operation. Figure 3.26 shows the principle of a flue gas condensation unit. The whole plant normally consists of three parts, the economiser (recovery of sensible heat from the flue gas), the condenser (recovery of sensible and latent heat from the flue gas), and the air pre-heater (pre-heating the combustion air and the air used to dilute the saturated flue gas before entering the chimney).

Figure 3.26 Diagram of a flue gas condensation unit for biomass combustion plants



The amount of energy that can be recovered from the flue gas depends on the moisture content of the biomass fuel, the amount of excess oxygen in the flue gas (as already explained), and the temperature of the return of the network of pipes. The lower the temperature of the return water, the higher the amount of latent heat that can be recovered when the flue gas is cooled below the dew point (see Figure 3.27). Consequently, the energy recovery potential strongly depends on the quality of the heat exchangers, the hydraulic installations and the process control systems installed, because they determine the return temperature.

Figure 3.27 Efficiency of biomass combustion plants with flue gas condensation units as a function of flue gas temperature.



Explanations: oxygen concentration of the flue gas 9.5 vol% (d.b.); biomass fuel used: wood chips / bark; H-content 6.0 wt.% (db.); gross calorific value of the fuel 20 MJ/kg (d.b.); dew point calculated for water, efficiency = heat output (boiler + flue gas condensation unit) / energy input fuel (NCV).

The dust precipitation efficiency of 40-75% mentioned can be significantly increased by placing a simple aerosol electrostatic filter behind the condensation unit. Early test runs showed a dust precipitation efficiency of 99.0% at temperatures below 40^0C. Due to the low flue gas temperature, the electrostatic precipitator (ESP) can also be kept small and therefore economically acceptable. Furthermore, not only aerosols but also water droplets entrained with the flue gas achieve efficient precipitation, lowering the amount of dilution air that has to be added to the saturated flue gas leaving the condensation unit. The condensation sludge has to be separated from the condensate (by sedimentation units) because it contains significant amounts of heavy metals upgraded in this fine fly-ash fraction. It has to be disposed of or industrially utilised. Moreover, research has shown that the separation of sludge and condensate should be done at pH values > 7.5 in order to prevent dissolution of heavy metals and to meet the limiting values for a direct discharge of the condensate into rivers.

Flue gas condensation systems can also be operated as quench processes. The disadvantage of such systems is that quenching slightly lowers the amount of heat that can be recovered and requires even lower temperatures of the return in order to be energetically efficient.

With regard to economic aspects, flue gas condensation is generally recommendable for biomass combustion plants. Flue gas condensation units are of interest if wet biomass fuels are utilised (average m.c. 40-55 $wt\%_{wb}$), if the return of the network of pipes is below 60^0C and if the nominal boiler capacity is above 2 MW_{th}.

3.7 Techno-economic aspects concerning the design of biomass combustion plants

Biomass combustion plants are complex systems with numerous components. In order to ensure a sustainable and economic operation of such plants, professional dimensioning and engineering are essential.

The engineering process consists of several steps:
- Identification of the bases of the biomass combustion plant
- Feasibility study
- Planning of the design
- Approval procedure
- Planning of the execution
- Initialising and placing orders
- Supervision of the construction work
- Commissioning test and documentation

The main components of a biomass combustion plant are (optional components in brackets):
- Fuel storage (long-term storage, daily storage)
- Fuel-feeding and handling system
- Biomass furnace
- Boiler (hot water, steam, thermal oil)
- Back-up or peak-load boiler (e.g. oil-fired boiler)
- Heat recovery system (economiser or flue gas condensation unit)
- Ash manipulation and pre-treatment
- Flue gas cleaning system
- Stack
- Control and visualisation equipment
- Electric and hydraulic installations
- (Heat accumulator)
- (CHP unit)
- (Network of pipes for district heating plants).

Recommended technical and economic standards for biomass combustion and biomass district heating plants

In Austria, technical and economic standards have been defined for biomass district heating plants in order to secure an economically reasonable investment. Keeping these standards is a requirement for new biomass district heating or co-generation projects in Austria; otherwise no investment subsidies are granted.

$$\text{Simultaneity factor [\%]} = \frac{\text{effective peak heat load - district heating network}}{\Sigma\ \text{consumer nominal connection capacity}}$$

$$\text{Boiler full load operating hours [h p.a.]} = \frac{\text{boiler heat produced per year}}{\text{boiler nominal capacity}}$$

Annual utilisation rate - biomass combustion plant [%]	$\dfrac{\text{boiler heat produced per year}}{\text{fuel heat input (NCV) per year}}$
Network heat utilisation rate [kWh/m]	$\dfrac{\text{heat sold per year}}{\text{length of pipe network}}$
Specific investment (boiler) [€/kW]	$\dfrac{\text{investment costs of total system [€]}}{\text{nominal rapacity of biomass boiler}}$
Heat generation costs [€/MWh]	$\dfrac{\text{(capital costs + payments) per year}}{\text{heat sold per year}}$

Plant dimensioning / boiler size

The nominal thermal capacity of a biomass district heating or heat controlled CHP plant is determined by the energy demand (heat, electricity) and has to allow for future developments. Therefore, as a first step, a detailed and precise survey of capacity and heat requirements in the supply area is necessary. Moreover, the simultaneity of heat demand of the district heating clients, described by the simultaneity factor, has to be taken into consideration. This factor depends on the number and type of consumers and fluctuates between 0.5 (large district heating networks) and 1 (micro-networks).

In most cases, the energy demand is not constant over the whole year. The heat load of district heating networks especially varies during the year, reaching a maximum during the winter season and a minimum during summer. Therefore, on the basis of the results of the survey of capacity and heat requirements, the annual heat output line has to be calculated (see Figure 3.28). In boiler planning, a distinction must be made between base load and peak load for economic reasons. Base load is covered by one or more biomass boilers, peak load boilers are usually run on fossil energy for economic reasons. The installations of heat accumulators can also contribute to peak load coverage. This distinction between base load and peak load is necessary to achieve a high number full-load operating hours of the biomass boiler and to decrease total heat generation costs. The correct determination of the boiler sizes depends on the capital costs of the combustion unit as well as on the operating costs (mainly fuel costs - see Table 3.4).

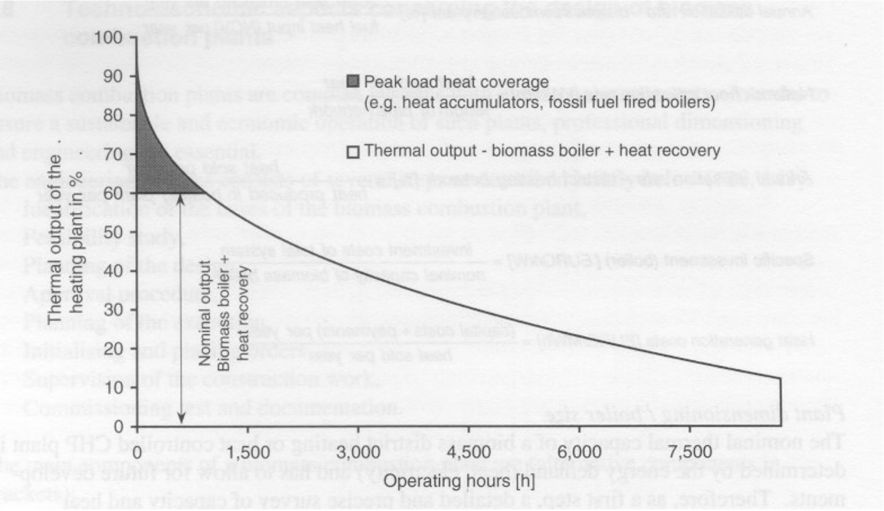
Table 3.4 Comparison of specific investment and fuel costs for biomass and oil-fired combustion systems

Combustion system	Specific investment costs	Fuel costs
Biomass	High (about 100 € / kW) 1)	Low (about 15 € / MWh_{NCV})
Fuel oil	Low (about 20 € / kW) 2)	High (about 30 € / MWh_{NCV})

1) $5MW_{th}$ biomass combustion unit (fuel feeding, furnace, boiler, multicyclone, ESP, precipitator, stack);

2) $5MW_{th}$ fuel oil boiler with burner and stack; specific investment costs related to nominal boiler capacity.

Figure 3.28 Example of distribution between base load and peak load on the basis of annual heat output line.



Annual utilisation rate of the biomass system

The annual utilisation rate of the biomass system (biomass boiler + heat recovery) in the overall plant should be at least 85%. Therefore, the installation of a heat recovery system (e.g. economiser or flue gas condensation unit) is recommended.

Size of the fuel storage unit

The fuel storage unit should be small and should be designed for just-in-time operation (biomass storage capacity unit <10% of annual fuel use). Care should be taken to arrange for appropriate fuel supply contracts, organised fuel purchase, and regional co-ordination.

Construction and civil engineering costs

The costs of the buildings should be less than 750 € per m^2; the costs of the storage unit should be less than 75 € per m^3 of usable volume.

Network of pipes

The costs of the heat distribution network account for 35-55% of total investment costs of complete district heating plants. Thus, it is important to calculate the network correctly in order to achieve high rates of utilisation and to concentrate on a small and efficient network of pipes. For biomass district heating networks, the network heat utilisation rate should exceed 800 kWh per metre; the targeted value is 1,200 kWh per meter. Moreover, a maximum temperature spread between feed and return should be achieved. The targeted value for biomass district heating plants is 40^{0}C or higher. The annual utilisation rate of district heating networks should exceed 75%.

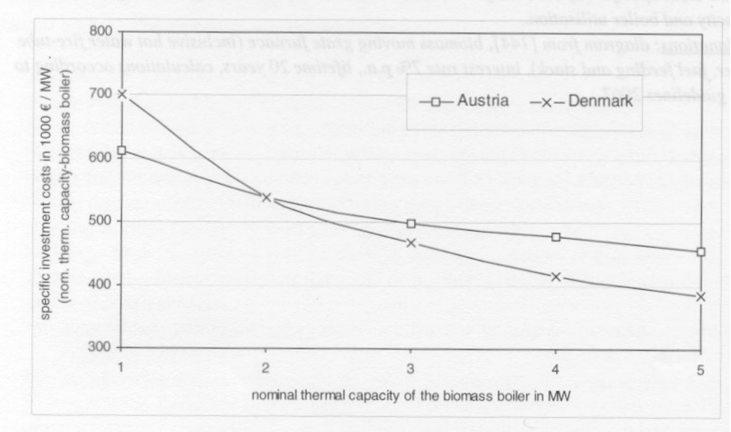
Heat generation costs and economic optimisation

The calculation of heat generation costs is preferably based on VDI Guideline 2067. This cost calculation scheme distinguishes four types of costs:

- capital costs (depreciation, interest costs),
- consumption-based costs (fuel, materials like lubricants),
- operation-based costs (personnel costs, costs for maintenance), and
- other costs (administration, insurance).

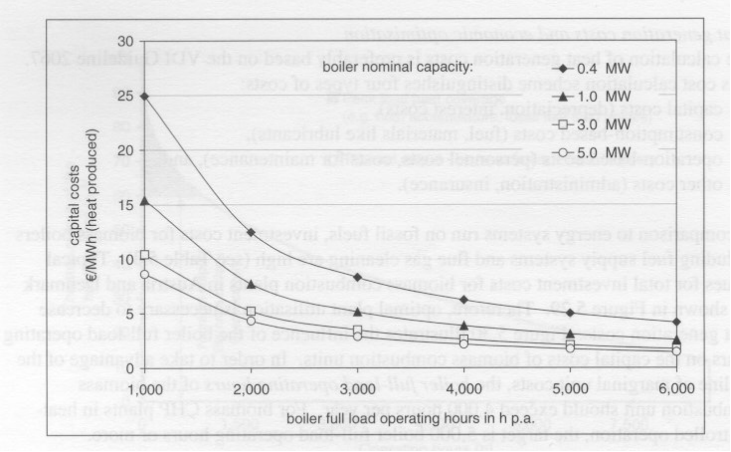
In comparison to energy systems run on fossil fuels, investment costs for biomass boilers including fuel supply systems and flue gas cleaning are high (see Table 3.4). Typical values for total investment costs for biomass combustion plants in Austria and Denmark are shown in Figure 3.29. Therefore, optimal plant utilisation is necessary to decrease heat generation costs. Figure 3.30 illustrates the influence of the boiler full-load operating hours on the capital costs of biomass combustion units. In order to take advantage of the decline of marginal unit costs, the boiler full-load operating hours of the biomass combustion unit should exceed 4,000 hours per year. For biomass CHP plants in heat-controlled operation, the target is 5,000 boiler full-load operating hours or more.

Figure 3.29 Comparison of specific investment costs for biomass combustion plants in Austria and Denmark as a function of biomass boiler size



Explanations: Investment costs include: biomass grate furnace for wood chips, hot water fire-tube boiler, back-up boiler (fuel oil), fuel storage, fuel-feeding system, flue gas cleaning, stack, buildings, hydraulic and electric installations, engineering and construction costs (network of pipes is not included). Price level 1998

Figure 3.30 Specific investment costs for biomass combustion plants as a function of biomass capacity and boiler utilisation



Explanations: biomass moving grate furnace (inclusive hot water fire-tube boiler, fuel feeding and stack), price level 1998, interest rate 7% per year, lifetime 20 years, calculations according to VDI Guideline 2067.

4 POWER GENERATION AND CO-GENERATION

4.1 Introduction

Power generation by combustion can be divided into closed thermal cycles and open processes. *Open cycles* are used for gaseous and liquid fuels to drive internal combustion engines and gas turbines. The fuel is burnt either directly inside an internal combustion engine, or in a combustion chamber and then led through an open gas turbine for expansion. In *closed thermal cycles*, the combustion of the fuel and the power generation cycle are separated by a heat transfer from the hot combustion gas to a process medium used in a secondary cycle. By this separation between fuel and engine, the engine is in contact with a clean process medium. Hence undesired elements in the fuel and flue gas do not cause damage to the engine.

4.1.1 Closed processes

Since biomass fuels and the resulting flue gases can contain elements that may damage engines, such as fly-ash particles, metals, and chlorine components, the technologies for power production through biomass combustion used nowadays are based on closed thermal cycles. The processes and engine types are:

- **Steam turbines** used as expansion engines in the Rankine cycle with water as process medium; the water is evaporated under pressure and superheated;
- **Steam engines** used in the Rankine cycle with or without superheating;
- Steam turbines used in an **organic Rankine cycle (ORC)** with evaporation of an organic medium in a tertiary cycle separated from the heat production (the combustion heat is transferred to a thermal oil in the boiler which is fed to an external evaporator for the organic medium with a lower boiling temperature than water);
- **Stirling engines** (indirectly fired gas engines) which are driven by a periodic heat exchange from flue gas to a gaseous medium such as air, helium or hydrogen;
- **Closed gas turbines** using a hot gas cycle with a turbine as expansion engine in two different configurations:
  a) a completely closed secondary cycle (similar to a Stirling engine) with use of air, helium or hydrogen, and
  b) by a heat transfer to compressed air, which is expanded over a gas turbine and then fed to the boiler as combustion air.

Table 4.1 gives an overview of the working cycles for power generation from biomass. In the steam cycles the high enthalpy difference between liquid phase and gas phase is used in the process. On the opposite side, the process medium does not undertake a phase change in the closed gas cycles using Stirling engines and closed gas turbines. Hence heat exchangers and engines for closed gas cycles have significantly larger dimensions than heat exchangers and engines used in steam cycles. The various technologies cover a wide capacity range, from a few kW (Stirling engines) to several hundred MW (steam turbines used nowadays and closed gas turbines developed earlier). As is shown in Table 4.1, the current state of the various processes varies from concept to proven technology.

Table 4.1 Closed processes for power production by biomass combustion.

Working medium	Engine type	Typical size	Status
Liquid and vapour (with phase change)	Steam turbine	0.5 - 500 MW_e	Proven technology
	Steam piston engine	100 kW_e - 1 MW_e	Proven technology
	Steam screw engine	Not established, similar size as steam piston engine	Development
	Steam turbine with organic medium (ORC)	500 kW_e - 1 MW_e	Some commercial plants with biomass
Gas (without phase change)	Closed gas turbine (hot air turbine)	Not established, similar size as steam turbine	Concept and development
	Stirling engine	20 kW_e – 100 kW_e	Development and pilot

4.1.2 Open processes

Beside the closed processes, the following open processes are also being considered for biomass combustion:

a) directly fired gas turbines by pressurised combustion of biomass with expansion of the flue gas over a gas turbine to atmosphere;

b) directly fired gas turbines by atmospheric combustion of biomass with expansion to vacuum, followed by gas cooling and a compression of the cold gas to enable gas exhaust to atmosphere.

The necessary separation of particles and metals from the hot flue gases is regarded as a relevant disadvantage of these processes. Therefore the directly fired gas turbine cycles for biomass are still in the early stages of development or even in the stage of conception. For the use of open gas turbine cycles, the application of producer gas from biomass gasification is regarded as a promising technology and is being widely investigated, as described in the literature.

4.2 Steam turbines

Power generation by use of steam turbines is a highly developed technology for applications in thermal power stations and in combined heat and power (CHP or co-generation) plants. Heat generated in a combustion process is used to produce high-pressure steam in a boiler (typically 20 to 200 bar) and in cases of steam turbines superheated to increase efficiency and to achieve dry steam. The steam is expanded through the expansion engine and delivers mechanical power to drive an electricity generator.

Steam turbines are often applied in medium- and large-scale power plants ranging from 5 MW_e to more than 500 MW_e, and operated as condensing plants. In the range from 0.5 to 5 MW_e, steam turbines are also operated as back-pressure plants with heat extraction for thermal use and hence reduced electric efficiency.

For small turbines, axial and radial flow type machines exist, while large turbines are built as axial flow type only. Small units are built as single-stage turbines, while large turbines are used as multistage expansion machines (see Figure 4.1, Figure 4.2, and Figure 4.3).

In multistage turbines, high-pressure ratios between inlet and outlet pressure (and hence high efficiencies) are achieved, while the pressure ratio in a single-stage expansion is limited. At the turbine entrance, the steam velocity is typically around 60 m/s. Due to stationary blades, this axial speed is redirected and accelerated into a radial speed of about 300 m/s, while the steam pressure decreases. This kinetic energy of the steam is converted into rotational energy of the rotor. Expanded steam leaves the turbine in an axial direction. As steam passes one turbine blade, the enthalpy difference before and after the blade is relatively low. Depending on the operating conditions, a turbine therefore consists of many blades, varying in length and width.

Figure 4.1 Single-stage radial flow steam turbine with gear shaft and generator used in a biomass-fired CHP plant of approx. 5 MW_{th} and 0.7 MW_e.

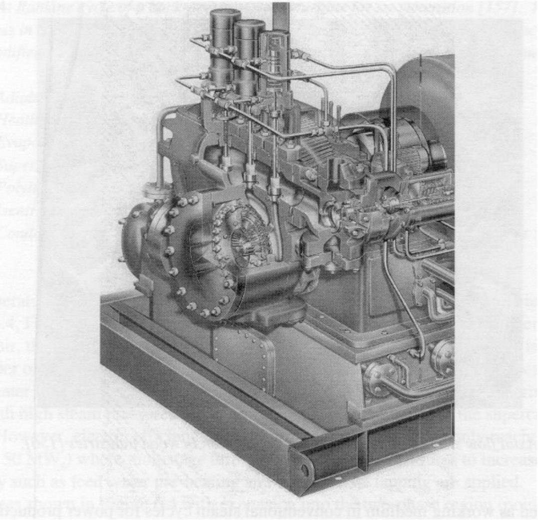


Figure 4.2 Rotor of a two-stage radial flow steam turbine (2.5 MW_e).

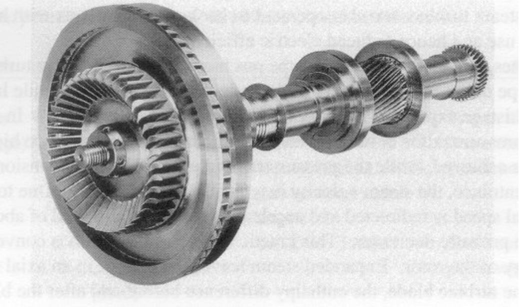
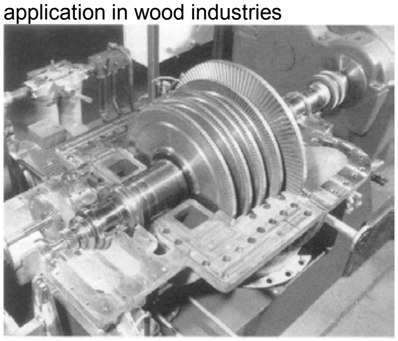


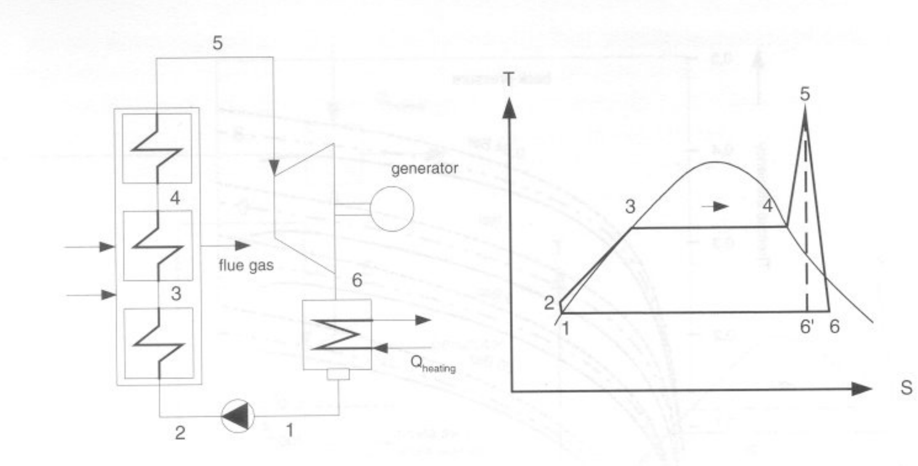
Figure 4.3 Axial flow steam turbine, typical for application in wood industries



Water is used as working medium in conventional steam cycles for power production. Figure 4.4 shows a T/s-diagram (temperature versus entropy) of a power plant with a back-pressure steam turbine for co-generation. The electric efficiency in the Rankine cycle depends on the enthalpy difference before and after the turbine and therefore on the difference between inlet and outlet pressure and temperature: high steam pressures are needed to achieve high efficiencies. On the other hand, high pressure and temperature increase the investment costs, and the risk of corrosion. Especially for biofuels with high chlorine content such as straw, the temperature in the superheater can be limited due to high temperature corrosion. To increase the electric efficiency in such cases, topping with natural gas (a separate gas-fired superheater) can be applied.

The temperature in the condenser should be as low as possible for a high electric yield (Figure 4.4, Figure 4.5). If no heat recovery is applied and the condenser is operated with ambient air, the condensation temperature varies with outside temperature and is typically in the order of approximately 30°C to approximately 0.04 bar. This enables pressure ratios greater than 5000 and electric efficiencies of more than 40% in large thermal power plants with high steam pressures (up to more than 200 bar and also in the supercritical phase). However, electric efficiencies in this order of magnitude are only reached at large plants (>50MW$_e$) where multistage turbines and additional measures to increase the efficiency such as feed water pre-heating and intermediate tapping are applied.

Figure 4.4 Rankine cycle of a back-pressure steam turbine for co-generation. Flow sheet and process in the T/s diagram (temperature versus entropy).



1-2 Adiabatic pressure increase of the water in the water supply pump

2-3 Heating of the water to evaporation temperature in the pre-heater

3-4 Evaporation of the water in the boiler

4-5 Superheating of the vapour in the superheater

5-6 Polytrope expansion of the vapour in the steam turbine (real process)

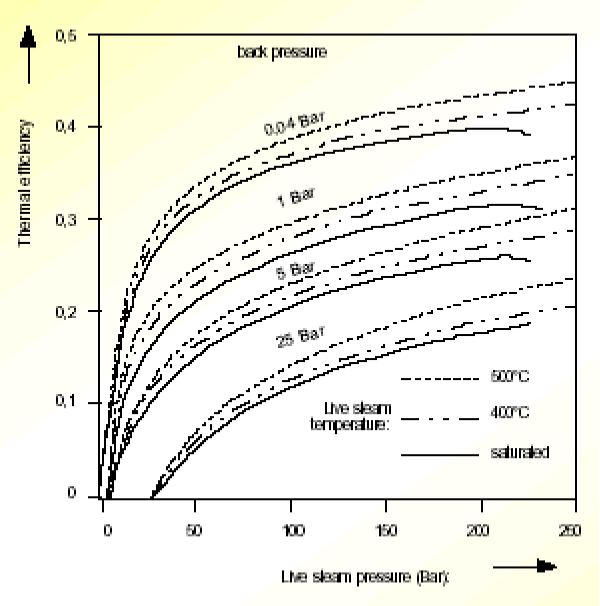
5-6' Isentrope expansion of the vapour (ideal process)

6-1 Condensation of the vapour, utilisation of heat in the condenser, typically at > 1 bar.

To improve the efficiency, the cycle can be modified by feed water pre-heating, intermittent steam re-heaters and further measures.

The process shown in Figure 4.4 with expansion into the two-phase region (point 6) is typical for large steam turbines, which allow a certain concentration of droplets in the turbine (typically 10% to 15% of wetness). Small turbines have to be operated with dry steam (point 6 has to be outside the two-phase region), which limits their efficiency. In co-generation plants, the temperature in the condenser is usually between 90°C to 140°C, with a back-pressure of approximately 1 to 5 bar. This leads to a reduction of the electric efficiency of approximately 10%, since the enthalpy difference is only partly used for power production (see Figure 4.5). However, the overall efficiency indicated as the sum of electric efficiency and heating efficiency can be increased by co-generation, reaching up to 80%.

Figure 4.5 Efficiency of the steam cycle as a function of live steam parameters and back-pressure



Explanation: The efficiency of a power plant can be calculated as a product of thermal efficiency, boiler efficiency, turbine efficiency, and generator efficiency. Hence the plant efficiency is significantly lower than the illustrated thermal efficiency of the Rankine cycle.

To enable heat production allowing for a varying heat demand, steam can be used on an intermediate pressure level for heat production (Figure 4.6). This enables an operation of the plant at maximum overall efficiency in winter with high heat production and at maximum electric efficiency in summer with low heat production. Steam turbines are highly sensitive to scale economies. In smaller plants of up to 1 MW_e, fire tube boilers, which allow steam pressures of only 20-30 bar, are applied instead of water tube boilers for economic reasons. Furthermore, the complexity of these plants is limited (single-stage turbines or turbines with few stages, dry steam, no intermediate tapping, etc.). They are usually operated as back-pressure turbines for combined heat and power production, which avoids vacuum operation and then leads to electric efficiencies of approximately 8%-12%. Efficiencies around 20%-25% are reached at condensing plants of 5-10 MW_e without co-generation.

Since steam turbines have been applied in many situations, research has been done on improved designs and materials. Higher conditions of inlet steam raise the isentropic efficiency, so new materials have been developed that can stand higher inlet temperatures. When wet steam is applied, corrosion can easily occur; therefore materials need to be selected carefully. Research is also being done for relatively small installations (0.25-10 MW_e). Especially in this power range, the efficiencies at partial loads are still low. The efficiency in practical operation is lowered further since the entrance angle of steam on the blades can only be optimised for one single operating situation (steam consumption, pressure, etc.).

Figure 4.6 Condensing plant with use of steam at intermediate pressure for varying demand.

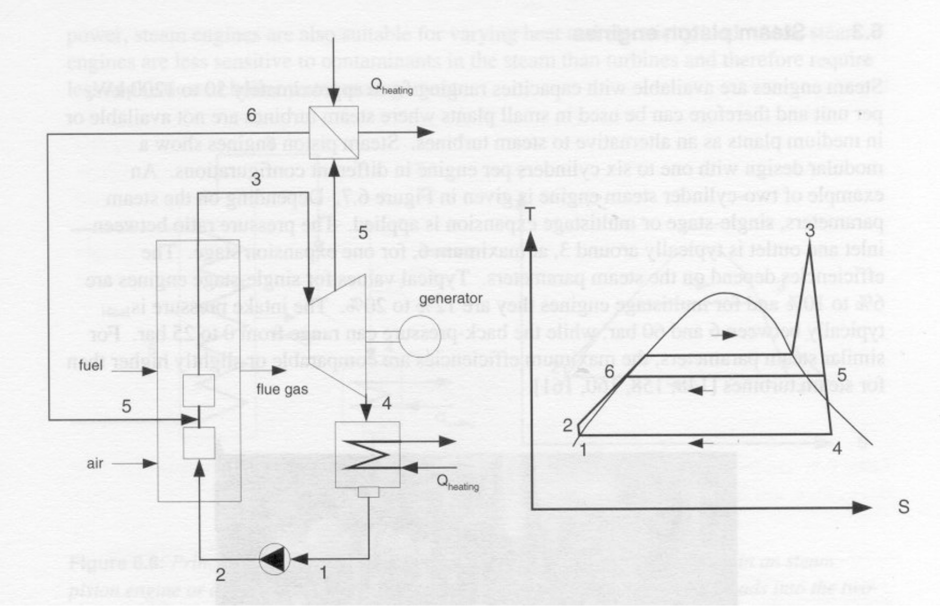


Table 4.2 summarises the main advantages and weak points of steam turbines for use in biomass combustion.

Table 4.2 Technological evaluation of steam turbines for use in biomass combustion

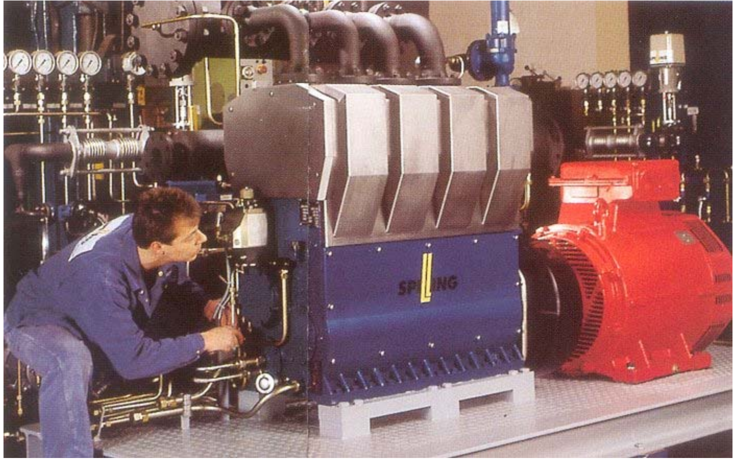
Advantages	Weak points
• Mature, proven technology • Broad power range (no upper limit regarding plant size) • For large installations: high efficiencies can be obtained by high steam temperatures and pressures • Separation between fuel and thermal cycle, enabling the use of fuel containing ash and contaminants • Well applicable for large-scale installations (>2 MW_{el}	• Small steam turbines <1 MW_e offer only limited efficiencies • Low efficiency and special control systems required at partial load • High specific investment costs for small turbines • For biomass application: limited superheater temperature (and thus efficiency) because of risk of high temperature corrosion • High quality steam is necessary • Educated steam boiler operator necessary • High operating costs (maintenance, feed water treatment)

4.3 Steam piston engines

Steam engines are available with capacities ranging from approximately 50 to 1200 kW_e per unit and therefore can be used in small plants where steam turbines are not available or in medium plants as an alternative to steam turbines. Steam piston engines show a modular design with one to six cylinders per engine in different configurations. An example of a four-cylinder steam engine is given in Figure 4.7. Depending on the steam parameters, single-stage or multistage expansion is applied. The pressure ratio between inlet and outlet is typically around 3, at maximum 6, for one expansion stage. The efficiencies depend on the steam parameters. Typical values for single-stage engines are

6% to 10% and for multistage engines they are 12% to 20%. The intake pressure is typically between 6 and 60 bar, while the back-pressure can range from 0 to 25 bar. For similar steam parameters, the maximum efficiencies are comparable or slightly higher than for steam turbines.

Figure 4.7 Example of a steam engine (four cylinders) from Spillingwerk



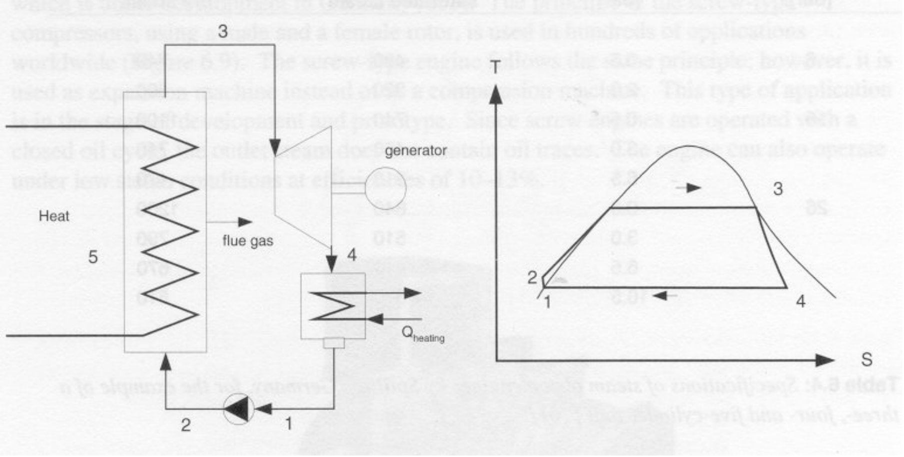
Steam engines have several advantages in comparison to steam turbines. Steam engines are less sensitive to water droplets in the outlet and a wetness of 12% is acceptable, even for small plants. They can even be operated with low-pressure, saturated steam. Although this reduces the efficiency, investment savings on the steam boiler up to 30% are obtainable.

For steam engines operated with superheated steam in co-generation mode, the process cycle is the same as shown in Figure 4.4 for steam turbines. If operated with saturated steam, the process cycle is as shown in Figure 4.8 with expansion from 3 to 4 into the two-phase region to a wetness of up to 12%.

Furthermore, steam piston engines have a higher part load efficiency than turbines. Since they reach up to 90% of the maximum efficiency between 50% and 100% of the nominal power, steam engines are also suitable for varying heat and electricity load. Also, steam engines are less sensitive to contaminants in the steam than turbines and therefore require less sophisticated boiler water management.

An important disadvantage of steam piston engines so far is the need for the injection of oil into the steam for lubrication before it enters the engine. This oil has to be removed from the condensate before it enters the feed water tank by use of a two-stage process with oil separator and active carbon filter. The oil consumption is around 0.2 g/kWh. Since traces of oil can still be found after the oil filter, the steam often cannot be used directly for food processing equipment. Concentrations of oil exceeding 1 mg/l can cause problems in the feed water tank and the boiler. Periodic control of the oil concentration and change of the oil filters increase maintenance work. To avoid this disadvantage, a new technology introduced in 1999 makes oil-free operation of piston engines possible and even allow retrofitting of existing engines. Another disadvantage of steam engines, if operated at speeds between 750 to 1500 rpm, is the production of heavy noise and vibrations.

Figure 4.8 Principle and T/s-Diagram for a steam cycle using saturated steam in a steam piston engine or a steam screw-type engine



Explanation: The expansion from 3 to 4 leads into the two-phase region water/vapour, which is not feasible for steam turbines.

Although saturated and dry steam can be used to operate steam engines, steam consumption and condensation will be less for dry steam (and hence efficiency will be higher), due to the higher enthalpy of dry steam. Table 4.3 indicates the power output for both dry and saturated steam, assuming a constant steam flow of 10 t/h. Table 4.4 presents a technological evaluation of steam engines.

Table 4.3 Output power of a steam engine when using 10 t/h of dry and unsaturated steam

		Engine power [kW]	
Entrance pressure [bar]	Exhaust pressure [bar]	saturated steam	dry steam
6	0.5	480	740
	2.0	320	500
16	0.5	740	1100
	3.0	460	710
	6.5	310	470
26	0.5	840	1200
	3.0	510	790
	6.5	410	670
	10.5	320	510

Table 4.4 Technological evaluation steam piston engines

Advantages	Weak points
• Suitable for lower power ranges	• Maximum power output per steam engine 1.2 MW_e
• Saturated steam can be used	• High maintenance costs
• Very good performance at partial load	• Heavy vibration and noise production
• Steam extraction at various pressures possible due to modularity	• Traces of oil in expanded steam for older engines
• Oil free construction (in newer engines)	

avoids steam contamination

4.4 Screw-type steam engines

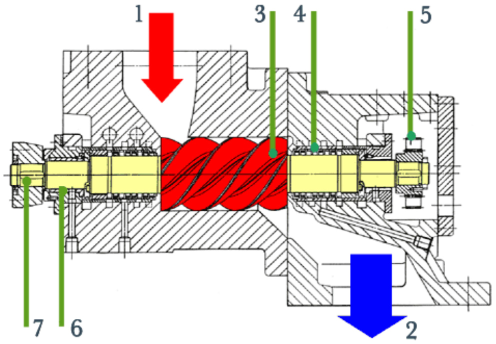
A promising technology for small-scale power generation is a screw-type steam engine, which is under development in Germany. The screw-type engine cycle is based on the conventional Rankine process. In opposite to the steam turbine process the steam is expanded in a screw-type engine, which is connected to a generator producing electric power. This type of application is in the stage of development and prototype.

The screw-type engine is derived from the screw compressor and is consequently based on comprehensive engine know-how. Screw-type engines are suitable for biomass CHP plants in the range of 200 to 2,500 kW_e, where steam parameters can vary, due to variations of the fuel water content and the kind of biomass fuel used, and where a simple and heavy duty design is needed causing low operating and maintenance costs.

Principle of the screw-type engine technology

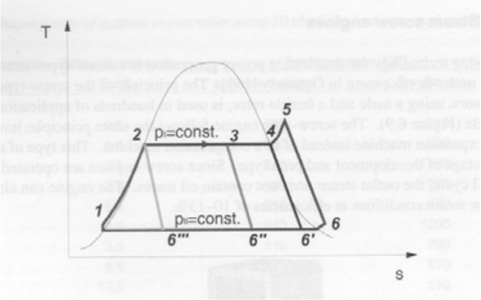
The screw-type engine is a displacement rotary engine. Similar to piston engines, displacement-type engines are characterised by a closed working chamber. The volume of the working chamber changes cyclically, which leads to a decrease of the energy content of the fluid in the chamber. The main parts of a screw-type engine are the male rotor, the female rotor and a casing, which together form a V-shaped working chamber whose volume depends solely on the angle of rotation. The steam enters the casing through the intake port in the passage formed between the tips of the rotor teeth. During rotation the volume of the chamber increases. Intake is finished when the rotor faces pass the guiding edges and the chamber is separated from the intake port. At this stage steam expansion starts and mechanical power is produced at the output shaft. During expansion the volume of the chamber continues to increase, whereas the energy content of the fluid decreases. This process continues until the exhaust process starts and the steam is extruded. It leaves the machine through the exhaust port. How often this process takes place during one rotation of the male rotor depends on the number of teeth on the male rotor. A detailed section drawing of the screw-type engine can be seen in Figure 4.9.

Figure 4.9 Section drawing of a screw-type engine



Explanations: 1...live steam inlet, 2...exhaust steam outlet, 3...male rotor, 4...shaft seal, 5...synchronisation gearwheels, 6...friction type bearing, 7...output shaft

Figure 4.10 Various processes with screw engines in the T/s diagram.



5,6:	process with superheated steam
4,6'	process with saturated steam and expansion into the two-phase region
3,6''	process with wet steam and
2,6'''	process with hot water at boiling

temperature

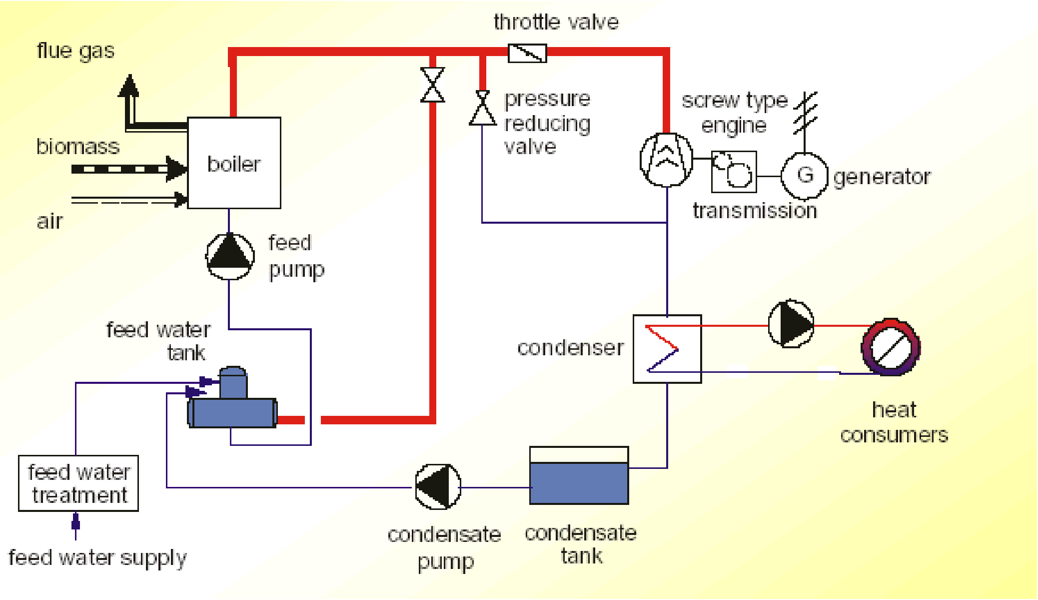
The screw-type engine is a very compact machine with a long lifetime and low maintenance costs. It is insensitive to steam quality fluctuations and can be operated under several different steam conditions. Beside the expansion of superheated steam and saturated steam leading to the cycles described in Figure 4.4 and Figure 4.8, even wet steam or compressed hot water at boiling temperature can be expanded in the two-phase screw-type engine, see Figure 4.10. Water droplets in steam are no problem for screw-type steam machines in contrast to steam turbines and conventional steam engines.

Although the use of steam with low enthalpy leads to limited efficiencies, the screw-type engine offers specific applications that are not suitable for piston engines or steam turbines. The use of hot water screw engines especially offers a potential for small CHP plants, because no steam boiler is necessary. Figure 4.11 shows the flow sheet with the control device for part load operation by use of a pressure-reducing valve and a throttle valve.

Advantages of the screw-type engine process for small-scale biomass CHP applications:

- Comparatively high electric efficiency (10-13%) for small-scale CHP units (< 1,000 kW_e)
- A very good partial-load efficiency over a wide range of load conditions
- Load fluctuations between 30 and 100 % of nominal electric power production are no problem
- Insensitive to steam quality fluctuations.
- The steam cycle and the oil cycle are completely separated by an air-lock system
- The fully automatic operation and easy handling saves staff costs
- The screw-type engine is a very compact machine and causes low maintenance costs

Figure 4.11 Principle of co-generation using a steam engine showing the control of the engine by pressure-reducing valve and throttle valve.

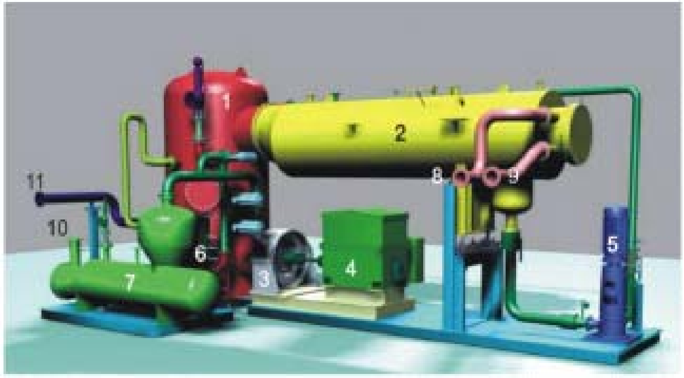


4.5 Organic Rankine Cycle

The principle of electricity generation by means of an organic Rankine cycle (ORC) process corresponds to the conventional Rankine process. The substantial difference is that instead of water an organic working medium with favourable thermodynamic properties is used. This enables operation at relatively low temperatures (70^{0}C to 300 ^{0}C). Many ORC plants have been installed in geothermal power stations with a few ORC generators in operation with industrial waste heat.

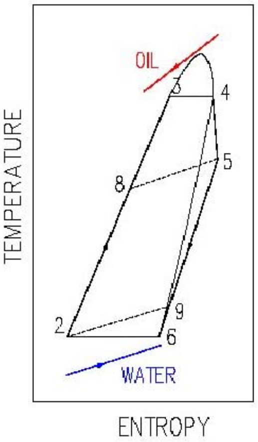
The working principle and the different components of the ORC process are shown in Figure 4.12. The ORC process is connected with the thermal oil boiler via a thermal oil cycle. The ORC unit itself operates as a completely closed process utilising silicon oil as organic working medium. This pressurised organic working medium is vaporised and slightly superheated by the thermal oil in the evaporator and then expanded in an axial turbine which is directly connected to an asynchronous generator (see Figure 4.13). Subsequently, the expanded silicon oil passes through a regenerator (where in-cycle heat recuperation takes place) before it enters the condenser.

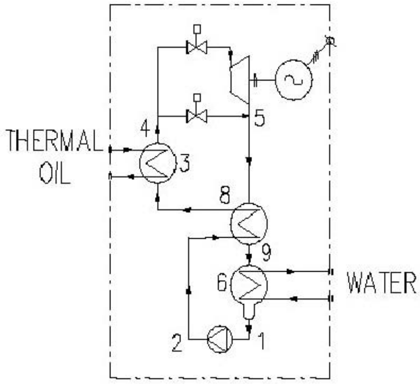
Figure 4.12 Working principle and components of an ORC plant



Explanation:
1 = Regenerator
2 = Condenser
3 = Turbine
4 = Electric generator
5 = Circulation pump
6 = Pre-heater
7 = Evaporator
8 & 9 = Hot water inlet and outlet
10 & 11 = Thermal oil inlet and outlet

Figure 4.13 Principle of co-generation with an ORC process (above) and process in the T/s diagram.



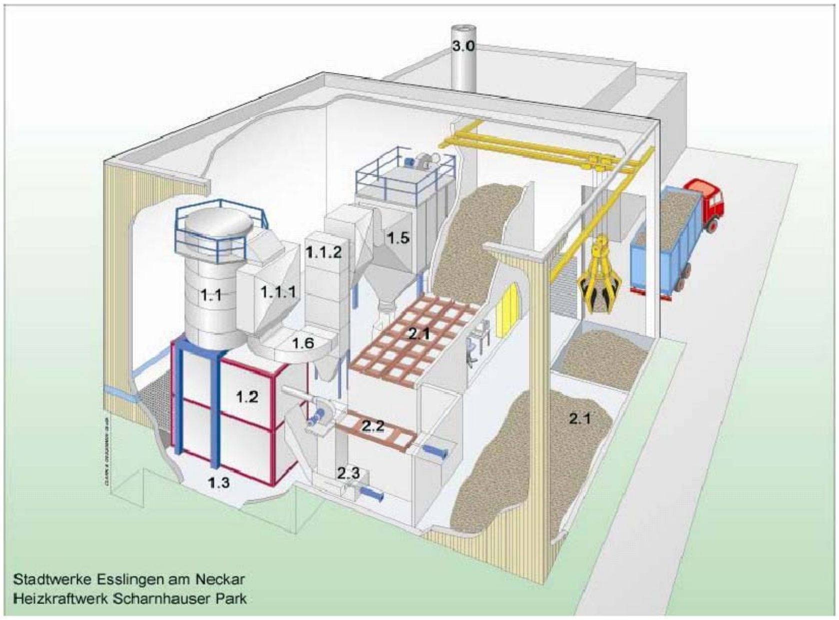


Explanation: In the evaporator (3) the organic working medium (silicone oil) evaporates due to the supply of thermal oil (3-4). Medium steam expands (4-5) in a slow-running two-stage turbine, and after cooling in the pre-heater (or regenerator) (5-9) it finally condenses in the condenser (6). From the condenser the condensation heat is fed into the district heating network.. The ORC cycle closes after the increase in pressure (1-2), pre-heating (2-8) and supply of the organic working medium to the evaporator (8-3).

The condensation of the working medium takes place at a temperature level which allows the heat recovered to be utilised as district or process heat (hot water feed temperature about 80 to 100°C). The liquid working medium then passes the feed pumps to again achieve the appropriate pressure level of the hot end of the cycle. The ORC can be operated at feed-water temperatures of about 80°C the whole year round, although the feed-water temperature required for the district heating network amounts to 90 to 95°C in winter. ORC plants are relatively silent (the highest noise emissions occur at the encapsulated generator and amount to about 85 dB(A) at a distance of 1 m).

No steam boiler is needed and therefore investment costs and maintenance of the boiler are considerably lower than for a comparable steam plant. Another advantage in comparison to conventional steam turbine plants is the possibility of part load operation in the range between 30% and 100% of full load. Efficiencies up to 13% are reached in ORC generators of approximately 300 to 1000 kW_e at thermal oil feed temperatures of 300 ^oC when operated as CHP plants. For optimised processes, up to 17% efficiency is expected. This efficiency, which is slightly higher than that of a steam turbine, is due to the fact that well-developed two-stage turbines are available for this application.

Figure 4.14 Artist impression of the biomass ORC plant in Esslingen, Germany



Explanation: 2.1 through 2.3 = fuel storage, conveyor and hopper. 1.1 = thermal oil boiler. 1.1.1 = economiser (thermal oil and hot water), 1.1.2 = (air pre-heater), 1.2 = boiler, 1.3 = automatic ash removal, 1.4 = multicyclone for flue gas de-dusting (particle removal), 1.5 = flue gas cleaning (electrostatic filter or flue gas condensation), 1.6 = flue gas pipes, 3.0 = chimney

Since the cycle of the ORC process is closed and thus no losses of the working medium are possible, the operating costs are low. Only moderate consumption-based costs (lubricants) and maintenance costs are incurred. The usual lifetime of ORC units is greater than twenty years, as has been proven by geothermal applications. The silicone oil used as working medium has the same lifetime as the ORC since it does not undergo any relevant ageing.

Thermal oil as well as ORC cycles are applied in industries for many years. Biomass CHP plants based on the ORC processes are now entering the market. As of 2004, 11 ORC units were installed in Austria, Switzerland, Italy and Germany with 13 more units (nominal capacities between 200 and 1,500 kW$_e$) under construction in these countries.

Table 4.5 Technological evaluation of the ORC process

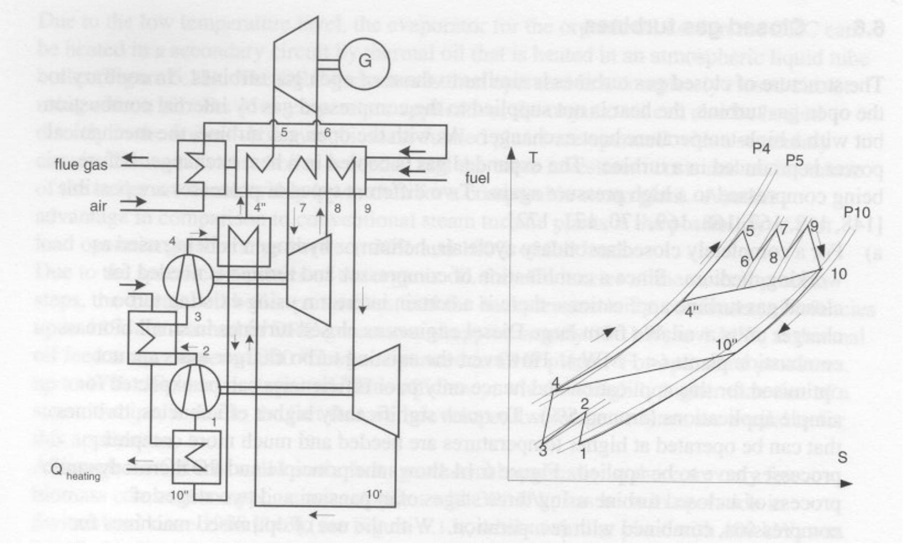
Advantages	Weak points
• Mature and robust technology • Very good controllability and high degree of automation • Very good performance at partial load • Low temp. waste heat can be utilised • No educated steam boiler operator required • Low maintenance requirement and costs	• Relatively high investment costs (no serial production yet) • Long term experience using biomass still missing • Thermal oil cycle necessary (organic thermal oil is inflammable and toxic)

4.6 Closed gas turbines

The structure of closed gas turbines is similar to those of open gas turbines. In contrary to the open gas turbine, the heat is not supplied to the compressed gas by internal combustion but with a high-temperature heat exchanger. As with the open gas turbine, the mechanical power is produced in a turbine. The expanded gas is cooled in a heat exchanger before being compressed to a high pressure again.

Figure 4.15 shows the principle and the thermodynamic process of a closed turbine using three stages of expansion and two stages of compression, combined with recuperation.

Figure 4.15 Diagram and T-s-chart of a closed gas turbine with recuperation



1 - 2	Polytrope compression of air, compressor	6 - 7	Heating of air, furnace /air heater
2 - 3	Intermediate cooling, cooler	7 - 8	Polytrope expansion of air, turbine 2
3 - 4	Polytrope compression of air, compressor	8 - 9	Heating of air, furnace / air heater
4 - 4"	Waste heat recuperation, recuperator	9 - 10	Polytrope expansion of air, turbine 1
4" - 5	Heating of air, furnace / air heater	10 - 10"	Waste heat recuperation, recuperator
5 - 6	Polytrope expansion of air, power turbine	10" - 1	Carrying off of heat for heating, heat exchanger

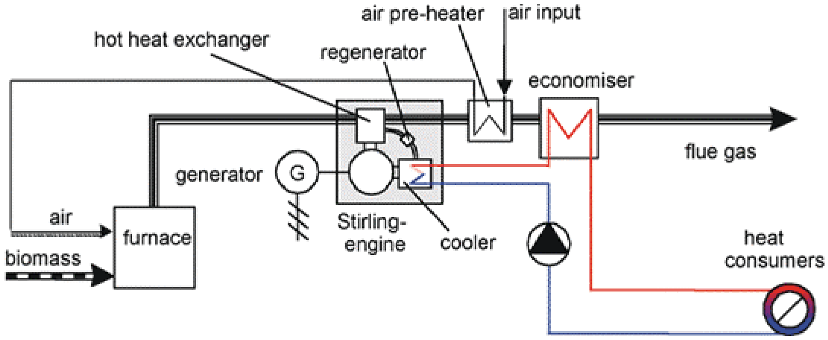
One of the main problems in using closed gas turbines is the high-temperature heat exchanger. This component is heavily burdened by high temperatures of up to 1000^0C, in combination with possible particle and corrosive components in the flue gas. If hot particle removal is applied to render the heat exchanger unnecessary, the complexity of the plant will again increase and the particle removal is another unsolved problem. Furthermore, the heat exchanger has to be very big compared to a gas-water-heat exchanger to support the big volume flux of the hot gases. The closed gas turbine process for the production of electric power from biomass is still in research. A test plant with a performance of 500 kW, has been installed in Belgium. Due to the many uncertainties, an economic application in practice is not expected.

4.7 Stirling engines

The CHP technology based on Stirling engines represents a promising small-scale application for electricity production from solid biomass. At the moment, there is no biomass CHP technology available on the market in the power range below 100 kW_e.

Stirling engines are based on a closed cycle, where the working gas is alternately compressed in a cold cylinder volume and expanded in a hot cylinder volume. The advantage of the Stirling engine over internal combustion engines is that the heat is not supplied to the cycle by combustion of the fuel inside the cylinder, but transferred from the outside through a heat exchanger in the same way as in a steam boiler. Consequently, the combustion system for a Stirling engine can be based on proven furnace technology, thus reducing combustion related problems typical of solid biomass fuels. The heat input from fuel combustion is transferred to the working gas through a hot heat exchanger at a high temperature typically between 680°C and 780°C The heat that is not converted into work on the shaft is rejected to the cooling water in a cold heat exchanger at 25°C - 75°C (see Figure 4.16). Figure 4.17 shows the operating principles of an ideal Stirling engine, while Figure 4.18 demonstrates the mechanical principles of a Stirling engine.

Figure 4.16 CHP biomass combustion plant with Stirling engine



In order to obtain a high overall electric efficiency of the CHP plant, the temperature in the hot heat exchanger should be as high as possible. Therefore, it is necessary to preheat the combustion air with the flue gas leaving the hot heat exchanger by means of an air pre-heater. Typically the temperature of the combustion air is raised to 500-600°C, resulting in very high temperatures in the combustion chamber. This can cause ash slagging and fouling problems in biomass combustion systems and in the hot heat exchanger.

Figure 4.17 Operating principle of the Stirling engine

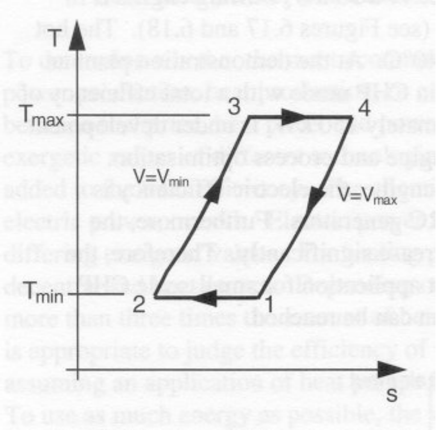
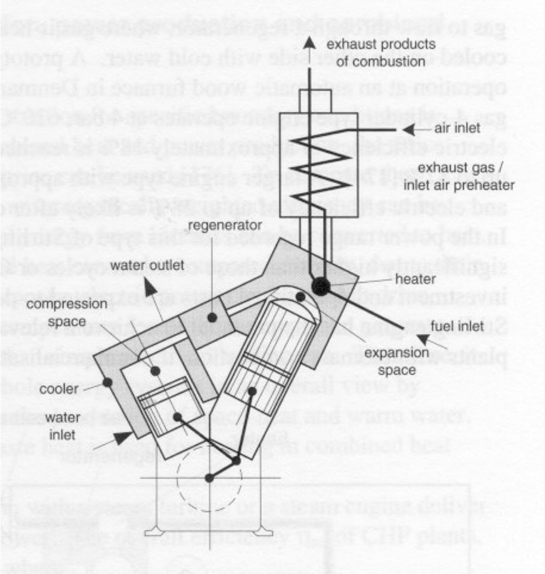


Figure 4.18 Example of a Stirling engine in V-shape



2 - 3: Heating at constant volume: *The gas is heated while the volume remains constant. The pressure increases.*
3 - 4: Expansion: *With constant temperature, the gas is allowed to expand over a piston. The piston drives a crankshaft.*
4 - 1: Cooling: *Gas is allowed to cool, while the volume remains constant. The pressure decreases.*
1 - 2: Compression: *Gas is compressed while the temperature remains constant. The piston drives the crankshaft further.*

The closed Stirling cycle makes it possible to use a working gas, which is better suited for heat transfer to and from the cycle than air. The use of Helium or Hydrogen is most efficient, but utilisation of these low molecular weight gases makes it difficult to design a piston rod seal, which keeps the working gas inside the cylinder and prevents the lubrication oil from entering the cylinder. Many solutions have been tested, but it is still a delicate component in the engine. An attractive possibility is to bypass the problem by designing the engine as a hermetically sealed unit with the generator incorporated in the pressurised crankcase, just like the electric motor in a hermetically sealed compressor for refrigeration. Only static seals are necessary and the only connections from the inside to the outside of the hermetically sealed crankcase are the cable connections between the generator and the grid.

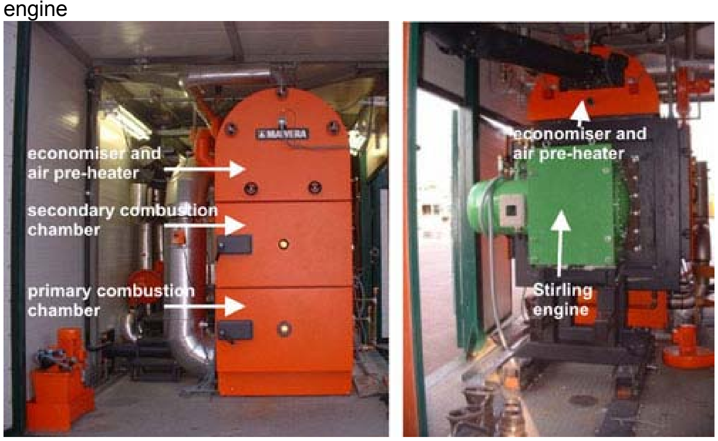
The problems concerning utilisation of biomass fuels in connection with a Stirling engine are concentrated on transferring the heat from the combustion of the fuel into the working gas. The temperature must be high in order to obtain an acceptable specific power output and efficiency, and the heat exchanger must be designed so that problems with fouling are minimised.

Because of the high temperatures in the combustion chamber and the risk of fouling, it is not possible to utilise a Stirling engine designed for natural gas, as narrow passages in the hot heat exchanger are blocked after less than an hour of operation with biomass fuels. The risk of fouling in biomass combustion processes is mainly due to aerosol formation and condensation of ash vapours when the flue gas gets cooled

An advantage of the Stirling engine is its high theoretic efficiency which in actual use is reduced by friction, limited heat transfer and heat recuperation, pressure losses, and other influences. Therefore, the actual efficiency for power production is in the range of 15-30%. In the power range regarded for the Stirling engine, the electric efficiency is significantly higher than those of steam cycles or ORC generators. Furthermore, the investment and operational costs are expected to decrease significantly.

Biomass CHP plants based on a Stirling engine process have reached the demonstration phase. Between 1999 and 2003 the first Austrian, Danish and EU funded demonstration projects were taken in operation. Small series production of seven 35 kW_e units is scheduled by the Austrian firm Mawera for 2004/2005. Figure 4.19 shows the pilot 35 kW_e Stirling engine in operation in Austria since September 2002.

Figure 4.19 Pictures of the CHP pilot plant based on a 35kW_e Stirling engine



Explanation: The furnace of the CHP plants is equipped with underfeed stoker technology. The Stirling engine is mounted in a horizontal position downstream of the secondary combustion chamber for convenient maintenance. The air pre-heater and the economiser are placed on top of the furnace in order to achieve a compact design of the plant.

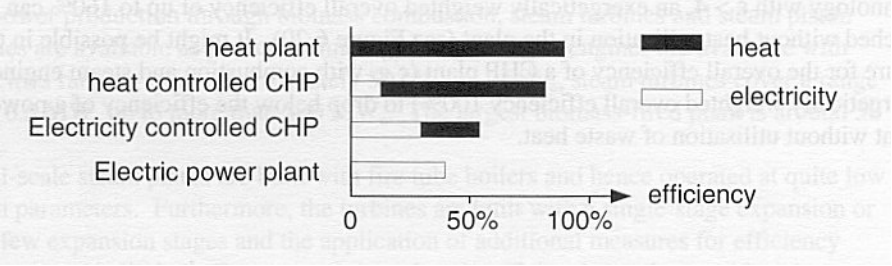
Table 4.6 Technological evaluation ORC process

Advantages	Weak points
• Engine operates independently of heat source	• No reliable solution for sealing problems
• Low quality demand with respect to fuel	• High specific investment costs
• Low maintenance demand because of few moving parts and "external combustion"	• Heat exchanger is exposed to extreme wear because of high temperature steam
• Formation of emissions like CO and CH can be avoided due to external combustion of (bio)gas	• Risk of high temperature corrosion in ash containing flue gases
• Compact design	• Automatic cleaning system for the hot heat exchanger necessary
• Fully automatic operation	• Lack of long-term experience in biomass fired boilers
• Low noise emissions	• Application limited to non-contaminated wood fuels

4.8 Comparison of heat production, power production and CHP production

To determine whether the most economic solution for specific boundary conditions is power production, heat production, or combined heat and power production, the ratios between fuel price, heat price, and power price can be used. To compare the exergetic values of different technologies, an exergetically-weighted efficiency can be added to the total efficiency. The chemical energy from biomass can be converted to heat, electric power, or both. Electric power and heat cannot be compared directly, due to their differing exergetic values. For heating purposes, electric power can be used to drive decentralised heat pumps. Depending on temperature levels, heat pumps can generate more than three times the amount of heat than simple electrical heating can. Therefore, it is appropriate to judge the efficiency of whole energy systems in an overall view by assuming an application of heat pumps for the production of space-heat and warm water. To use as much energy as possible, the waste heat is used for heating in combined heat and power plants. See Figure 4.20.

Figure 4.20 Percentage of heat and electric power production in heating, CHP and power plants



Modern decentralised CHP plants < 1 MW, with a steam turbine or a steam engine deliver about 10% of the fuel energy as electric power. The overall efficiency η_{tot} of CHP plants, including the use of heat, is 80% at most, where $\eta_{tot} = \eta_e + \eta_h$ and e stands for electricity and h for heat. If the electric power is used to run a heat pump with a coefficient of performance ε or COP, an exergetically-weighted overall efficiency of up to 100% can be reached, where η_{ex} is defined as $\eta_{ex} = \varepsilon\,\eta_e + \eta_h$

For an averaged value, the seasonal performance factor has to be considered.

The efficiency of electric power production is approximately 20-30% in typical biomass power plants without heat utilisation, as they exist nowadays. The exergetically-weighted overall efficiency can reach 60%, much less than with a CHP plant, if the electric power is used to operate a heat pump with $\varepsilon = 2.0$. The overall efficiency is therefore higher at a CHP plant than at a power plant without waste heat utilisation.

If the conversion efficiency to power η_e and the coefficient of performance ε are high, power plants and heat pumps can use the exergy more efficiently than CHP plants with heat pumps. This is possible due to the high efficiency of large power plants. CHP plants cannot be built as large as power plants due to the distribution of the heat. Furthermore, the necessary temperature level of the heat limits the CHP plants' conversion efficiency to power. Table 4.7 shows typical data achieved at plants installed nowadays and expected values for future technologies.

It will be possible to reach higher efficiencies in power production from biomass in the future. Targets of up to 45% have been set for IGCC plants. With improved heat pump technology with $\varepsilon > 4$, an energetically weighted overall efficiency of up to 160% can be reached without heat utilisation in the plant (see Figure 4.21).

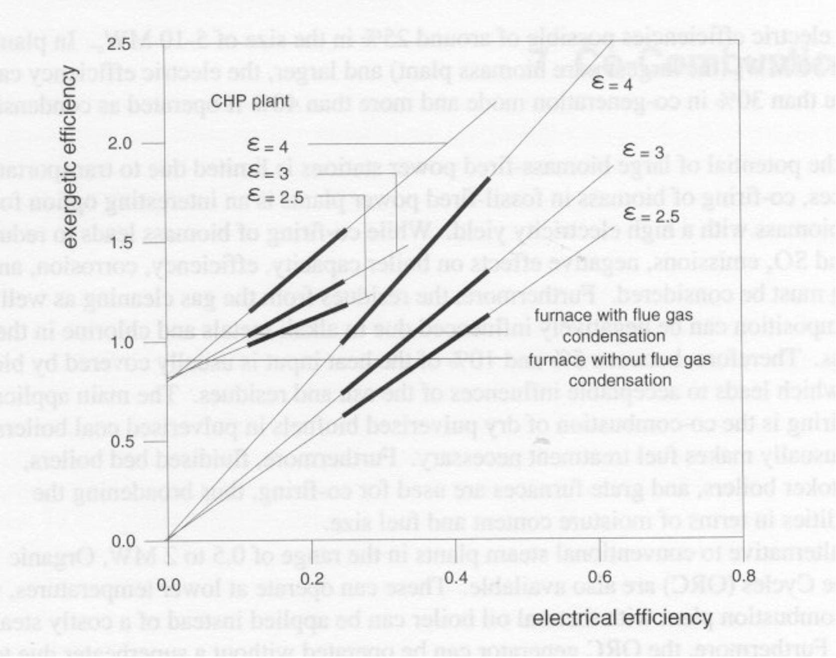
It might be possible in the future for the overall efficiency of a CHP plant (e.g. with combustion and steam engine, energetically weighted overall efficiency 100%) to drop below the efficiency of a power plant without utilisation of waste heat.

Table 4.7 Typical efficiencies for heating, CHP and power plants today and expected values in the future

	Today			Future	
	Heating plant	CHP < 1 MW_e	Power plant >10 MW_e	CHP < 1 MW_e	Power plant >10 MW_e
Technology	Stove,	St. engine, St.	Steam	Gasifier and	IGCC

	boiler	turbine, ORC	turbine	IC-engine, Stirling engine	
η_h	0.85	0.68	0	0.55	0
η_e	0	0.12	0.25	0.25	0.45
ε	2.5	2.5	2.5	4	4
$\eta_{tot} = \eta_e + \eta_h$	0.85	0.8	0.25	0.8	0.45
$\eta_{ex} = \varepsilon\, \eta_e + \eta_h$	0.85	0.98	0.63	1.55	1.8

Figure 4.21 Comparison of heat. CHP and power plant efficiencies by an exergetically weighed efficiency



4.9 Conclusions and summary

For power production through biomass combustion, steam turbines and steam piston engines are available as proven technology. While steam engines are available with capacities ranging from approximately 50 kW_e to I MW_e, steam turbines cover a range from 0.5 MW_e up to more than 500 MW_e. The largest biomass-fired plant is around 50 MW_e.

Small-scale steam plants are built with fire tube boilers and hence operated at quite low steam parameters. Furthermore, the turbines are built with a single-stage expansion or with few expansion stages and the application of additional measures for efficiency improvement is limited. Due to moderate electric efficiencies and to avoid vacuum operation, plants smaller than 1 MW_e are usually operated as back-pressure CHP plants and aim at electric net efficiencies of typically 10-12% and heating efficiencies up to 70%.

Steam plants operated in CHP mode are mainly feasible for the production of process heat, which enables long operation periods. Steam piston engines can be operated in single mode or multistage mode, reaching efficiencies between 6% and 10% and 12% and

20%, respectively. Steam engines can also be operated with saturated steam, thus reducing investment costs by reduced electric efficiency. Large steam turbine plants are operated with water tube boilers and superheaters, thus making high steam parameters and the use of multistage turbines possible. Furthermore, process measures such as feed water pre-heating and intermediate tapping are implemented for efficiency improvement. This makes electric efficiencies possible of around 25% in the size of 5-10 MW_e. In plants around 50 MW_e (the largest pure biomass plant) and larger, the electric efficiency can rise to more than 30% in co-generation mode and more than 40% if operated as condensing plant.

Since the potential of large biomass-fired power stations is limited due to transportation distances, co-firing of biomass in fossil-fired power plants is an interesting option for using biomass with a high electricity yield. While co-firing of biomass leads to reduced NO_x and SO_x emissions, negative effects on boiler capacity, efficiency, corrosion, and fouling must be considered. Furthermore, the residues from the gas cleaning as well as the ash composition can be negatively influenced due to alkali metals and chlorine in the biomass. Therefore, between 5% and 10% of the heat input is usually covered by biomass, which leads to acceptable influences of the ash and residues. The main application of co-firing is the co-combustion of dry pulverised biofuels in pulverised coal boilers, which usually makes fuel treatment necessary. Furthermore, fluidised bed boilers, understoker boilers, and grate furnaces are used for co-firing, thus broadening the possibilities in terms of moisture content and fuel size.

As an alternative to conventional steam plants in the range of 0.5 to 2 MW_e Organic Rankine Cycles (ORC) are also available. These can operate at lower temperatures, so that a combustion plant with thermal oil boiler can be applied instead of a costly steam boiler. Furthermore, the ORC generator can be operated without a superheater due to the fact that the expansion of the saturated steam of the organic medium leads to dry steam. Therefore, the ORC can have advantages in the areas of process design and operation. Furthermore, similar or slightly higher efficiencies are achieved thanks to the availability of well-designed two-stage turbines for this specific application. ORC plants are a well-proven technology for geothermal applications. A few plants are in operation with biomass combustion and therefore further process improvement and cost reduction for this application is expected.

An interesting option for small-scale power production is the externally fired Stirling engine with air or helium coupled to a biomass furnace. A critical component in such plants can be the gas-to-gas heat exchanger operated with hot flue gas from biomass combustion. Native wood is the most suitable fuel so far, since biofuels with high ash levels are a challenge for the heat exchange. Operational experience exists with a 35 kW_e prototype plant, reaching approximately 20% electrical efficiency in CHP operation. Up to 28% efficiency is expected to be reached by process improvement and upscaling to 150 kW_e. The expected investment and operational costs are significantly lower than for steam plants. If these electric efficiencies can be guaranteed in practice without operational problems, and if Stirling engines can be made commercially available, they may be a viable option for small-scale power production by biomass combustion in future.

For medium-scale applications, closed gas turbine cycles or hot air turbines are also being considered in various research and development projects. However, they are characterised by high complexity for multistage expansion, recuperation, and multistage compression, if significantly higher efficiencies than in steam plants are to be reached. The use of existing components for a simple process design will probably result in moderate efficiency. Before market introduction of such technologies, both process and component design (especially heat exchanger and/or hot gas particle separation) need to be further developed.

5 EMISSIONS FROM BIOMASS COMBUSTION

5.1 Introduction

In this chapter, emission reduction measures applicable for biomass combustion will be presented. First the emissions from complete biomass combustion are presented, followed by the emissions from incomplete combustion. Particle emissions can be the result of both complete and incomplete combustion and are treated in both sections.

The various emissions components are first presented as well as their sources and impacts on climate, environment and health. This is followed by an overview of typical emissions levels from various (industrial) biomass combustion applications. These emission levels will be very dependent on the emission reduction measures implemented and are only guiding values reflecting today's typical biomass combustion applications, not the emission reduction potential.

5.2 Emissions from complete combustion

The following emissions are emitted to the atmosphere as a result of complete combustion in biomass combustion applications.

Carbon Dioxide (CO_2)

CO_2 is a major combustion product from all biomass fuels, originating from the carbon content in the fuel. In general, the emission of CO_2 to the atmosphere is problematic, since CO_2 is considered a major cause of the greenhouse effect. However, during the combustion of biomass, not more CO_2 is developed than bound during its growth. Furthermore, during combustion the same amount of CO_2 is developed as during the decomposition that is the final alternative to the use of biomass for energy purposes. Biomass is thus considered CO_2-neutral.

Nitric oxides (NO_x)

Nitric oxides NO and NO_2, summarised as NO_x, from combustion processes are formed in three different processes:

- Thermal NO_x are formed at high temperature (relevant concentrations can be found >1300 °C) by the oxidation of nitrogen in the air;
- Prompt NO_x can be formed during the combustion of hydrocarbons of molecular nitrogen with free radicals in the frame;
- Fuel NO_x are formed from the nitrogen contained in the fuel.

Fuel NO_x as an oxidation product are usually maximally at high combustion quality. Since typical combustion temperatures in today's biomass firing plants are between 800 and 1200 °C only fuel NO_x are of great importance.

Fuel NO_x emissions increase with increasing nitrogen content in the biomass, excess air ratio, and combustion temperature, up to a point where all fuel nitrogen intermediates have been converted to either NO_x, N_2O, or N_2. However, the fraction of fuel nitrogen converted to NO_x decreases with increasing nitrogen content in the fuel, as can be seen in Figure 5.1. Figure 5.2 illustrates the relative importance of the fuel, thermal and prompt NO_x mechanisms as a function of combustion temperature.

The emission of NO_2 is usually much smaller than the emission of NO. NO_x emissions may be reduced by both primary and secondary emission reduction measures.

Figure 5.1 Measured fraction of fuel nitrogen converted to NOx for various wood fuels in various wood combustion applications as a function of fuel nitrogen content, together with a trend line

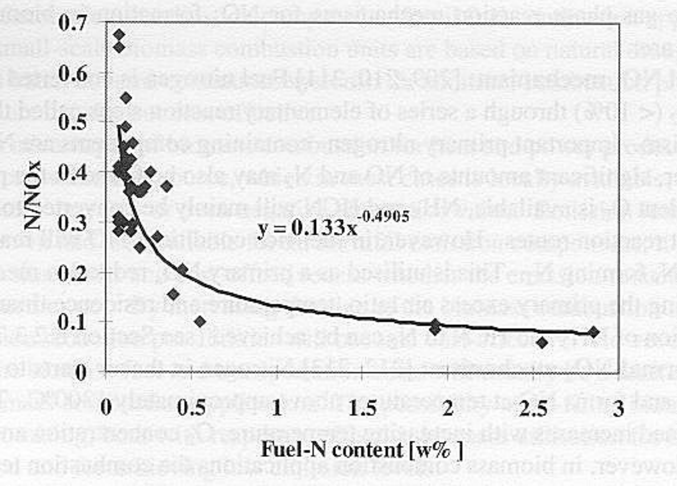
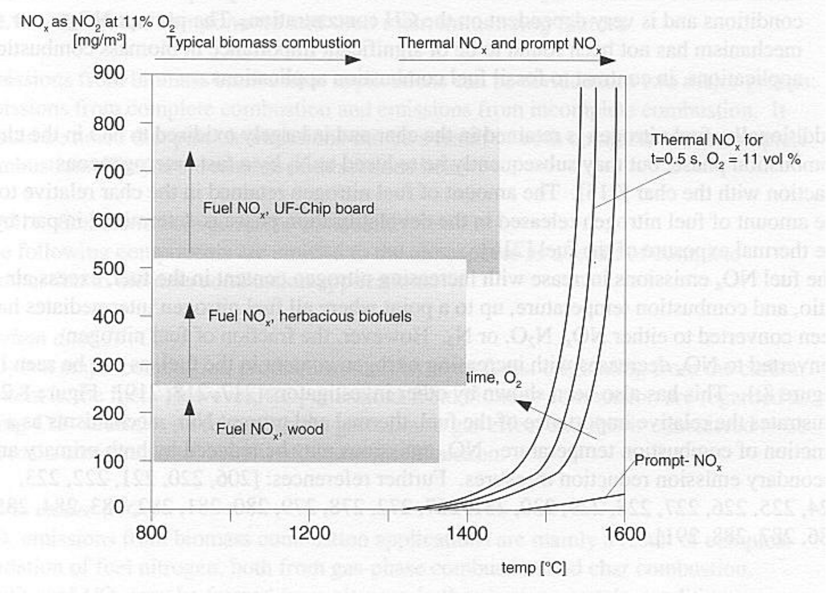


Figure 5.2 Level and types of NO_x emission as a function of temperature and fuel type



Nitrous oxides (N_2O)

N_2O emissions are the result of complete oxidation of fuel nitrogen. The N_2O emissions of biomass combustion are very low but they contribute to some degree to the greenhouse effect because of the high Global Warming Potential (GWP) factor of N_2O and to ozone depletion in the atmosphere. N_2O emissions may be reduced by primary emission reduction measures.

Sulphur Oxides (SO_x)

Sulphur oxides are a result of complete oxidation of fuel sulphur. Mainly SO_2 (>95%) is formed. However, some SO_3 (<5%) may be formed at lower temperatures. Sulphur from the combustion of biomass comes from sulphur compounds that have been absorbed by the tree during its growth. Therefore, the combustion of biomass does not change the total amount of sulphur present in the environment, but it entails that the emission of sulphur with the smoke contributes to the pollution of the air. However, fresh wood from the forest contains only a very limited amount of sulphur, and the sulphur content in fuel chips often show values that are below the laboratory equipment limits of detection. Fuel sulphur will not be completely converted to SO_x; a significant fraction will remain in the ashes while a minor fraction is emitted as salt (K_2SO_4) or as H_2S at lower temperatures. SO_2 emissions may be reduced by primary measures such as lime or limestone injection or by secondary measures.

Firing with biomass at heating plants causes much less SO_2 emission than the fuel oil or coal the biomass often replace. If the alternative is natural gas, and if it is sulphur-free at production, there will be no SO_2 advantage by using biomass as a fuel.

Hydrogen chloride (HCl)

Part of the chlorine content in the fuel will be released as HCl. The chlorine content of wood is very low. However, significant amounts of HCl may be formed from biomass fuels containing higher amounts of chlorine, such as miscanthus, grass and straw, or when wood chips from near-shore forests containing salt from sea fog are used.

The fuel chlorine will not be completely converted to HCl; the main fraction is retained in salts (KCl, NaCl) by reaction with K and Na, while traces are emitted as dioxins and organic chlorine components. Like sulphur dioxide, hydrogen chloride (HCl) contributes to the acidification, but condenses faster (to hydrochloric acid) and can therefore locally contribute to damage to materials in particular, but also to plants.

HCl emissions may be reduced by washing of fuel, which is utilised to some extent for straw due to its high chlorine content, and by secondary emission reduction measures.

Particles

Particles originate from several sources. Among these are fly ash, which is a result of entrainment of ash particles in the flue gas, and salts (KCl, NaCl, K_2SO_4) which is a result of reactions between K or Na and Cl or S. Other particles, occurring as a result of incomplete combustion, are discussed in Section 5.3. Secondary particle emission reduction measures are utilised to reduce the particle emission level in large-scale biomass combustion applications. By optimal design of combustion chambers, fly ash particles entrained in the flue gas may to some degree be prevented from leaving the combustion chamber, instead falling down to the bottom of the combustion chamber to be removed as bottom ash.

Heavy metals

All virgin biomass fuels contain heavy metals to some degree (most important: CU, Pb, Cd, and Hg). These will remain in the ash or evaporate, and also attach to the surface of particles emitted to the atmosphere or be contained inside fly-ash particles. Contaminated biomass fuels, such as impregnated or painted wood may contain significantly higher values of heavy metals. One example is the presence of Cr and As in impregnated wood. Heavy-metal emissions can be reduced by secondary emission reduction measures.

5.3 Emissions from incomplete combustion

The components below are emitted to the atmosphere as a result of incomplete combustion in biomass combustion appliances. As already indicated in Section 2.6 emissions caused by incomplete combustion are the results of either:

- inadequate mixing of combustion air and fuel in the combustion chamber, which produces local fuel-rich combustion zones;
- an overall lack of available oxygen;
- too low combustion temperatures;
- too short residence times.

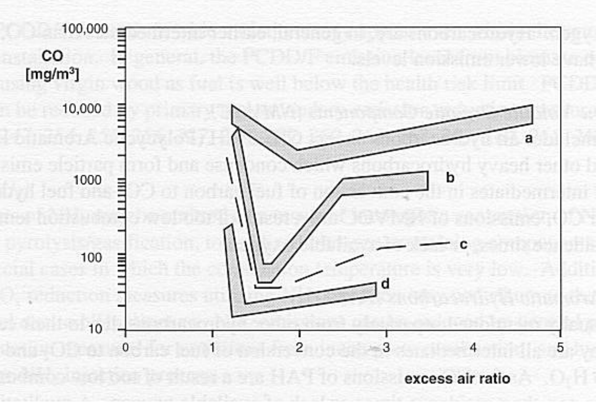
These variables are linked together, however, in cases in which sufficient oxygen is available temperature is the most important variable.

Carbon Monoxide (CO)

Conversion of fuel to CO_2 takes place through several elementary steps, and through different reaction paths. CO is the most important final intermediate. It is oxidised to CO_2 if oxygen is available. The rate at which CO is oxidised to CO_2 depends primarily on temperature. CO can be regarded as a good indicator of the combustion quality.

Large-scale combustion applications usually have better opportunities for optimisation than small-scale combustion applications. Hence, CO emissions are usually higher for the latter. Figure 5.3 shows the CO emission level as a function of excess air ratio for various biomass combustion applications, while Figure 5.4 shows the CO emission level as a function of combustion temperature. For a given system, there exists an optimum excess air ratio: higher excess air ratios will result in decreased combustion temperature while lower excess air ratios will result in inadequate mixing conditions. In addition, sufficient residence time is important to achieve low CO emission levels, mainly because CO is generally a later intermediate than hydrocarbons.

Figure 5.3 CO emissions as a function of excess air ratio λ



Explanation:

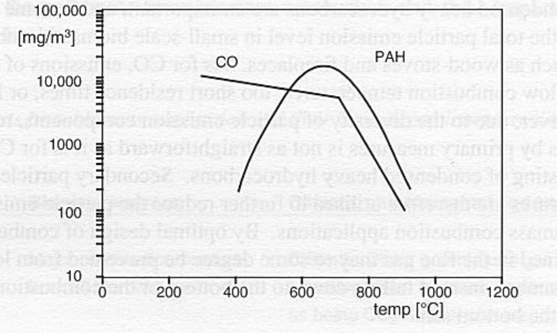
a) a single, manually charged wood boiler

b) a downdraft wood log boiler

c) an automatic furnace with combustion technology as of 1990

d) an automatic furnace with enhanced combustion technology as of 1995.

Figure 5.4 CO emissions in mg/Nm3 as a function of combustion temperature, together with a qualitative comparison with the influence of combustion temperature on PAH emissions



Unburned hydrocarbons (C_xH_y)

Methane (CH_4) is usually mentioned separately from other hydrocarbons since it is a direct greenhouse gas. In biomass combustion it is an important intermediate in the conversion of fuel carbon to CO_2 and fuel hydrogen to H_2O. Other unburned hydrocarbons (C_xH_y), also referred to as Non-Methane Volatile Organic Components (NMVOC), are also the products of incomplete combustion. They originate during the devolatisation of the fuel. As earlier intermediaries than CO the emission of C_xH_y are generally lower. Polycyclic Aromatic Hydrocarbons (PAHs), a group of hundreds of organic components consisting of two or more aromatic rings, constitute a special category of hydrocarbons due their carcinogenic effects. As for CO, emissions of methane, NMVOC and PAHs are a result of too low combustion temperatures, too short residence times, or lack of available oxygen.

Particle

Particle emissions from incomplete combustion can be found as soot, char, or condensed heavy hydrocarbons (tar). Soot consists mainly of carbon, and is the result of a local lack of available oxygen in the flame zone and/or local flame extinction. Char particles may be entrained in the flue gas due to their very low specific density, especially at high flue gas flow rates. As for CO, emissions of particles may be a result of too low combustion temperatures, too short residence times, or lack of available oxygen. However, due to the diversity of particle emission components, reducing particle emission levels is not as straightforward as it is for CO, except for particles consisting of condensed heavy hydrocarbons. Secondary particle emission reduction measures are therefore utilised to further reduce the particle emission level in large-scale biomass combustion applications. By optimal design of combustion chambers, particles entrained in the flue gas may to some degree be prevented from leaving the combustion chamber, instead falling down to the bottom of the combustion chamber to be removed with the bottom ash.

Polychlorinated dioxins and furans (PCDD/F)

Polychlorinated dioxins and furans are a group of highly toxic components. They can originate during the combustion of chlorine-containing biomass fuels in the temperature range 180-500°C and in the presence of carbon, catalysts (Cu) and oxygen. The emissions of PCDD/F are highly dependent on the conditions under which combustion and flue gas cooling takes place; therefore, wide variations are found in practice. In general, the PCDD/F emission level from biomass combustion applications using virgin wood (e.g. forestry chips) as fuel is well beyond the health risk limit. However, the combustion of painted, impregnated, chlorine or Cu-containing wood can easily lead to the production of dioxins if the combustion plant is not specially designed for these fuels. PCDD/F emissions can be reduced by primary and secondary emission reduction measures.

Ammonia (NH_3)

Small amounts of NH_3 may be emitted as a result of incomplete conversion of NH_3 to oxidised nitrogen-containing components. This occurs in special cases in which the combustion temperature is very low. Additionally, secondary NO_x reduction measures utilising NH_3 injection may contribute to the NH_3 emission level due to NH_3 slippage. NH_3 emissions may be reduced by general primary emission reduction measures for emissions from incomplete combustion, and by optimising the NH_3 injection process.

(Ground level) Ozone (O_3)

O_3 is a secondary combustion product formed from photochemical atmospheric reactions including CO, CH_4, NMVOC, and NO_x. It is a direct greenhouse gas and also influences the local and regional environment. As such, it is a highly unwanted by-product of biomass combustion appliances. O_3 emissions can be reduced indirectly by reducing emissions from incomplete combustion, and by NO_x emission reduction measures.

5.4 Emission levels

Emissions from biomass combustion can generally be divided into emissions that are mainly influenced by combustion technology and process conditions, and emissions that are mainly influenced by fuel properties. The amount of pollutants emitted to the atmosphere from various types of biomass combustion appliances is highly dependent on the combustion technology implemented. When available, such data generally refer to a single fuel-technology combination. In order to obtain an objective view of emission levels from various combustion applications, it is necessary to collect emission data from a wide range of fuel-technology combinations.

A problem in trying to compare emission levels from different biomass combustion applications is that many different denominators are used. Usually the basic data needed to recalculate the emission level into other denominators are not given, and assumptions must be made.

Typical ranges of emission levels from industrial wood furnaces are presented in Table 5.1 and Table 5.2.

Table 5.1 Emissions that are mainly influenced by combustion technology and process conditions

Level of standard	Poor standard	High standard
Excess-air ratio, λ	2-4	1.5-2
Component	**Emissions (mg/m^3_0 at 11% O_2)**	
CO	1000-5000	20-250
C_xH_y	100-500	< 10
PAH	0.1-10	< 0.01
Particles	150-500	50-150$^{*)}$

Explanation: Data from investigations various automatic industrial wood furnaces (understoker furnaces, grate firings, and dust firings) in Switzerland. $^{*)}$: except for dust firings, usually >150 mg/m^3_0

Table 5.2 Emissions that are mainly influenced by fuel properties

Component	**Fuel type**	**Typical emissions (mg/m$^3_0$ at 11% O_2) *)**
NO_x	Native wood (soft wood) Native wood (hard wood) Straw, grass, miscanthus, chip boards Waste and demolition wood	100-200 150-250 300-800 400-600
HCl	Native wood Waste and demolition wood, straw, grass, miscanthus, chip boards (NH_4Cl)	<5 raw gas: 100-1000 with HCL absorption: <20
Particles	Native wood Straw, grass, miscanthus, chip boards Waste and demolition wood	after cyclone: 50-150 after cyclone: 150-1000 after bag filter or ESP: <10
Σ Pb, Zn, Cd, Cu	Native wood Waste and demolition wood	<1 raw gas: 20-100 after bag filter or ESP: <5
PCDD/F (in ng TE/m$^3_0$)	Native wood Waste and demolition wood	0.01 - 0.5 (typical <0.1) 0.1 - 20 (typical 2)

Explanation: Data from investigations various automatic industrial wood furnaces (understoker furnaces, grate firings, and dust firings) in Switzerland. *): PCDD/F emission in ng TE/m$^3_0$)

Table 5.3 shows the results of an Austrian assessment of emissions data from various biomass combustion plants, representing capacities ranging from 0.5 to 10 MW$_{th}$. The fuels used were particle board, wood chips, MDF and bark.

Table 5.3 Emissions from wood-fired installations, using particle board, wood chips, MDF and bark

Component	**Emission (mg/m^3 at 11% O_2 dry)**	**No. observations**
CO	125-1000	25
C_xH_y	5.0-12.5	25
PAH	0.00006 - 0.06	unknown
Benzo[a]pyrene	$5 \cdot 10^{-6}$ - $1.0 \cdot 10^{-3}$	4
NO_x (as NO_2)	162 - 337	22
Particles	37 -312	29
SO_2	19 -75	17
Cl	1 -10	12
F	0.25	unknown

It can be seen from Table 5.3 that the emissions of SO_2, Cl and F are relatively low. This is due to the low content of these elements in the fuel. The emissions of C_xH_y, PAH and Benzo[a]pyrene are also low. By further optimisation of the combustion process they can be further reduced. Emissions of CO were found to be relatively high, particularly for old combustion installations.

CO emissions can be reduced by avoiding intermittent boiler operation through improved combustion process control (see Section 5.5.4). NO_x emissions can be further reduced by applying/optimising staged combustion, see Section 5.5.5 and 5.5.6. From a study of the individual measurements carried out as part of the Austrian assessment, the following was concluded:

- The emissions usually decrease as the size of the combustion installation increases, due to improved process control possibilities and efficient flue gas cleaning facilities. Combustion installations exceeding 4 MW_{th} often have electrostatic filters or flue gas condensation units in addition to a cyclone, which is usually installed in smaller installations. For smaller combustion installations, such investments are usually not economically viable. However, in Scandinavian installations, economic flue gas condensation has been achieved with boiler installations below 1 MW_{th}.
- NO_x emissions are an exception. The fuel NO_x emissions increase with increasing nitrogen content in the fuel, excess air ratio, and combustion temperature, up to a point where all fuel nitrogen intermediates have been converted to either NO_x, N_2O, or N_2. At low combustion temperatures, the temperature influence is more important than the influence of excess air ratio, resulting in lower NO_x emissions for smaller combustion installations.

5.5 Primary emission reduction measures

Reduction of harmful emissions can be obtained by either avoiding creation of such substances (primary measures) or removing the substances from the flue gas (secondary measures). In this section, measures for reducing primarily emissions from incomplete combustion and NO_x are presented.

Primary emission reduction measures aim at preventing or reducing formation of emissions and/or reduction of emissions within the combustion chamber. Several possible measures exist. Measures discussed here include:

- modification of the moisture content of the fuel,
- modification of the particle size of the fuel,
- selection of the type of combustion equipment,
- combustion process control optimisation,
- staged-air combustion,
- staged fuel combustion and reburning,

In practice, these measures often are interrelated.

5.5.1 Modification of the moisture content of the fuel

The moisture content in biomass can vary widely. Wood residuals from wood industry where drying has been applied may contain as little as 10% (w.b.) moisture, whereas fresh wood from the forest may contain up to 60% (w.b.) of moisture.

A high moisture content in the fuel makes it difficult to achieve a sufficiently high temperature in the combustion chamber. Often a temperature above 850^0C is desired to ensure a sufficiently low level of CO. If high temperature is not reached, incomplete combustion occurs with high emissions as a result.

In general, unless waste heat from another process can be accessed at a very low cost, the cost of artificial drying is too high to make the drying process itself economically feasible. The design of the combustion chamber is of great importance when biomass with a high moisture content is used. Improvement of the combustion quality can be reached by use of a high amount of ceramic linings and insulation of the combustion chamber. This measure, together with a high pre-heating temperature of the combustion air, may make it possible to utilise fuel with a high moisture content in an environmentally acceptable manner. Even in cases where improvement of the combustion process is achieved, a certain decrease in the boiler efficiency must be accepted. This is due to the fact that the amount of moisture in the fuel leads to a higher flow of flue gas, including water vapour, from the boiler. This represents an energy loss. But combustion of wood with high moisture content can be advantageous if combined with a flue gas condensing system, and provided there is a sufficiently low heat sink. Condensation of the water vapour in the flue gas raises the overall efficiency to such an extent that improvement of the overall economy of plant operation may be achieved.

5.5.2 Modification of the particle size of the fuel

The fuel particle size is very relevant for the combustion technology selection process. Especially in large-scale biomass combustion applications with automatic fuel feeding, the fuel size can be decisive. The fuel size in biomass combustion applications may vary from whole wood logs to fine sawdust. If the fuel consists of both very small and very large pieces, a shredder or chipper can be used to reduce the particle size of the largest particles. In this way, a more homogeneous particle size is obtained and a wider range of technology options can be used. However, particle size reduction is only attractive if the benefits outweigh the additional investment and energy costs.

5.5.3 Selection of the type of combustion equipment

When selecting the combustion technology for a biomass combustion application there are several aspects to be considered, both with respect to the combustion process and to primary and possibly secondary emission reduction measures. Also, the heat/power capacity of the application usually limits the choice of combustion technology, either due to technological or economic considerations.

First of all, fuel characteristics such as fuel composition, moisture content, and particle size are important. For wood fuels only the nitrogen content may limit the choice of combustion technology, if there are NO_x emission limits to be met. The moisture content, however, will be very decisive for wood fuels such as wood chips and bark if drying of the fuel prior to combustion is not an option. For other types of biomass fuels, additional fuel constituents, such as ash, chlorine, potassium and sulphur may influence the combustion process in such a way that certain preferences should be made when selecting the combustion technology, e.g. for straw combustion.

5.5.4 Combustion process control optimisation

In order to obtain optimal combustion, with minimal emissions from incomplete combustion, sufficiently high combustion temperatures, sufficiently long residence times, and optimal mixing of fuel gases and air, also with changing heat and/or power output have to be achieved. These factors are partly determined by the combustion technology and design of the furnace, and partly by the combustion process operation. For the optimisation of combustion processes a number of combustion process control systems have been developed.

A process control systems, or process controller, aims to govern selected process parameters according to a predefined scheme. The primary aim of a process control device in a biomass combustion application is to adjust the heat production according to the heat demand. In addition to this, the process control device can be programmed for simultaneous optimisation of the combustion process with respect to minimising emissions and maximising thermal efficiency. For biomass combustion, typical process parameters that can be used as process control parameters are the CO, C_xH_y and O_2 concentrations in the flue gas, as well as combustion chamber temperatures and boiler temperature. Process variables that can directly be adjusted to achieve the targets for the mentioned process parameters are typically the amount of fuel fed into the furnace, and the amount of primary and secondary combustion air supplied.

Minimising emissions

The combustion quality can be modified by adjusting the amounts of fuel and primary and secondary air, based on measured concentrations of CO, C_xH_y, O2 and the combustion chamber temperature.

In the case of direct process control, CO and C_xH_y are measured continuously and the governing variables are adjusted to obtain minimum emissions. Because of process fluctuations, the concentrations of CO and C_xH_y often remain high.

In the case of indirect process control, the ideal excess air ratio (λ) is first established for all expected process conditions (boiler load, fuel moisture content, etc.) to obtain minimum emissions. Then the measured value for O_2 is used as a process control parameter. Control of λ ensures a stable combustion process, but as the actual process parameters often deviate from expected values, emissions are not always minimised in practice.

Direct and indirect process control can also be combined to obtain a stable combustion process with minimised emissions.

Controlling heat output

In addition to minimising emissions, there is a need to control the heat output from the furnace or boiler. This control can be based on using measured temperature difference and mass flow of boiler water. However, the control of the boiler water temperature is more commonly used. The relation between fuel input and the input of primary and secondary air is established after the boiler is installed. Using this relation, these parameters can be adjusted to keep the boiler water temperature at a given value.

Modification of an existing biomass boiler

Existing boilers can often be modified to successfully achieve reduced emission levels, higher thermal efficiency, and improved control of heat output. As experiments by e.g. the Dutch research institute TNO demonstrate, boiler emissions can be reduced and efficiency increased with relatively simple process modifications.

5.5.5 Staged-air combustion

Staged-air combustion is widely applied in biomass combustion applications. However, the possibilities for an accurate control of the combustion air are usually limited in small-scale applications, which may result in higher emission levels. Staged-air combustion makes a simultaneous reduction of both emissions from incomplete combustion and NO_x possible through a separation of devolatilisation and gas phase combustion. This results in improved mixing of fuel gas and combustion air. In the first stage, primary air is added for devolatilisation of the volatile fraction of the fuel, resulting in a fuel gas consisting mainly of CO, H_2, C_xH_y, H_2O, CO_2 and N_2. For NO_x emission reduction, also the fuel gas content of NH_3, HCN and NO is of particular interest. In the second stage, sufficient secondary air is supplied to ensure a good burnout and low emission levels from incomplete combustion.

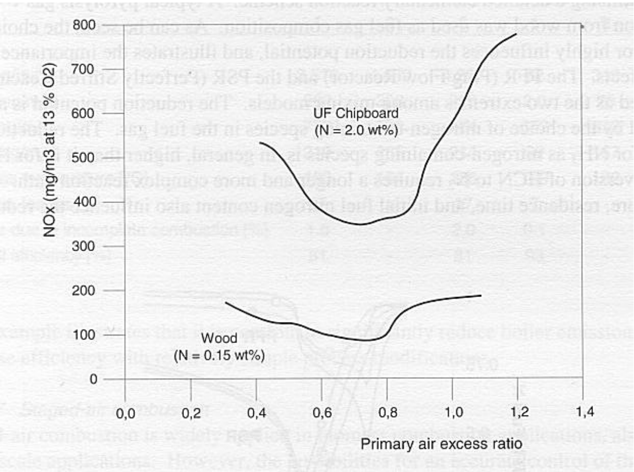
An improved mixing of fuel gas and secondary air reduces the amount of secondary air needed, resulting in higher flame temperatures, and also a lower overall excess air ratio. Hence, emissions from incomplete combustion are reduced by a temperature increase, which speeds up the elementary reaction rates, and improved mixing, which reduces the residence time needed for mixing the fuel gas and the secondary combustion air. However, this does not mean that the NO_x emission level is automatically reduced as well. An efficient reduction of both emissions from incomplete combustion and NO_x emissions can only be achieved by optimisation of the primary excess air ratio.

As mentioned before, fuel nitrogen is converted to NO (> 90%) and NO_2 (< 10%) through a series of elementary reaction steps, called the fuel NO_x mechanism. Important primary nitrogen-containing components are NH_3 and HCN. However, significant amounts of NO and N_2 may also be found in the pyrolysis gas. If sufficient O_2 is available, NH_3 and HCN will mainly be converted to NO through different reaction routes. However, in fuel-rich conditions NO will react with NH_3 and HCN, forming N_2. This mechanism is utilised as a primary NO_x reduction measure. By optimising the primary excess air ratio, temperature and residence time a maximum conversion of NH_3 and HCN to N_2 can be achieved.

To investigate the NO_x reduction potential though staged-combustion, Verenum Research in Zurich built a 25 kW test reactor with fixed bed updraught gasification followed by gas phase combustion. The test reactor can be operated as an understoker firing, which allows receiving reference NO_x values for conventional combustion. In the reactor it has been shown that a NO_x reduction of up to 50-75% can be achieved by staged-air combustion (see Figure 5.5). The percentage increases with increasing fuel-N content, at optimum conditions:

- residence time in the reduction chamber ~0.5 s (> 0.3 s);
- reduction chamber temperature ~ 1100- 1200 ^{o}C;
- primary excess air ratio ~ 0.7.

Figure 5.5 NO_x emission level as a function of primary excess air ratio for the 25 kW test reactor at Verenum Research.



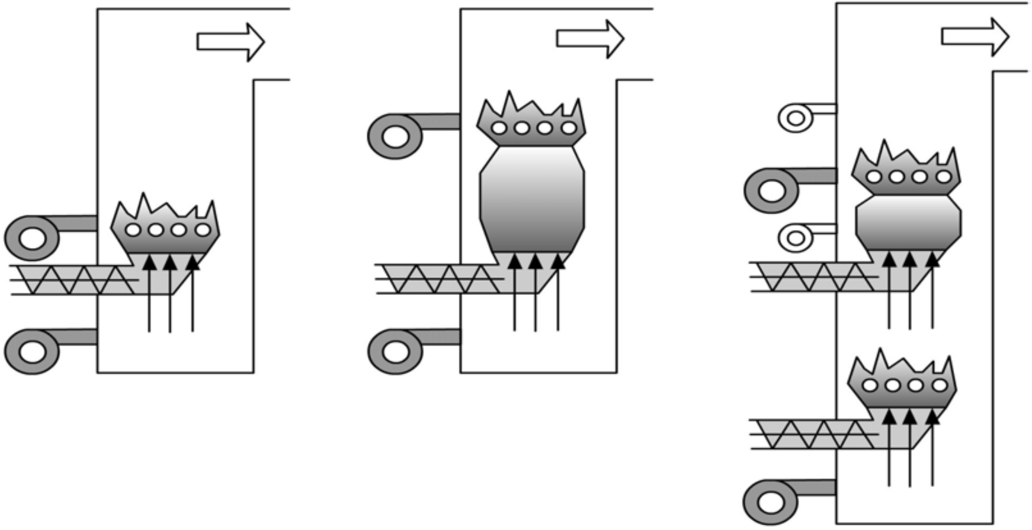
5.5.6 Staged fuel combustion and reburning

Staged fuel combustion and reburning are other possible methods for NO_x reduction in biomass combustion applications. The primary fuel is combusted with an excess air ratio above 1, and no significant NO_x reduction occurs. A secondary fuel is then injected into the flue gas after the primary combustion zone, without additional air supply. A sub-stoichiometric reducing atmosphere is created in which NO_x formed in the primary zone may be reduced, by reactions with NH_3 and HCN formed from the secondary fuel (if the secondary fuel contains nitrogen), in a similar manner as for staged-air combustion (see Figure 5.6. Additionally, NO is converted back to HCN by reactions with HCCO and CH_i radicals (i = 0-3) formed from the secondary fuel. This is called reburning. Under typical reburning conditions, HCCO appears to be the most effective radical for removing NO. Finally, a sufficient amount of air is added after the reducing zone to achieve a good burnout with an overall excess air ratio above 1.

The potential of staged fuel combustion was investigated in the understoker furnace at Verenum Research, where the secondary fuel was introduced on a second grate above the main fuel bed with an energy input ratio of approximately 70% primary and 30% secondary fuel. An NO_x reduction of 52-73% was achieved with a temperature in the reduction zone of approximately 700^{o}C. Hence, an NO_x reduction potential similar to that of staged-air combustion can be achieved at a significantly lower temperature level in the reduction zone. The optimum overall excess air ratio in the reduction zone is approximately between 0.7 and 0.9. As is the case in staged-air combustion, residence time and initial fuel nitrogen content also influence the reduction potential. The properties of the secondary fuel will also be of importance, especially the fuel-N and the volatile content.

Staged fuel combustion requires automatic feeding of the primary and secondary fuel, and the secondary fuel must be easily adjustable. This limits the use of staged fuel combustion to large-scale biomass combustion applications since a suitable combustion process design with two fuel-feeding systems as well as an accurate combustion process control is necessary. Natural gas, fuel oil, pyrolysis gas, biomass powder, sawdust, or similar fuels can be used as secondary fuels.

Figure 5.6 Three principles of combustion. Diagrams from left to right: conventional combustion, staged-air combustion, and staged fuel combustion.



5.6 Secondary emission reduction measures

Secondary measures can be applied to remove emissions from the flue gas once it has left the boiler. For virgin wood combustion, particle removal is of particular relevance. For other types of biomass, additional secondary measures may be necessary.

In this section, emission reduction measures for removal of primarily particles and NO_x will be presented. Other components that also can be reduced by secondary measures are SO_x, HCI, heavy metals, and PCDD/F. However, secondary emission reduction measures for these components will not be presented in detail. For the removal of SO_x (mainly SO_2) wet, dry and wet-dry throwaway processes have been developed. SO_x emission levels from wood combustion are usually not significant. HCl emission levels are reduced in wet throwaway processes applied for SO_x reduction. Furthermore, adsorptives such as activated lignite can be used for a combined extraction of HCl, SO_2, and PCDD/F. PCDD/F emission levels can be reduced by an efficient particle separation at temperatures well below the temperature range of the de novo synthesis. Emissions of heavy metals can be significantly reduced in particle-collecting devices such as bag filters or electrostatic filters.

5.6.1 Particle control technologies

Not every particle control technology suits every need. Among the determining factors are the particle's size, required collection efficiency, gas flow size, allowed time between cleanings, the detailed nature of the particles, and the presence of tars in the flue gas. The following rules of thumb may be helpful in selecting particle control technologies for biomass combustion applications:

1. Sticky particles (e.g. tars) must be collected in a liquid, as in a scrubber, or in a cyclone, bag filter, or an electrostatic filter whose collecting surfaces are continually coated with a film of flowing liquid. There must also be a way to process the contaminated liquid thus produced.
2. Particles that adhere well to each other but not to solid surfaces are easy to collect. Those that do the reverse often need special surfaces, e.g. Teflon-coated fibres in filters that release collected particles well during cleaning.
3. The electrical properties of the particles are of paramount importance in electrostatic filters, and they are often significant in other control devices where friction-induced electrostatic charges on the particles can aid or hinder collection.
4. For non-sticky particles larger than about 5 µm, a cyclone separator is probably the only device to consider.
5. For particles much smaller than 5 µm normally electrostatic filters, bag filters, and scrubbers are considered. Each of these can collect particles as small as a fraction of a micron.
6. For large flows the pumping cost makes scrubbers very expensive; other devices are preferable.
7. Corrosion resistance and dew point must always be considered.

Table 5.4 presents the characteristics of some common particle control technologies in wood chips combustion whereas

Table 5.5 summarises the typical sizes of particles removed by several proven particle control technologies, and typical removal efficiencies.

Figure 5.7 shows the collection efficiencies for some advanced particle control technologies.

Table 5.4 Characteristics of selected particle control technologies

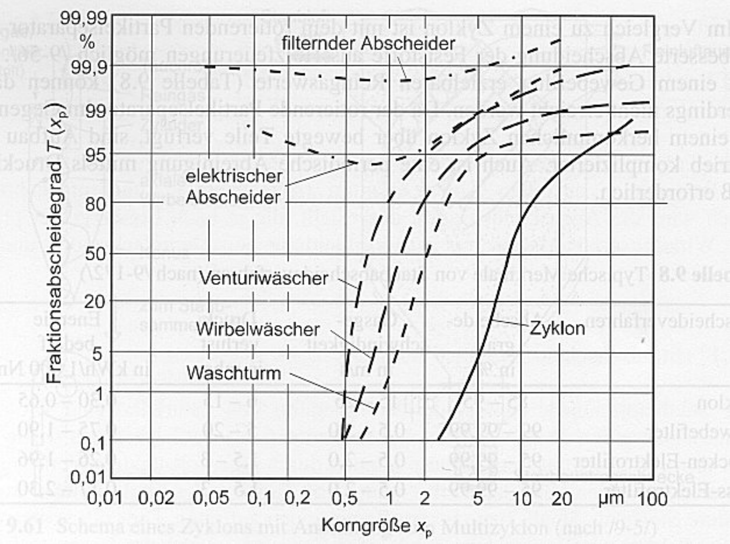
	Settling chamber	**Cyclone**	**Electrostatic filter**
Separation efficiency	~10% for particles < 30 µm; ~40% for particles < 90 µm	85-95%	95-99.99%
Gas velocity	1 - 3 m/s	15-25 m/s	0.5-2 m/s
Pressure drop	< 20 Pa	60-150 Pa	15-30 Pa
Temp. range	< 1300 ^{0}C	< 1300 ^{0}C	< 480 ^{0}C
Pressure range	< 100 bar	< 100 bar	< 20 bar
Application	First separation step	First or final particle separation step	Final particle separation step

Table 5.5 Summary of typical sizes of particles removed by various particle control technologies.

Particle control technology	Particle size (µm)	Efficiency (%)
Settling chambers	>50	<50
Cyclones	>5	<80
Multicyclones	>5	<90
Electrostatic filters	<1	>99
Bag filters	<1	>99
Spray chambers	>10	<80
Impingement scrubbers	>3	<80
Cyclone spray chambers	>3	<80

Venturi scrubbers	>0.5	<99

Figure 5.7 Collection efficiencies for various particle control technologies as a function of particle diameter



Explanation:
X-axis: particle size,
Y-axis: separation efficiency;
filternder Abscheider = bag filter,
elektrischer Abscheider = electrostatic filter;
Venturiwäscher = Venturi scrubber;
Wirbelwäscher = fluidised bed scrubber
Waschturm = spray chamber;
Zyklon = cyclone.

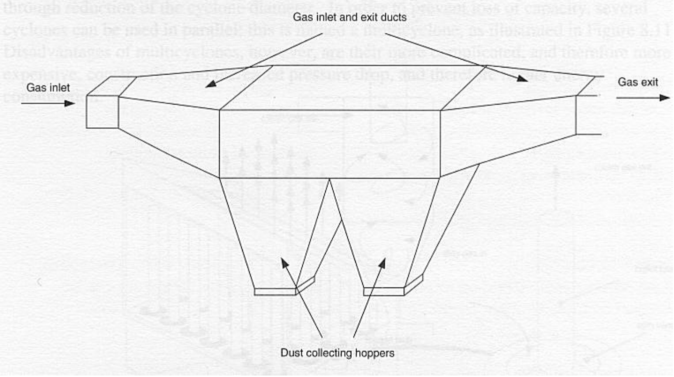
Below, the following particle control technologies will be discussed:

- settling chambers
- cyclones
- multicyclones
- electrostatic filters
- bag filters
- scrubbers
- rotating particle separator.

Settling chambers

Particle separation in a settling chamber is based on the principle of gravity, see Figure 5.8. The main disadvantage of this method is the low collection efficiency. However, it is still widely applied because of its ability to extinguish the flame. Typical characteristics of a settling chamber are given in Table 5.4.

Figure 5.8 Settling chamber



Cyclones

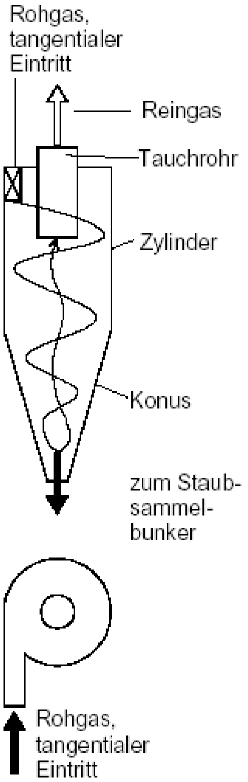
Particle separation in a cyclone is based on the principle of gravity in combination with centrifugal forces. Gas and solid particles are exposed to centrifugal forces, which can be done in two ways:

- gas flows into the cyclone in tangential direction;
- gas flows into the cyclone in axial direction, and is brought into rotation using a fan.

Because of the centrifugal forces, particles hit the wall and slide down into a container. Figure 5.9 illustrates the principle. Cyclones have higher collection efficiency than settling chambers due to the centrifugal force principle.

The energy loss is mainly determined by the pressure drop over the cyclone and is about 0.2 kWh per 1000 m^3 of flue gas. Typical characteristics of a cyclone are given in Table 5.4.

Figure 5.9 Principle of a cyclone



Explanation:

Raw gas, tangential inlet

Clean gas

Cylinder

Cone

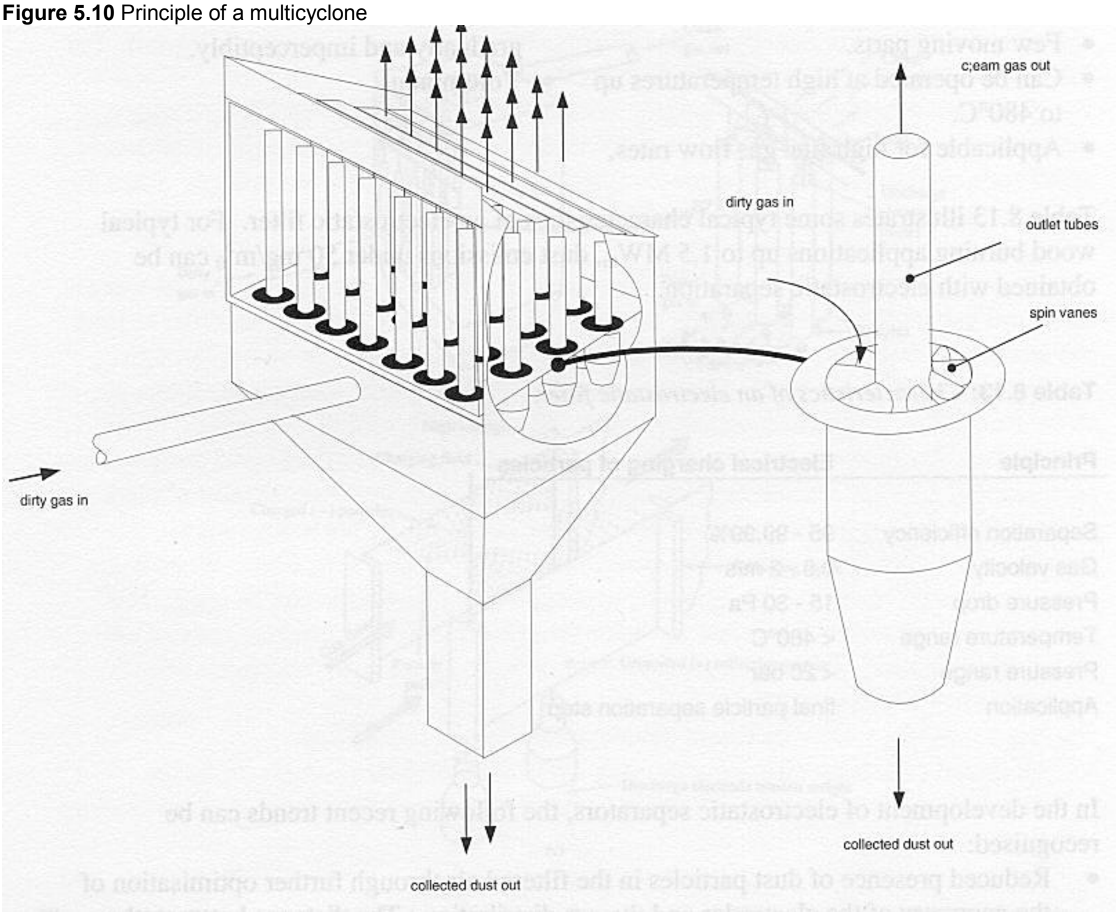
to dust hopper

Raw gas, tangential inlet

Multicyclones

The separation efficiency of a cyclone can be improved by increasing the centrifugal force through reduction of the cyclone diameter. In order to prevent loss of capacity, several cyclones can be used in parallel; this is named a multicyclone, as illustrated in Figure 5.10. Disadvantages of multicyclones, however, are their more complicated, and therefore more expensive, construction and increased pressure drop, and therefore higher energy consumption.

Figure 5.10 Principle of a multicyclone

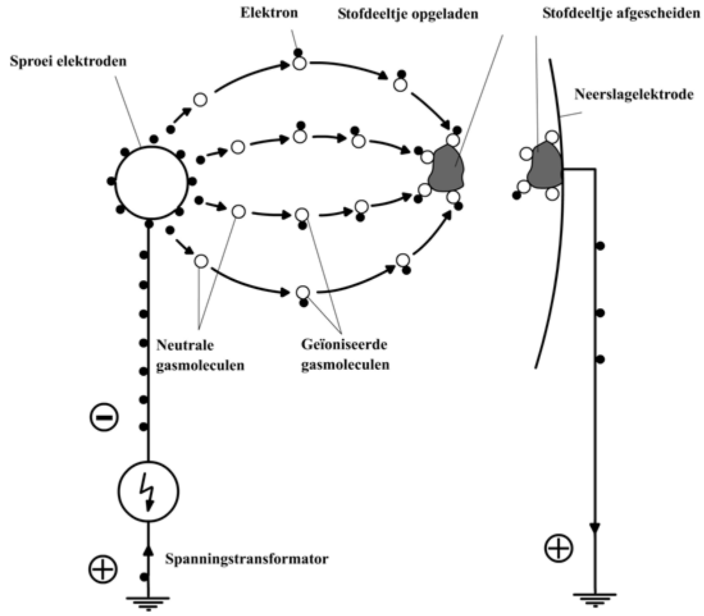


Electrostatic filter

In an electrostatic filter, the particles are first electrically charged. Then they are exposed to an electrical field in which they are attracted to an electrode. Periodically, this electrode is cleaned through vibration, by which the dust falls off the electrode into a collection unit. Figure 5.11 shows a picture of an installed electrostatic filter. The principle of an electrostatic filter is illustrated in Figure 5.12. Figure 5.13 shows different types of electrostatic filters. Typical characteristics of an electrostatic filer are given in Table 5.4.

Figure 5.11 An electrostatic filer installed at a biomass combustion plant

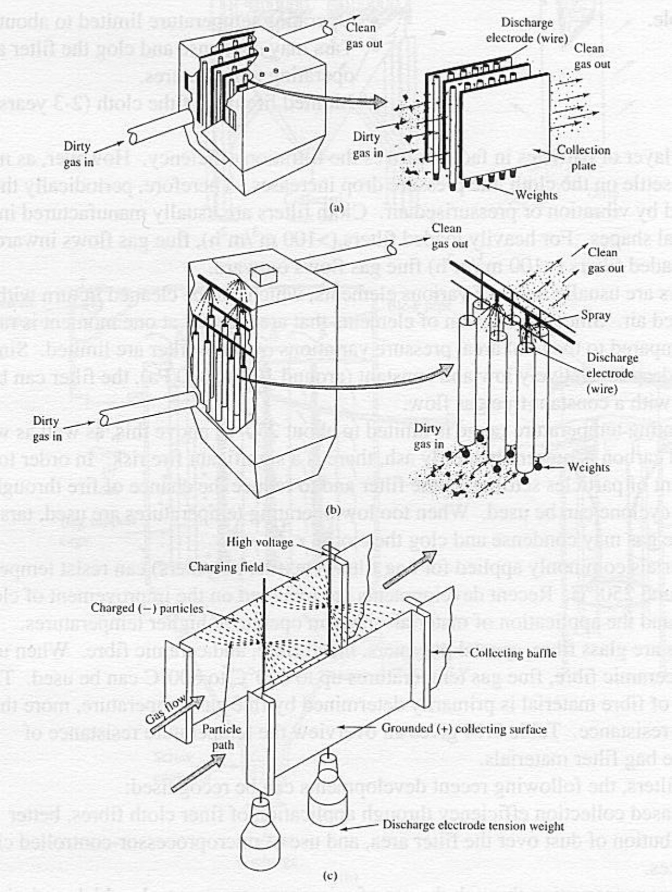
Figure 5.12 Principle of an electrostatic filter



<u>Explanation:</u>

Sproei elektroden = spray electrodes;
Elektron = electron;
Stofdeeltjes opgeldan = dust particle charged;
Stofdeeltje afgescheiden = dust particle separated,
Neerslagelektrode = discharge electrode;
Neutrale gasmoleculen = neutral gas molecules;
Geioniseerde gasmoleculen = ionised gas molecules;
Spanningstransformator = Voltage transformer;

Figure 5.13 Different types of electrostatic filters



(a) plate type, (b) tube type, (c) detailed view of plate type filter element

In practice, separation can be done in one or two stages. Most of the electrostatic filters found in practice are one-stage filters. In case of two-stage separation, charging of the particles is first done in a very strong electric field, after which a relatively weak field separates the particles. For typical wood burning applications up to 1.5 MW_{th}, dust emissions under 50 $mg/m^3{}_0$ can be obtained with electrostatic separation.

In the development of electrostatic filters, the following recent trends can be recognised:

- Reduced presence of dust particles in the filtered air through further optimisation of the geometry of the electrodes and the gas distribution. The distance between the electrodes has been increased to about 800 mm, so that the velocity of charged dust particles, and consequently the throughput, can be increased.
- Application of advanced, rnicroprocessor-based controllers of high-voltage generators and cleaning mechanisms, according to the filter load and specific dust characteristics. This effectively reduces the energy consumption.
- Application of new construction materials that allow operation above at >480"C.
- Application of pressure vessels and airtight isolators that allow operation at >20 bar.
- Application of pulsated electrode voltage, in order to limit reversed flow of highly charged particles.

Bag filters

As Figure 5.14 shows, the construction of a bag filter is relatively simple. It consists of a filter or cloth, tightly woven from special fibres and hung up in a closed construction through which flue gas passes. The separation efficiency of bag filters is quite high, even with high flue gas flow rates and high particle content.

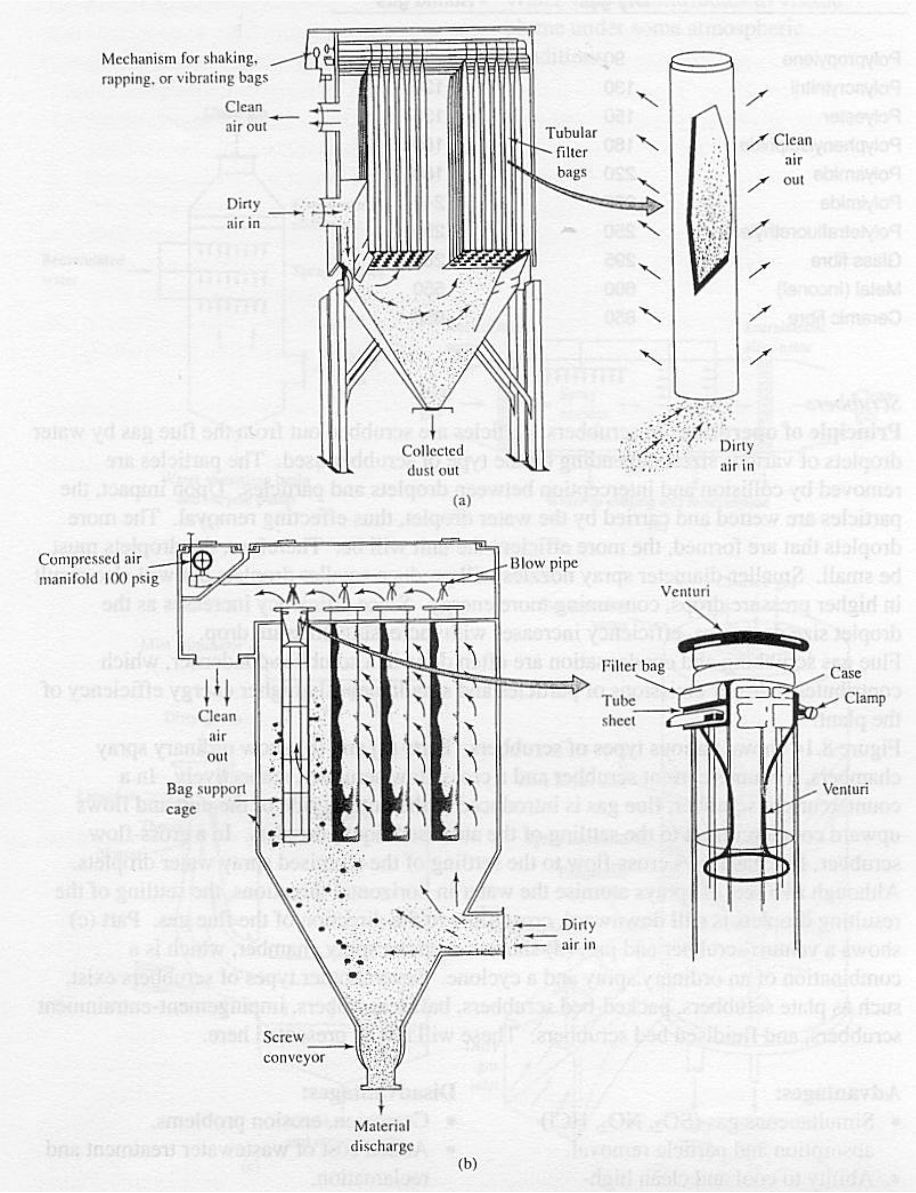
The first layer of particles in fact improves the filtration efficiency. However, as more particles settle on the cloth, the pressure drop increases. Therefore, periodically the cloth is cleaned by vibration or pressurised air. Cloth filters are usually manufactured in cylindrical shapes. For heavily loaded filters (>100 m^3/m^2h) flue gas flows inward. For lightly loaded filters (<100 m^3/m^2h) flue gas flows outward.

Bag filters are usually made of various elements, which can be cleaned in turn with pressurised air. Since the fraction of elements that are cleaned at one moment is rather small compared to the total area, pressure variations over the filter are limited. Since the pressure drop is relatively low and constant (around 1000-3000 Pa), the filter can be operated with a constant flue gas flow.

The operating temperature range is limited to about 250^{o}C; above this, as well as when unburned carbon is present in the fly ash, there is a significant fire risk. In order to limit the amount of particles settling on the filter and to reduce the chance of fire through sparks, a cyclone can be used. When too low operating temperatures are used, tars present in the flue gas may condense and clog the cloth.

The materials commonly applied for bag filters (textile, polymers) can resist temperatures up to around 250^{o}C. Recent developments are focussed on the improvement of cloth cleaning and the application of materials that can operate at higher temperatures. Examples are glass fibre, special polymers, metal fibre, and ceramic fibre. When using metal or ceramic fibre, flue gas temperatures up to 600 °C to 800 °C can be used. The selection of fibre material is primarily determined by flue gas temperature, more than chemical resistance.

Figure 5.14 Bag filters



(a) shaker bag filter, (b) pulse-jet bag filter

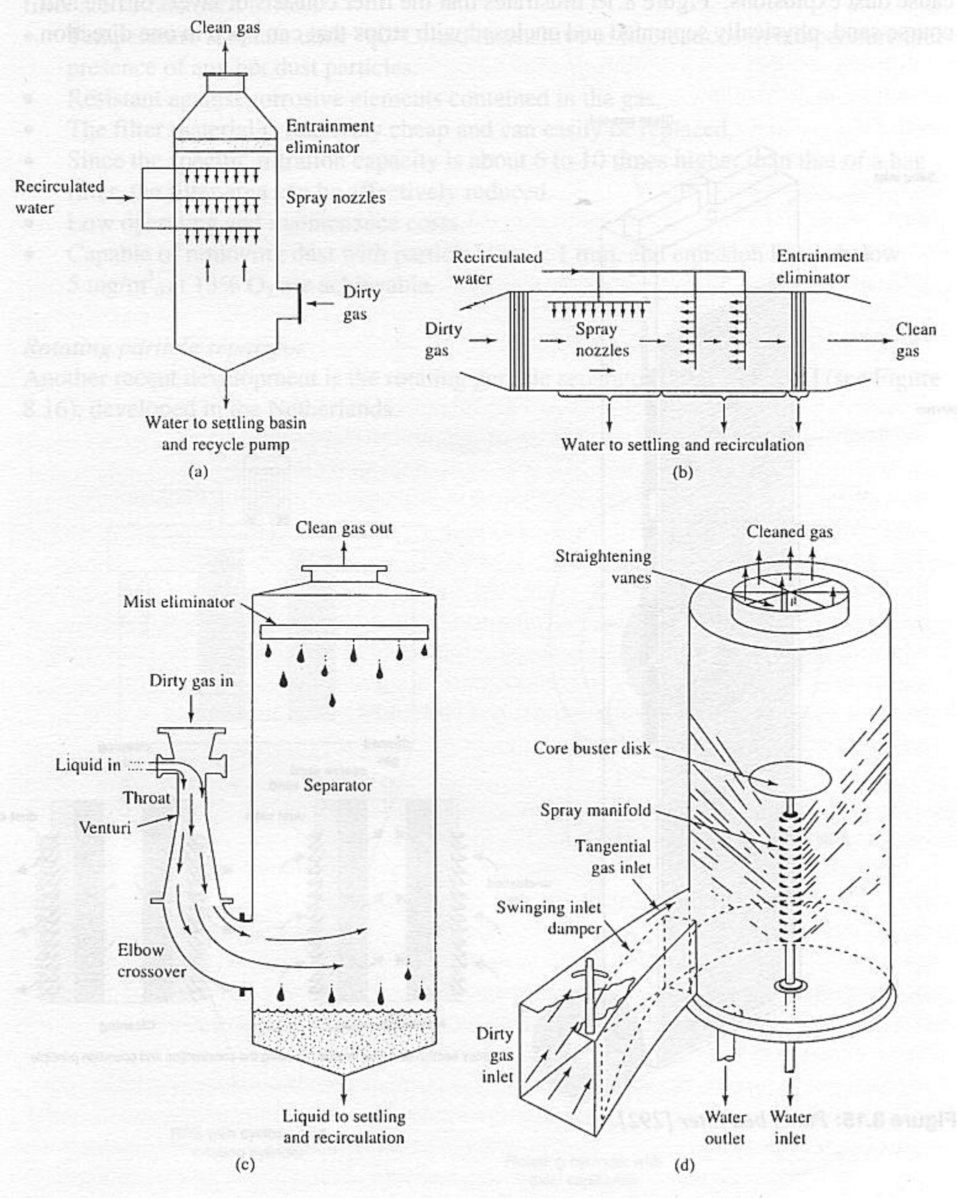
For bag filters, the following recent developments can be recognised:

- Increased collection efficiency through application of finer cloth fibres, better distribution of dust over the filter area, and use of microprocessor-controlled cleaning devices.
- Energy conservation through the use of microprocessor controls which optimises and reduces the use of pressurised air for cleaning of the filter.
- Reduction of the filter dimensions through increase of the specific filter area load, and optimisation of the geometry of various components.

Scrubbers

In scrubbers, particles are scrubbed out from the flue gas by water droplets of various sizes, depending on the type of scrubber used. The particles are removed by collision and interception between droplets and particles. Upon impact, the particles are wetted and carried by the water droplet, thus effecting removal. The more droplets that are formed, the more efficient the unit will be. Therefore, the droplets must be small. Smaller-diameter spray nozzles will produce smaller droplets but will also result in higher pressure drops, consuming more energy. Since efficiency increases as the droplet size decreases, efficiency increases with increasing pressure drop. Flue gas scrubbing and condensation are often done in a scrubber-condenser, which contributes to lower emissions of particles and simultaneously higher energy efficiency of the plant.

Figure 5.15 Scrubbers



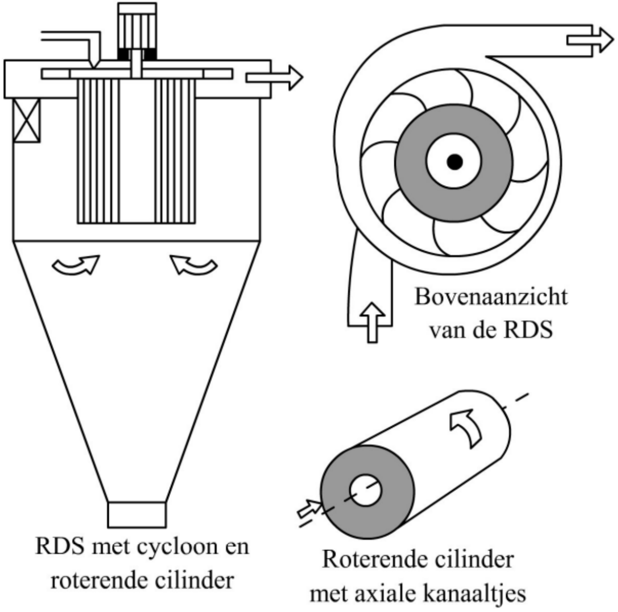
(a) counter-current spray chamber, (b) cross-flow spray chamber, (c) venture scrubber, (d) cyclone spray chamber

Figure 5.15 shows various types of scrubbers. Parts (a) and (b) show ordinary spray chambers, a counter-current scrubber and a cross-flow scrubber, respectively. In a counter-current scrubber, flue gas is introduced at the bottom side of the unit and flows upward counter-current to the settling of the atomised liquid droplets. In a cross-flow scrubber, flue gas flows cross-flow to the settling of the atomised spray water droplets. Although two sets of sprays atomise the water in horizontal directions, the settling of the resulting droplets is still downward, cross-flow to the direction of the flue gas. Part (c) shows a venturi scrubber and part (d) shows a cyclone spray chamber, which is a combination of an ordinary spray and a cyclone. Several other types of scrubbers exist, such as plate scrubbers, packed-bed scrubbers, baffle scrubbers, impingement-entrainment scrubbers, and fluidised bed scrubbers.

Rotating particle separator

A recent development is the rotating particle separator (see Figure 5.16).

Figure 5.16 Schematic view of the rotating particle separator



Explanation:

Left: RPS with cyclone and rotating cylinder

Top right: Top view of the RPS;

Bottom right: Rotating cylinder with axial capillaries

The technology can either be used together with a conventional multicyclone or instead of an electrostatic filter. Several field tests are presently being carried out in biomass combustion plants. The separation efficiency depends on the particle size. It is claimed that for particles greater than 1 μm, separation efficiencies up to 99.9% can be achieved. The fly-ash concentration can be limited to 50 mg/m^3_0 at 11% O_2.

In the rotating particle separator, flue gases first enter a circular movement by means of an integrated fan. Here, heavy particles are forced outwards by centrifugal forces and fall down into the ash pit, while the light particles pass through a rotating filter element. This filter element consists of a multitude of axial channels. The liquid and solid particles are pushed against the channel walls and are periodically removed by passing air or water flowing through these channels at high velocity.

Technological evaluation

Table 5.6 presents a technological evaluation of the various proven particle control technologies.

Table 5.6 Technological evaluation of various particle control technologies

Advantages	**Weak points**
Settling chambers • Low pressure loss • Simplicity of design and maintenance • High capacity. • Low costs. • Ability to extinguish the flame.	• Much space required. • Low collection efficiency.
Cyclones • Simplicity of design and maintenance. • Little floor space required. • Dry continuous disposal of collected dusts. • Low to moderate pressure loss. • Handles large particles. • Handles high dust loadings. • Temperature independent. • Low costs. • Ability to extinguish the flame.	• Much headroom required. • Low collection efficiency of small particles. • Sensitive to variable dust loadings and flow rates. • Tars may condense in the cyclone.
Electrostatic filters • Above 99% efficiency obtainable. • Very small particles can be collected. • Particles may be collected wet or dry. • Pressure drops and power requirements are small compared with other high-efficiency collectors • Maintenance is nominal unless corrosive or adhesive materials are handled. • Few moving parts. • Can be operated at high temperatures up to 480"C. • Applicable for high flue gas flow rates	• Relatively high initial costs. • Sensitive to variable particle loadings or flow rates. • Resistivity causes some material to be economically uncollectable. • Precautions are required to safeguard personnel from high voltage. • Collection efficiencies can deteriorate gradually and imperceptibly. • Voluminous.
Bag filters • Above 99% efficiency obtainable. • Dry collection possible. • Decrease of performance is noticeable. • Collection of small particles possible.	• Sensitive to filtering velocity. • High-temperature gases must be cooled. • Affected by relative humidity (condensation). • Susceptibility of fabric to chemical attack. • Voluminous. • Operating temperature limited to about 250 ^{0}C. • Tars may condense and clog the filter at low operating temperatures. • Limited lifetime of the cloth (2-3 years).

Scrubbers	
• Simultaneous gas (SO_2, NO_2, HCI) absorption and particle removal. • Ability to cool and clean high- temperature, moisture-laden gases. • Corrosive gases and mists can be recovered and neutralised. • Reduced dust explosion risk. • Efficiency can be varied.	• Corrosion, erosion problems. • Added cost of wastewater treatment and reclamation. • Low efficiency on sub-micron particles. • Contamination of effluent stream by liquid entrainment. • Freezing problems in cold weather. • Reduction of buoyancy and plume rise. • Water vapour contributes to visible plume under some atmospheric conditions.

5.6.2 NO_x control technologies

Nitrogen oxides (NO_x) are often lumped together with sulphur oxides (SO_x) as air pollution control problems, because of the similarities between the two:

- NO_x and SO_x react with water and oxygen in the atmosphere to form nitric and sulphuric acids, respectively. These two are the principal contributors to acid rain. Because the acid rain process removes both NO_x and SO_x from the atmosphere, neither is believed to be increasing in concentration in the global atmosphere.
- NO_x and SO_x undergo atmospheric transformations leading to or contributing to the formation of PM_{10} (particles of 10 μm or less in diameter) in urban areas.
- In high concentrations, NO_x and SO_x are severe respiratory irritants.
- NO_x and SO_x are released to the atmosphere in large quantities from fossil fuel combustion. Coal-fired power plants are the largest emitters. Emissions of NO_x and SO_x from biomass combustion, however, are substantially lower.

However, focusing on biomass combustion applications, the following major differences can be pointed out between fuel NO_x and SO_x formation:

- Formation of NO_x in combustion chambers can be greatly reduced by optimisation of the combustion process through primary NO_x emission reduction measures, such as staged-air combustion and staged fuel combustion. No such optimisation is practically possible for SO_x. However, in special combustion applications such as fluidised bed reactors, lime or limestone injection may be used to convert SO_x to $CaSO_4$, which then can be removed from the flue gas in the form of particles.
- The ultimate fate of sulphur oxides removed by pollution control or fuel-cleaning processes is to be turned into $CaSO_4$, which is an innocuous, low solubility solid, commonly deposited in landfills. There is no correspondingly cheap, innocuous, and insoluble salt of nitric acid, so landfilling is not a suitable solution for the NO_x collected in pollution control devices. The ultimate fate of NO_x is to be converted into molecular nitrogen.
- It is relatively easy to remove SO_2 from combustion gases by dissolving SO_2 in water and causing a reaction with alkali. Aqueous SO_2 quickly forms sulphurous acid, which reacts with the alkali and then is oxidised to sulphate. Collecting nitrogen oxides this way is not nearly as easy because NO, the principal nitrogen oxide present in combustion flue gas, has a very low solubility in water. Unlike SO_2, which quickly reacts with water to form acids, NO must undergo a two-step process to form an acid,

in which NO first reacts with oxygen to form NO_2, which then reacts with water to form HNO_3. The first reaction is relatively slow. It is fast enough in the atmosphere to lead to the formation of acid precipitation in the several hours or days that the polluted air travels before encountering precipitation. However, it is too slow to remove significant quantities of NO in the few seconds that a flue gas spends in a wet limestone scrubber used for SO_2 control. Some of the NO_2 in the flue gas is removed in such scrubbers, but normally only a small fraction of the total nitrogen oxides is NO_2 (<10%).

NO_x emissions can be controlled both by primary emission reduction measures, as shown in Section 5.5, and/or by secondary emission reduction measures. The secondary emission reduction measures involve chemical treatment of the flue gas after the combustion chamber aimed at converting NO_x to N_2.

The secondary NO_x emission reduction measures applicable for NO_x reduction in biomass combustion applications are mainly Selective Catalytic Reduction (SCR) or Selective Non-Catalytic Reduction (SNCR). Both utilise injection of a reducing agent, mainly ammonia or urea, to reduce NO_x to N_2, with or without a catalyst, respectively.

Selective Catalytic Reduction (SCR)

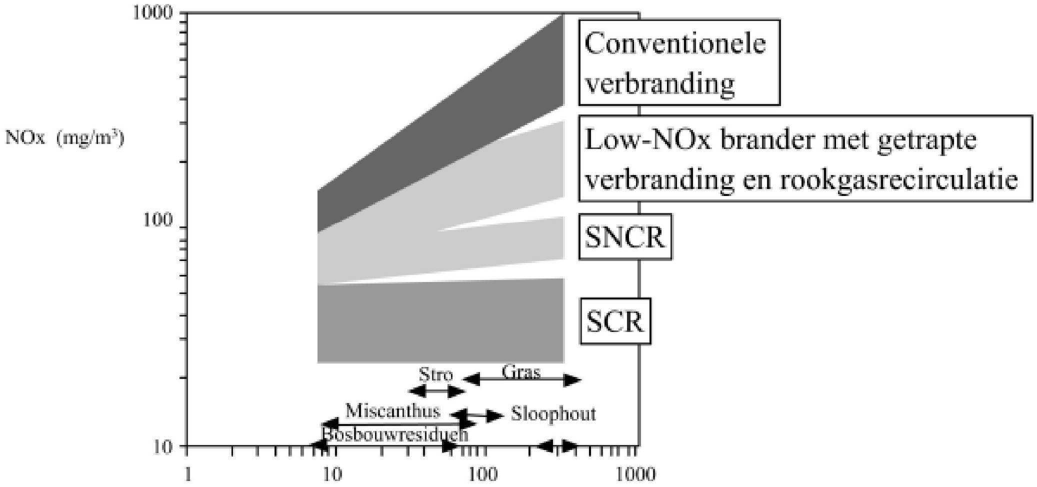
SCR reduces NO_x to N_2 by reactions with, usually, ammonia or urea in the presence of a platinum, titanium, or vanadium oxide catalyst. SCR operates optimally in a temperature range of 220-270^{o}C using ammonia, and 400-450^{o}C using urea, where a vaporised reducing agent is injected. Approximately an 80% NO_x reduction has been reported for SCR in fossil fuel combustion, where it is the most widely used secondary NO_x control technology. For wood fired systems up to 95% NO_x reduction at 250^{o}C without significant slippage of ammonia has been reported. A platinum-based catalyst is typically used. It is usually carried on an alumina support. The properties of the catalyst must be weighed against the properties of and the impurities in the fuel. Depending on the particle load, SCR is carried out prior to or after particle removal. For the SCR process, the long-term behaviour of the catalyst can be a problem, as deactivation is likely.

Selective Non-Catalytic Reduction (SNCR)

Because of the requirements and demands of catalysts, SNCR processes have been developed which do not require a catalyst for activation of the reaction. Instead, the reaction is run at higher temperatures. In the SNCR process, ammonia or urea is injected into the flue gas at a temperature usually between 850^{o}C and 950^{o}C. In a wood-firing system, a temperature between 840^{o}C and 920^{o}C proved optimal. Because of the high temperature, this process does not need a catalyst to initiate the reactions. Ammonia is injected at a rate of between 1:1 to 2:1 mole ammonia to mole of NO_x reduced. About 60-90% NO_x reduction can be reached with SNCR. The SNCR process requires an accurate temperature control to achieve optimum NO_x reduction conditions. If the temperature is too high, ammonia is oxidised to NO and if the temperature is too low, ammonia does not react at all and is emitted together with the NO_x. Hence, there exists an optimum temperature window for the SNCR process. Ammonia must be added to the flue gas in a quantity that is proportional to the NO_x content in the flue gas. Good mixing is very important to achieve optimum NO_x reduction conditions. Most SNCR processes have an ammonia slip of about 1 to 2 ppm in the flue gas leaving the stack.

In Figure 5.17 the NO_x reduction potential of SCR and SNCR is illustrated as a function of the fuel nitrogen content and compared to the NO_x reduction potential of staged-air combustion with a separate reduction chamber and also to conventional combustion.

Figure 5.17 Comparison of NO_x reduction potential for various NO_x reduction measures



NO_x reduction measures from top to bottom: Conventional combustion, Low-NO_x with air staging and reduction chamber, SNCR, SCR. Fuels from top to bottom: Grass, Straw, Urban waste wood, Miscanthus and native wood.

5.7 Emission limits

Maximum emission limits allowed for biomass combustion plants vary significantly from country to country, and are expressed in various units, making companions difficult. A recently published overview presents emission limits for CHP plants based on biomass combustion in Austria, Finland, Belgium, Denmark and Sweden (see Table 5.7). The parameters for which emission limits are defined include dust, CO, NO_x, SO_x, total organic carbon (TOC) and polychlorinated dibenzo-p-dioxins and dibenzofurans (PCDD/F). Not all 5 countries have defined emission limit for all parameters mentioned. The Finnish emission limits are expressed in mg/MJ and were re-calculated to mg/Nm^3.

The maximum emission limits have a large influence on the technology selection and system configuration.

Table 5.7 Overview of the emission limits for biomass combustion CHP plants

Country		**Austria**	**Finland$_{ex.}$**	**Finland$_{new}$**	**Belgium**	**Denmark**	**Denmark**	**Sweden**
Fuel		**W**	**W, S, P**	**BM / peat**	**BM**	**W, WW / S**	**BM$_{ex.}$/BM$_{new}$**	**BM**
Parameter	**FEI [MW]**	**mg/Nm$^3_{13\%O2}$**	**mg/Nm$^3_{13\%O2}$**	**mg/Nm$^3_{6\%O2}$**	**mg/Nm$^3_{11\%O2}$**	**mg/Nm$^3_{10\%O2}$**	**mg/Nm$^3_{6\%O2}$**	**mg/Nm$^3_{6\%O2}$**
Dust	< 0.12	150	-	-	150	-	-	350
	0.12 - 1	150	-	-	150	300[4]	-	100
	1 - 2	150	265	-	150	40	-	100
	2 - 5	50	265	-	150	40	-	100
	5 - 10	50	397[2]	-	30	40	-	100
	10 - 30	50	159[2]	-	30	40	-	35
	30 - 50	50	79[2]	-	30	40	-	35
	50 - 100	50	50[6]	50	10	-	100 / 50	35
	100 - 300	50	50[6]	30	10	-	100 / 30	30
	300 - 500	50	30[6]	30	10	-	100 / 30	30
	> 500	50	30[6]	30	10	-	50 / 30	30
CO	0.1 - 0.12	250	-	-	250	-	-	-
	0.12 - 1	250	-	-	250	500[4]	-	-
	1 - 5	250	-	-	250	500 / 625	-	250 / 500[7]
	5 - 50	100	-	-	200	500 / 625	-	250 / 500[7]
	> 50	100	-	-	100	-	-	250 / 500[7]
NO$_x$	< 0.1	-	-	-	r.c.o.	400 / -	-	-
	0.1 - 5	250	-	-	r.c.o.	400 / -	-	200 - 300
	5 - 10	250	-	-	r.c.o.	300	-	200 - 300
	> 10	200	-	-	r.c.o.	300	-	200
	> 100	200	199 / 238[3]	-	r.c.o.	300	-	200
	< 30	r.c.o.	-	-	400	r.c.o	-	200
	30 - 50	200	-	-	200	300	-	200
	50 - 100	200	373[6]	400	200	300	600 / 400	150 - 200
	100 - 300	200	373[6]	300	200	300	600 / 300	150 - 200
	300 - 500	200	124[6]	150	130	300	600 / 200	150 - 200
	> 500	200	124[6]	150	130	300	500[5] / 200	150 - 200
SO$_x$	< 50	-	-	-	300	-	-	-
	50 - 100	-	-	200 / 400	50	-	200	200
	> 100	-	-	200 / 200	50	-	200	200
TOC	> 0.1	50	-	-	-	-	-	-
PCDD/F[1]	> 5	-	-	-	0.1	-	-	-

Explanations: PCDD/F...dioxins and furans; FEI…fuel energy input; r.c.o….range covered otherwise; W…woody biomass; S…straw; P…peat; BM…biomass; WW…waste wood; ex….existing plant; new…new plant; TEQ…toxicity equivalent; 1)…expressed in ng TEQ/Nm^3; 2)…for plants built before 12 February 1987 to be calculated using emission limit [mg/MJ] = 85 – 4 x (FEI – 5) / 3; 3)…higher value valid for peat using burners, lower value valid for peat (other techniques) and wood or straw; 4)…valid for biomass; 5)…200 mg/Nm^3 from 1 January 2016; 6)…6% O$_2$; 7)…daily / hourly average.

6 CASE STUDIES

6.1 Introduction

In this chapter the following case studies are presented

1) Biomass fired cogeneration plant based on an ORC-cycle, Admont, Austria
2) Biomass fired tri-generation plant at Fischer/FACC, Ried im Innkreis, Austria
3) Straw fired neighbour heating plant, Søndre Nissum, Denmark
4) Wood fired cogeneration plant near log house factory, Karstula, Central Finland
5) Conversion of a district heating to a biomass CHP plant, Eksjö, South Sweden
6) Conversion of two coal boilers to biomass boiler, Jelenia Góra, Poland

The case studies were selected from a wide range of bioenergy projects that are reported on in database, websites, newsletters, reports, company brochures and other documents. Each case study represents an innovative element, be it of the system configuration (Admont and Ried), the organisational approach (Søndre Nissum), the type of combustion grate (Karstula), the electric power generation technology (Eksjö) or the project financing (Ried and Jelenia Góra).

6.2 Biomass CHP plant based on an ORC-cycle, Admont

6.2.1 Background

In 1999, a biomass-fired combined heat and power (CHP) plant based on an Organic Rankine Cycle (ORC) process, was implemented at the STIA timber processing factory in Admont, Styria, Austria. The objective of this project was to supply energy to both the timber processing factory and the local Benedictine monastery.

The STIA factory (see Figure 6.1) used to cover its process and space heat demand with one biomass-fired and two oil-fired furnaces. Three heavy heating oil-fired furnaces provided heat to the Benedictine monastery. As these old combustion units no longer complied with technical standards, STIA decided to replace them with a completely new biomass-based system.

Figure 6.1 STIA timber processing factory

The project was the first demonstration within the EU-15 of a biomass-fired plant based on the ORC process. Previously ORC processes had mainly been used in geothermal installations and not in biomass-based systems.

6.2.2 Description

The overall plant consists of two combustion units, one with a thermal oil boiler (nominal capacity 3.2 MW_{th}) and the other with a hot-water boiler (nominal capacity 4.0 MW_{th}). The plant uses sawdust and chemically untreated wood residues as fuel. Each furnace is followed by a dust precipitator combined with a common flue gas condensation unit. After the successful start-up of the new CHP plant the five old oil fired boilers were shut down, with two of them serving as stand-by units.

The ORC process (nominal electric capacity 400 kW, nominal thermal capacity 2.25 MW) is connected with the biomass-fuelled furnace via a thermal oil cycle and the thermal oil boiler (nominal capacity 3.2 MW; 0.95 MW of the thermal oil power are directly utilised via hot presses).

Figure 6.2 Working principle of the biomass fired ORC process

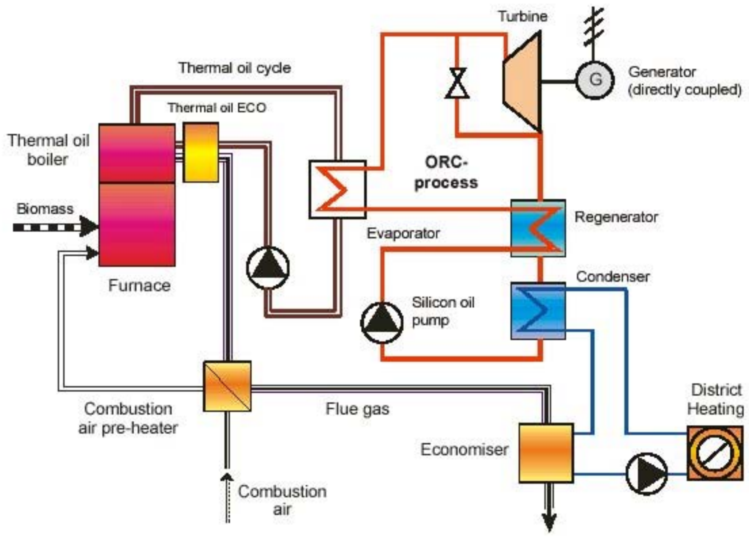
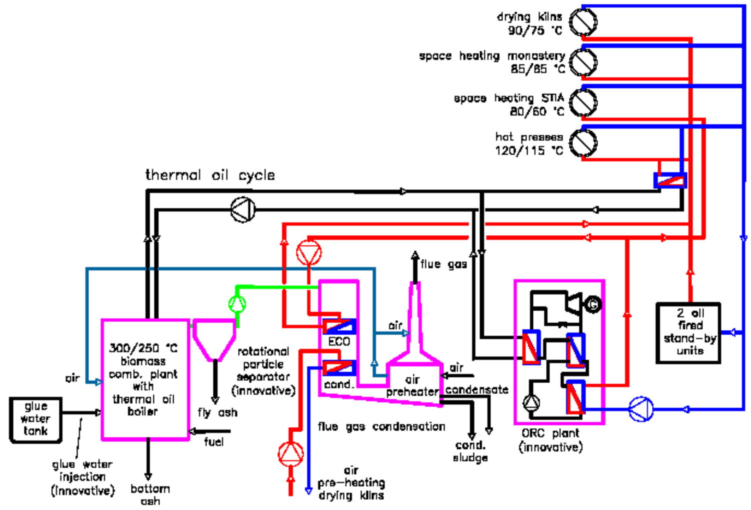


Figure 6.2 outlines the working principle of the ORC cycle. The ORC unit itself operates as a completely closed process utilising environmentally friendly silicon oil as organic working medium. This pressurised organic working medium is vaporised by the thermal oil in the evaporator and then expanded in a two-stage axial turbine that is directly connected to an asynchronous generator. Subsequently, the expanded silicon oil passes through a regenerator (where cycle-internal heat recuperation takes place) before it enters the condenser. There condensation of the working medium takes place at a temperature level which allows the heat recovered to be utilised (hot water feed temperature about 80 to 90°C). The liquid working medium then passes the feed pumps to achieve the appropriate pressure level of the hot end of the cycle again.

The thermal oil boiler covers the basic thermal load of the plant. If the heat consumption exceeds its capacity a standard hot-water boiler is brought into operation. From the new biomass-fired unit, a 470m district heating system supplies heat to the Benedictine monastery and the process heat consumers at STIA. The short pipe network (about 470 m only) guarantees low heat losses and a very high installed thermal power per meter pipe network.

One innovative aspect of the biomass CHP plant is the choice of a new, environmentally friendly and suitable organic working medium necessitated by the higher temperature levels on the hot and cold side of the ORC process as compared to electricity-only applications (as applied in geothermal applications). A further innovative aspect is the first demonstration of docking an ORC plant to a biomass-fired furnace via a thermal oil cycle. Further innovations (a rotational particle separator for particle precipitation and glue water injection directly into the furnace for NO_x reduction) had to be replaced with conventional solutions (a multicyclone for particle control and glue water addition directly to the fuel) due to operational problems that could not get solved.

Figure 6.3 Process flowsheet of the overall biomass CHP plant at STIA, Admont



6.2.3 Technical characteristics

Biomass input [tonnes]	5,000
Nominal capacity of the	3.2
Nominal capacity of the hot-water boiler [MW_{th}]	4.0
Nominal electric capacity of the ORC process [MW_e]	0.4
Nominal thermal capacity of the ORC process [MW_{th}]	2.25
Auxiliary electricity consumption [W/kW]	10-13
Thermal efficiency of the thermal oil boiler [%]	70-75
Thermal efficiency of the hot-water boiler	89%
Thermal efficiency of the ORC process	80%
Electrical efficiency of the ORC process	18%
Overall thermal efficiency of the plant	98%
Thermal and electrical losses	2%

6.2.4 Promoters and parties involved

- STIA: promoter, co-ordinator and owner of the project
- BIOS BIOENERGIESYSTEME GmbH: planning, engineering and monitoring
- Kohlbach: supply of combustion unit, flue gas condensation unit, and rotational particle separator
- Turboden: supplier of the ORC plant

Figure 6.4 Delivery of the ORC plant

Left hand: evaporator; right hand: regenerator; center in front: turbine & generator center back: condenser

6.2.5 Financial issues

The total investment costs for the biomass-fired CHP plant (excluding the hot-water boiler system) came to about €3,200,000 including monitoring and dissemination costs. The project was supported by a national grant of € 890,000 through the Austrian Kommunalkredit AG. The European Commission granted € 576,991. The remainder of the project costs was financed by own capital and bank loans.

The annual operation and maintenance costs amount to € 381,000, as specified below. The cost breakdown highlights one of the biggest advantages of ORC technology: its low maintenance demand and manpower need.

biomass fuel costs (€/year)	256,000 (67%)
personnel costs (€/year)	34,000
maintenance costs (€/year)	50,000
costs for auxiliary power demand (€/year)	26,000
other costs (administration, insurance, etc) (€/year)	10,000

The revenues of the project consist of heat sales to the STIA wood processing factory and to the monastery and electricity supply to the local utility. The payback period for the project is calculated at about 7 years.

6.2.6 Results

The performance of the plant was constant over the whole first year of operation. The plant was only shut down for a few days in summer for maintenance and due to minor problems related to incorrect measurement values on the instrumentation. After installation and testing, two original components of the system - a glue water injection system and a rotational particle separator - had to be removed due to operational problems that proved impossible to solve.

The total net electricity production came to over 1,900 MWh in 2001 (4,750 full load hours of operation). The net electrical efficiency of about 18% could be maintained at partial load operation, which is very important for CHP plants in heat-controlled operation. The thermal energy output was equivalent to approximately 10,000 MWh.

Revenues from heat and electricity sales total about €620,000 and €210,000 a year, respectively.

The new CHP plant substitutes fossil fuels at the Benedictine monastery and STIA wood-processing factory as well as electricity from fossil fuels by replacing the original five oil-fired combustion units. The oil-fired units at STIA are now only used as a stand-by. Furthermore, the new installation implies lower gaseous and particulate emissions, thus contributing to climate change mitigation and improvement of the air quality in the region. The emission reductions are about 68% CO_2 (2,800 toe/year), 86% SO_2 (15 toe/year), 48% NO_X (11 toe/year), 44% Total Organic Compounds (4 toe/year), 77% CO (21 toe/year), and 75% dust (10 toe/year).

6.2.7 Potential for replication

The results of the project have become a new technical standard for biomass-fired CHP systems in the 0.3-1.2 MW_{el} capacity range. In 2002, a new larger biomass district heating plant (1 MW_{el}) came into operation in Lienz as a follow-up EU demonstration project. Additionally, four more biomass-fired CHP plants based on an ORC process have been begun. The project serves as a model for decentralised biomass combustion systems in the timber processing industry as well as regional biomass district heating systems with an appropriate demand for process/district heat with a view to producing electricity for in-house use or for grid supply.

6.2.8 For more information

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6.2.9 References

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- Biomass fired CHP plant based on an ORC cycle - Project ORC-STIA-Admont. Final publishable report, STIA, Admont, March 2001

6.3 Biomass tri-generation plant at Fischer/FACC, Ried

6.3.1 Background

The company Fischer GmbH was founded in 1924 in Ried im Innkreis, Austria, and stands today as one of the premier sports brands in international skiing and tennis. The subsidiary company FACC (Fischer Advanced Composite Components) was founded in 1989 and is a leading company in Austria for the development, fabrication and maintenance of components for the aviation industry. The main business fields are structural aircraft components, engine fairings and aircraft interiors.

For about 30 years, the process heat demand of the high-tech Fischer production plant in Ried had been met by a 8.3 MW steam boiler in conjunction with a 4 MW back-up boiler, fuelled with 3.000 tpy of heavy oil, resulting in CO_2-emissions of about 9,456 tpy. This system was cheap but environmentally unfriendly. When it came to replace the old system, the company decided to meet its energy needs using biomass, rather than converting to the use of natural gas, in order to reduce CO_2 emissions.

The biomass tri-generation plant at the Fischer factory came into operation in Spring 2001. The new installation generates electricity, heat, and cooling for the manufacturing process of the two production facilities. Furthermore, it provides space heating and air-conditioning for the production hall and the offices. Fischer was the first factory worldwide to cover its heating and cooling needs with such an innovative biomass installation

The innovative technology was combined with a modern financing scheme. This solution was provided under a 15-year energy contracting arrangement signed with local consulting firm Scharoplan in June 1999. It comprises targets such as the use of renewable energy in a high-tech industry, high security of supply and safety levels, reduction of regional emissions and contribution to the Kyoto commitments, and creation of socio-economic benefits for the region.

The most important problem during the construction phase was that the production process of the company Fischer/FACC could not be hindered or interrupted. Therefore, the work needed had to be carried out at weekends or during periods when the production process was shut down. Careful planning and close co-operation between Scharoplan and the technical management of Fischer made it possible to avoid significant interruptions occurring.

The project (biomass tri-generation plant plus modern financing scheme) has been awarded the Energy Globe Award 2001 as an excellent example how a high-tech factory can fulfil its high quality energy needs with a biomass plant.

6.3.2 Plant description

The Fischer biomass tri-generation plant is fed by wood fuel, in particular bark, wood chips and sawmill wastes. Sufficient supply of biomass is ensured by the sawmill industry and the agricultural and forestry sector. About 180 m^3 of wood needs to be transported to the plant daily. To ensure an environmentally friendly fuel transport, a new railway branch line for biomass delivery was constructed on the premises of Fischer and FACC. A 30m long railway carriage in special design was constructed to transport part of the fuel demand (20,000 m^3/year, about 1/3 of total biomass use). The remainder is transported by road.

Figure 6.5 Unloading fuel for the biomass plant

The biomass is fed into a combustion chamber. By means of a heat exchanger, water is heated to a temperature of 400 ^{0}C at a pressure of 29 bar. The vapour then expands in a turbine, which is connected to an electrical generator. The water is condensed and transported back to the boiler. The remaining heat from the biomass boiler is used to provide process cooling and heating. Hot water at a temperature of 150 ^{0}C goes into an absorption-refrigeration unit, where it generates process cooling of 18-22 ^{0}C for the ski production unit. Process heat is delivered to the production unit at a temperature of 130 ^{0}C. Process heat at 250 ^{0}C and 290 ^{0}C is supplied to two different manufacturing processes for aeroplane components. The heat produced is also used for space heating (100 ^{0}C).

Figure 6.6 Assembly of the biomass plant

Figure 6.7 Completed biomass plant

6.3.3 Technical characteristics

Biomass input, bulk volume [m^3/year]	50,000-60,000
Steam production [t/h] and temperature [^{0}C]	10 / 380
Maximum operating pressure [bar]	32 (safety valve)
Boiler rating [kW]	7700
Fuel heating rating [kW]	9625
Combustion chamber temperature [^{0}C]	850-900
Electric capacity [kW_{el}]	915
Cooling capacity of the absorption refrigerating machine [kW]:	900
Turbine steam release pressure[bar]	4-6
Heat rating of the heating condenser and condensing refrigeration [kW]	6200
Condenser pressure [bar]	4-6

6.3.4 Promoters and parties involved

- Fischer/FACC: initiator
- Scharoplan: owner of the project, consultancy; planning, arranged funding (calling for tenders) and gathered all necessary planning authorisations for this project.
- Urbas GmbH: equipment supply

6.3.5 Financial resources

The total investment came to about € 5,000,000. Scharoplan financed 65% of this sum. The remaining investment costs were financed by subsidies from the EU, the national government and the federal state of Upper Austria.

The financing of the plant was arranged via a third-party financing scheme, for which Sharoplan is responsible under an energy supply contract with Fischer. The payback period for the project is estimated at 15 years.

6.3.6 Results

Technical	The annual output of the biomass tri-generation plant totals 26,000 MWh of heat, 1,000 MWh of cooling, 2,000 MWh of power, and 1,500 MWh of thermal oil.
Environmental	By using biomass to cover its total energy needs, Fischer substitutes 2,553 tonnes of heavy heating oil per year. The project reduces CO_2 emissions by 9,456 tonnes per year. It also contributes to a decrease in the amounts of other polluting emissions in the region.
Financial	The total value of the plant output is about €1,100,000 a year, of which 77% is heat production. The electricity generated each year has a value of €170,000.
Socio-economic	The socio-economic benefits of the project include the creation and preservation of jobs (about 100,000 person-hours during the planning and construction phases, and 3 person-years for the operation of the plant). Furthermore, the project generates additional income for the agricultural and forestry sectors. Both sectors receive about € 370,000 each per year from the sales of their waste by-products. The project also contributes to reducing the dependency on imported oil.

6.3.7 Potential for replication

For a successful replication of this project at another site, the supply of the necessary biomass needs to be guaranteed. Furthermore, the project must comply with the conditions necessary for it to receive a subsidy.

6.3.8 For more information

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6.4 Straw fired neighbour heating plant, Søndre Nissum

6.4.1 Background

About 70% of district heat in Denmark is produced by co-generation plants. Some smaller DH plant produce heat only and for this type of facility the trend is towards building ever smaller biomass-fuelled plant. On average, new DH plant from the 1990s are only half the size of plant from the 1980s. However, in Denmark, the market for DH plant is nearly saturated and at the moment neighbour heating is increasing in areas where conventional DH is too costly.

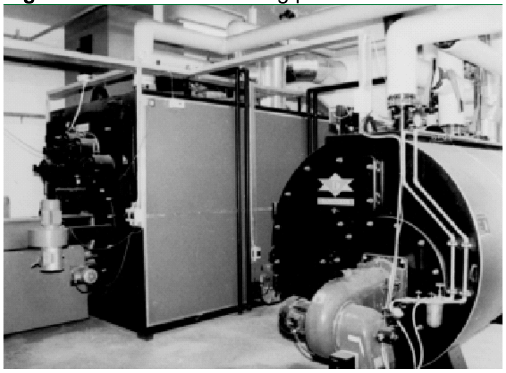
An example in case is the village of Søndre Nissum where the local authorities considered the implementation of a DH plant in the nineties. The cost would come to DKK 10 million (€1.35 m), which was considered prohibitive for a community of only 130 households, and the entire project was abandoned.

Henry Toft, a local farmer, then got the idea of setting up his own district heating grid using the straw from his 430 hectares of land. He first investigated the possibility of selling the straw to existing straw-fired district heating plants, but realised soon that the price was comparatively low. He therefore decided to build his own heating plant. Initially the farmer did not plan to supply heat to private households but instead to deliver all heat to a single large customer; the Ulfborg-Vemb Municipality, which had already shown itself favourable to the project and willing to buy district heat. When news spread about an agreement with the local authorities he was approached by a number of private individuals who were interested in participating in the project and buying heat from the planned straw-fired plant. Within a few weeks, 56 households had signed up and as a result of the local interest Toft bought a plant with twice the capacity initially planned. Today customers included >70 households, the local primary school, a sport centre and a rest home for elderly people

The differences between an actual DH plant and a neighbour heating plant larger than 250 kW are mainly size and type of ownership. A DH plant is typically larger than 1 MW and organised in the form of a private limited liability company or is publicly owned, e.g. by the local authorities. Neighbour heating plant are typically owned by farmers or established as a partnership with only one or two partners.

Figure 6.8 The boiler house in Søndre Nissum

Figure 6.9 Straw-fired heating plant



6.4.2 Description

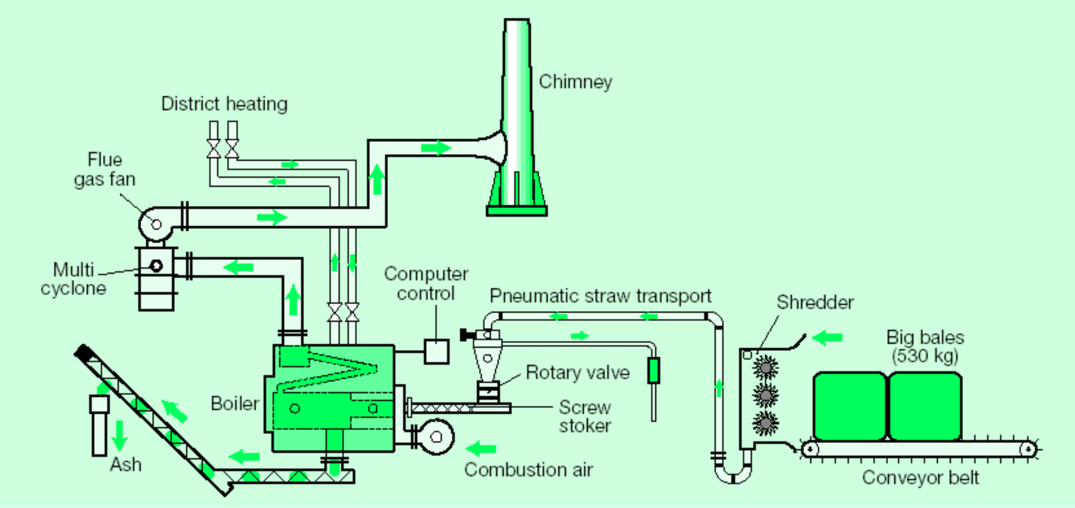
The straw-fired heating plant in Søndre Nissum has a capacity of 800 kW and is designed to supply 70–80% of the heat required for the coldest winter period. The size ensures optimum heating economy in the winter as well as in the summer when the heat production required is for domestic hot water only. During the coldest part of the winter, the system is supplemented by an oil-fired boiler, which also supplies heat when the straw-fired boiler is off-line for maintenance.

The process is straightforward (see Figure 6.10). Mr Toft moves big straw bales (530 kg each) on a conveyor belt once or twice a day. The bales are automatically transported to a straw shredder and then pneumatically into a feeding cyclone. From the cyclone the straw is fed directly to the boiler. The heat is piped via the heating network to its customers.

6.4.3 Promoters and parties involved

- Owner: Søndre Nissum Fjernvarme
- Main contractor: Lin-Ka Energy A/S
- Biomass expert: Danish Technological Institute, Aarhus
- Financial support: Danish Energy Agency

Figure 6.10 Diagram of the straw-fired neighbour heating plant in Søndre Nissum



6.4.4 Financial resources

Total investment	The total investment in the heating plant and infrastructure was about DKK 5,500,000 (€740,000), with cost of the heating plant, pipework and straw storage barn amounting to DKK 4,700,000 (approx. €630,000) and the total cost of the individual consumers' heating installations to DKK 800,000 (approx. €110,000).
Financing	The Danish Energy Authority provided a grant of about DKK 1.3 m (€175,000). The remainder was financed through a 10-year loan from a local bank. Each consumer made a token payment of DKK 600 to be connected to the heating plant and signed identical agreements for a 10-year period, which is the same as the depreciation period.
Source of revenue	The heating plant delivers heat to the local district heating grid.
Payback period	Calculated by the project initiator at ten years.

Construction of facilities and infrastructure was kept to the minimum (only a straw storage barn and a boiler house were built) and no expenses were paid for consulting and building supervision. As a result, investment costs were about half of the abandoned DH project.

Heat delivered from the neighbour plant is sold at DKK 485 (ca. €65) per MWh. Private consumers typically save DKK 2000 (ca. €270) per year by replacing old based heating with straw based heating.

6.4.5 Results

Initially Toft had 34 detached houses connected to the grid, together with the municipal buildings such as nursing home, school and the local sports hall. The price of heat from the straw-fired plant has been low and local residents have been joining the grid so that currently there are 75 detached houses connected to the heating network.

Energy production, fuel consumption	The annual energy production was 2 GWh from October 1999 to October 20001. During October 1999 - October 2000, straw consumption totalled 900 bales of 530 kg each, plus 1500 litres of heating oil.
Financial results	The cost of operating the plant is low as there are no employees. The good economics of the project have resulted in low heating prices (DKK 485/MWh) for customers.
Socio-economic impact	Use is made of an indigenous fuel. Mr Toft spends between 30-60 minutes a day operating the plant and his wife does the accounts.
Environmental benefits	The emissions of CO_2 from the power plant are the same as if the straw were to decompose naturally. If the heat were produced by an oil-fired boiler, approximately 200 tonnes of oil would be used each year. The annual carbon dioxide emissions from an equivalent oil-fired would be about 600 tonnes.

6.4.6 Potential for Replication

The low investment and the reduced operation and maintenance costs results in a low heating price and therefore the potential for similar small district heating plants is considered to be large.

6.4.7 For more information

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6.4.8 References

- Søndre Nissum 800 kW district heating plant. In: EU Renewable Energy: Best Practice Projects Yearbook 1997-2000, pg. 2.13-2.14
- Straw-based "Neighbour Heating" in Denmark. CADDET Renewable Energy Technical Brochure No. 151, March 2001.

6.5 Wood CHP plant at Honkarakenne Oy, Karstula

6.5.1 Background and motivation

Karstula is a town of 5,000 inhabitants in Central Finland, 100 km Northwest of Jyväskylä. The plant is located near to the log house factory Honkarakennne Oy. Honkarakenne produces 90,000 m^3 a year of sawn timber and logs for log houses.

The background to this project is:

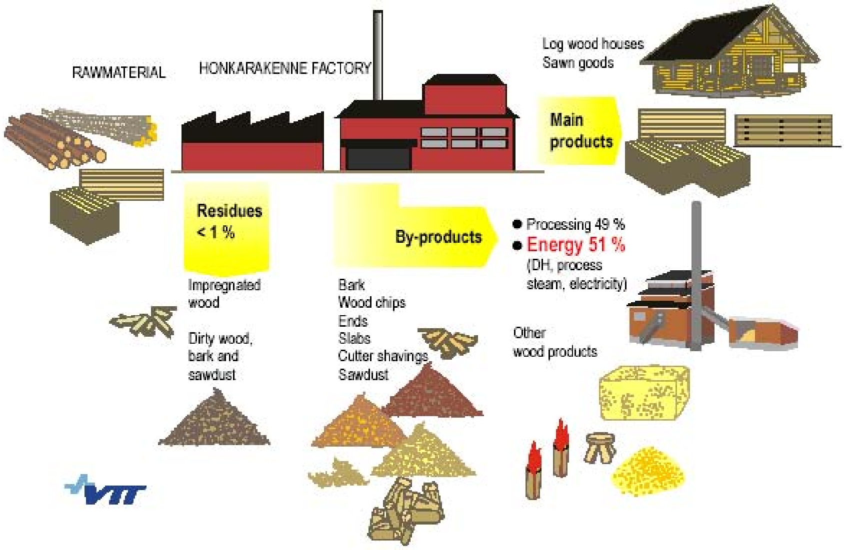
- Increased log house production at Honkarakenne Oy.
- Increasing energy demand for log drying and hot working.
- Possibility of using log-cabin factory's by-products, cutter shavings, bark and sawdust, as fuel.
- Extension of the district heating network in the town of Karstula.

Figure 6.11 Honkarakennne Oy. Karstula, Central Finland

6.5.2 Description

Honkarakenne Oy made considerable investments in developing and expanding its factories in Karstula. In order to meet increased energy demand (e.g. heat for wood dryers), the company decided to substitute old oil-fired boilers with a wood waste fired co-generation plant. The plant has a thermal capacity of 10 MW_{th} and produces 3 MW_{th} for the municipal district heating network, 3 MW_{th} process steam and 3 MW_{th} for Honkarakenne. In addition the plant produces 1 MW_e used in the wood processing company. The annual heat production is about 45 GWh, whereas the electricity production is about 7 GWh.

Figure 6.12 Wood raw material and wood fuel use at Honkarakenne Oy



Wood processing residues are used as fuel (see Figure 6.12). The factory produces 400 truckloads (70 GWh) of wood residues per year. The fuel is transported from the factory to the plant by wheel loaders to a storage with a capacity of 400 m^3. The measured average moisture content of the fuel (cutter shavings 44 %, bark 30 %, and sawdust 26 %) was 36 %. The actual moisture content depends on the combination of residues.

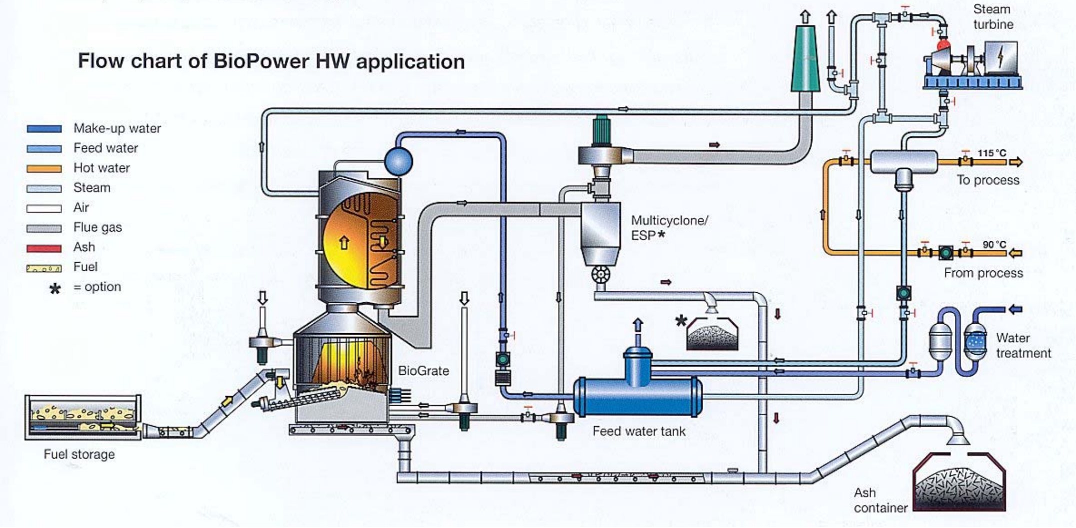
The biomass boiler was supplied by Wärtsilä Finland Oy (previously: Sermet Oy) and has a patented "BioGrate" combustion technique, capable of burning biomass fuels with moisture contents ranging from 30% to 65 %.

The BioGrate boiler (see Figure 6.13) is equipped with an underfeed rotating grate, which moves the fuel bed cyclically by hydraulics equipment. The movement of the grate is adjusted in such a way that the fuel is distributed as an even bed over the whole grate. The fuel dries and ignites on the grate. Main advantages of the rotating grate are: (a) there are no cold spots in the primary combustion chamber, (b) the burning surface is even, (c) movements of the grate zones are smooth, and (d) the secondary combustion chamber ensures complete combustion.

Figure 6.13 Wärtsilä BioGrate boiler

Electricity is produced by a generator driven by a steam engine. This modern design of the steam engine gives a high power to heat ratio when the heat load is matched and is therefore well suited to small electrical generating capacities. In this type of plant efficiency is high throughout the plant part load range and its operation is fully automatic and unmanned.

Figure 6.14 Flow chart Karstula wood fired co-generation plant



6.5.3 Technical data

Boiler	BioGrate, 10 MW
Steam values	22 bar, 350^{o}C, 12 t/h back pressure 0-1 bar
Electricity production per year	5 GWh
Heat production per year	45 GWh
Fuel used	cutter shavings, bark, and sawdust
Moisture content fuel	35-45 w-%

6.5.4 Promoters and parties involved

Wärtsilä Finland Oy supplied the plant. The owner of the plant is a new company called Puulaakson Energia Oy, which is jointly owned by Honkarakenne Oy, Keski-Suomen Valo (a power utility) and the Karstula town council.

6.5.5 Financial resources

Total investment	€ 4.540,000
Subsidy	Because new technology was applied, the Finnish Ministry of Trade and Industry contributed € 1 million investment support.

6.5.6 Results

Energy production	Estimated heat production will be 45 GWh and electricity production 5 GWh. The share of energy destined for the district heating network (11,000 MWh) does not meet demand, so the town of Karstula has its own district heat production as well.
Potential for replication	The large volume of wood residues produced by wood processing factories indicates that it should be possible to implement similar projects elsewhere.

6.5.7 Replication potential

The first Biograte CHP installation has been operating in Kiurevsei (Finland) since 1999. Since then, Wärtsilä sells each year a number of similar installations. More than 80 BioGrate plants are currently in operation in Europe (including Russia) and Canada.

6.5.8 For more information

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6.5.9 References

- CHP plant providing heat to a sawmill and town centre In: EU Renewable Energy: Best Practice Projects Yearbook 1997-2000, pg. 2.15-2.16
- A wood fuelled 10 MWth/1MWe CHP plant heating a wood processing factory and a town centre. Leaflet produced by OPET Finland in August 2000, Updated May 2002

6.6 Conversion of biomass DH to biomass CHP plant in Eksjö

6.6.1 Background

The municipality-owned energy company Eksjö Energi AB serves the South-Swedish town of Eksjö Energi AB with district heat from a single production plant.

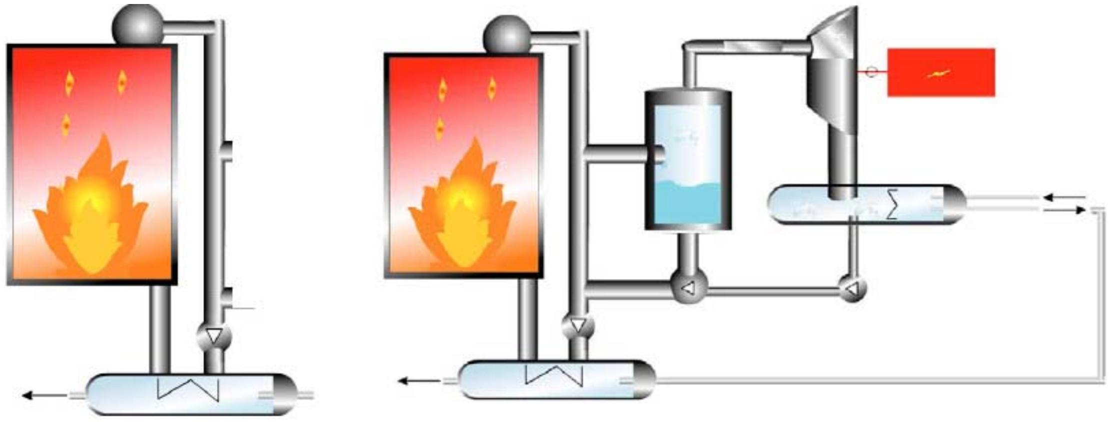
In the eighties, 4 boilers fired on biomass and waste were installed at the heat plant. In the early nineties Eksjö Energi developed an interest to produce its own electricity for the heat plant so that it could be self-sufficient and to reduce the cost of plant operation. The costs of building a new CHP plant were considered too high but in 1995 Dr Anders Kullendorf invented a process to produce power from a hot-water boiler. The power production was calculated to be about 10 % of the boiler effect. Eksjö Energi AB found the process interesting and adopted the process in 1996. After the Swedish Energy Authorities agreed to support the project the power generating equipment was installed in autumn 1997 by Vaporel AB. It has been under evaluation since spring 1998.

6.6.2 Description

The Eksjö plant is a conventional biomass and waste heating plant with a hot water boiler designed to operate at 16 bar, a common pressure in Swedish heating plants and sawmills. However, before its conversion the heat plant was operated at 6-7 bars only.

Electricity generation was made possible by increasing the pressure to about 14 bars and by converting some of the hot water into steam. From the existing boiler, hot water is piped to a flash-box, where a small amount is converted into steam and the pressure reduced. The steam generated in the flash-box releases some of its energy content during expansion in the Curtis turbine. The generator, which is connected directly to the turbine, converts the mechanical energy into electrical power. The role of the condenser is to cool down the exiting steam after the turbine to obtain additional electrical power. After having been heated, the cooling water is pumped to the municipal heating network, sawmill kilns or other users. The Eksjö plant now has an electric power generation capacity of 920 kW and produces 5,500 MWh of electricity annually. The electricity produced is used mainly for own purposes.

Figure 6.15 System sketch before and after rebuilding of the DH boiler according to the Eksjö concept



The innovation in the project is to connect a flash-box, a turbine and an existing hot-water boiler without replacement/reconstruction of the boiler, and to produce power from the hot water boiler. The advantage is a cheap and simple solution, The disadvantage is a limitation of power production.

Figure 6.16 Equipment before erection: flash-box, turbine and vacuum condenser respectively

The newly developed system was installed in the autumn of 1997 by the Swedish company Vaporel AB and has been under evaluation since the spring of 1998.

Figure 6.17 Equipment after erection: flash-box, turbine with generator and vacuum condenser respectively

6.6.3 Technical Characteristics

Production pressure/temperature to flash box [bar / ºC]	14.5 / 197
Pressure / temperature in the flash box [bar / ºC]	9.5 / 178
Steam production	
- Boiler type	Bubbling fluidised bed
- Boiler capacity [MW_{th}]	10
- Flash water flow [kg/sec]	107
- Steam generated [kg/sec]	3.8
- Percentage generated steam of inlet flow	3.4 %
Vacuum condenser	
- Pressure / Temperature [bar/ºC]	0.5 / 81
Electrical capacity [MW]	920
Annual electricity production [MWh/year]	5,500
Heat output (process of district heat) [MW_{th}]	Approx. 9,100

6.6.4 Promoters and parties involved

Eksjö Energi AB is the owner and operator of the plant. Vaporel AB developed the flash-box (steam generator) and was the main contractor.

6.6.5 Financial resources

Total investment	The total investment for the Eksjö plant's CHP conversion was 6.2 m SEK (approx. € 568,000). The specific investment costs for the Vaporel CHP-design were about 7,000 SEK (approx. €760) per installed $\mathrm{kW_e}$. The specific costs are lower for larger plant (capacity > 1 MW).
Subsidy	The Swedish government granted 1.6 m SEK (approx. €147,000) (a) because the technology was judged to be interesting for other district heating companies with hot-water boilers and (b) because of the technical risks involved
Source of revenue	The plant supplies hot water to the local heating grid.
Payback period	The pay back period is estimated at 5-6 years.

6.6.6 Results

Technical	• There were a few teething problems with electricity production during the first few months of operation. • The control system took some time to be implemented. • After Jan. 1999 the plant has been operating without any major difficulties, and the operational availability has been about 99%.
Environmental	The implementation of the steam generation the heating plant makes the plant self sufficient in electricity and thus contributes to reducing demand for electricity generation from other sources.
Financial	An economic analysis of the project has shown that the low investment and operating cost, due to the fact that it is providing electricity for own use, also makes this design profitable without a subsidy.
Socio-economic	The need to supervise the system has increased as a result of converting it from a heating plant to a CHP plant, which could therefore generate new employment.

6.6.7 Potential for replication

There are quite a few similar hot water boilers in Sweden, which could readily be converted to enable electricity generation as the design pressure of the boiler is higher than that needed for heat production. The low investment cost also makes this idea interesting for sawmills with drying processes.

6.6.8 For more information

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6.6.9 References

- Converting a heating plant to CHP plant at low cost. In: EU Renewable Energy: Best Practice Projects Yearbook 1997-2000, pg. 2.65-2.66
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6.7 Boiler retrofitting at Jelenia Góra greenhouse

6.7.1 Background

The Municipal Company of Communal Management of Jelenia Góra (MPGK) owns a small greenhouse complex where flowers and plants are cultivated. Before the retrofit, the heat demand of nine greenhouses and two adjacent offices was mainly covered by two brown coal boilers with a total capacity 640 kW_{th}.

However, heating with coal boilers had many disadvantages:

- it increases the greenhouse effect because it is a fossil fuel;
- the coal fired boilers have a very low rate of efficiency (less than 50%);
- the purchase of hard coal contributed significantly to the greenhouse running costs;
- high levels of environmental pollution from of dust, CO and SO_x.

Simultaneously, every year MPGK collects some 1100 m^3 of residues from the green spaces in maintains in Jelenia Góra and the neighbouring towns of Cieplice Zdroj and Sobieszowo. The green residues were dumped at a landfill, mulched or combusted in obsolete coal-fired boilers. It was calculated that at least 700 m^3 of green resides could be recovered immediately, and more than 3.5 times that amount in the near future, representing an energy potential of 2.100 and 7.500 GJ respectively, and sufficient to run a boiler of 350 kW_{th} and 1.250 kW_{th} respectively.

Figure 6.18 The boiler house in Jelenia Góra

Figure 6.19 A greenhouse in Jelenia Góra

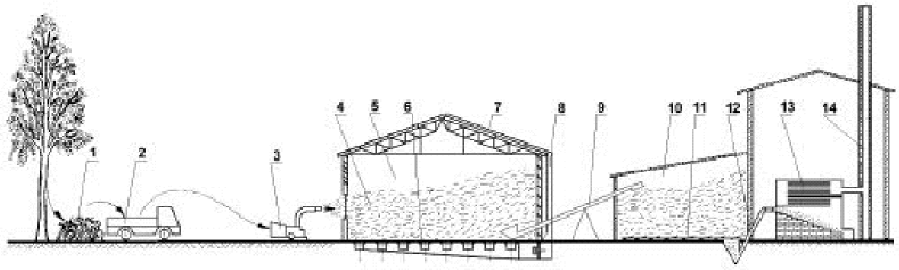
The idea was born to replace the boilers with an automatic, highly efficient installation for combusting waste wood. Project execution began in 1998. The boiler plant was officially inaugurated in October 2000. The investment was realised within the framework of the pilot phase of the Joint Implementation (JI) programme between Poland and The Netherlands. It was the first JI project in Poland.

6.7.2 Description

In the greenhouse complex described the two coal boilers with a total capacity 640 kW_{th} were replaced with a 350 kW_{th} woodchip fired automatic boiler. The plant provides heat to the greenhouse complex with a total area of 1200 m^2.

The plant pays 1.25 €/m^3 for waste wood from maintenance and is in charge of transport from the production site to the boiler plant. At the boiler plant site, the wood waste is shredded and stored for a couple of months in a long-term storage. The storage shed has a capacity of 1000 m^3 and is equipped with a drying system (air collectors connected with under-floor drying channels). From the long-term storage chips are fed pneumatically to the short-term storage located next to the heating plant. The latter has a capacity of 50 m^3 and is equipped with a fuel feeding system that consists of a screw conveyor and a walking-floor conveyor, which automatically feed the chips to the boiler.

Figure 6.20 Fuel supply chain and boiler plant



1- waste wood from tree maintenance, 2- transport vehicle, 3- wood chopper, 4- wood chips, 5- long term store, 6- floor channel dryer, 7,8- solar air collector, 9-screw conveyor, 10 - short term store, 11- walking floor, 12- screw conveyors, 13- KARA boiler, 14 - chimney

Depending on the humidity of the wood chips and the ambient temperature, the short-term storage is sufficient for 24-48 hours of boiler operation. Figure 6.20 illustrates the fuel supply chain and the combustion plant.

6.7.3 Promoters and parties involved

The partners in this joint venture project were:

Polish partners

- MPGK): beneficiary and local investor.
- Municipality of Jelenia Góra: project promotion
- EC Baltic Renewable Energy Centre: local co-ordination
- National Fund for Environmental Protection and Water Management, Executive Office for the Climate Convention, Secretariat-JI: overall responsibility for review of JI projects, co-ordination and reporting to the UNFCCC Secretariat,
- Polish Ministry of Environment: matters relating to JI activities in Poland

Dutch partners

- Biomass Technology Group: co-ordination of the Dutch JI project
- KARA Energy Systems: equipment supply
- SENTER / Netherlands Ministry of Economic Affairs: donor agency

6.7.4 Financial resources

The Dutch government financed the purchase of the boiler, preparatory and planning work, engineering, feasibility study and the monitoring system (total value about €207,500 as detailed below, assuming 1€ = 4PLN). The Polish investor financed construction costs, the building and technical equipment e.g. floor drying channels, valued at approx. €135,000.

Component	**€**
boiler with control equipment	107,500
stack (including assembly)	2,500
“moving floor” feeder	11,250
additional grid connection	17,500
storehouse for chips incl. drying system located under the floor	25,000
air solar collector	7,500
shredder	12,500
truck loader, roads and squares	12,500
boiler installation (mounting and crane)	6,250
documentation	5,000
Total expenses	**207,500**

Carbon credits from the project were shared between Poland and The Netherlands, as follows: 45% Poland, 55% the Netherlands.

6.7.5 Results

Technical

Before modernisation heating system for greenhouses in Jelenia Góra had the following parameters:

- total surplus of 2540 m^3 waste wood in the city
- at least 700 m^3 of wood waste deposited in dump
- 2 coal fired boilers, 256 kW each used
- yearly coal consumption 220 tonnes (@ EUR 90 per tonne)

After modernisation, the parameters were as follows:

- annual quantity of wood used for heating purposes (not dumped): 388 tonne. (1100 m^3, moisture content 55%)
- 1 automatic wood chips fired boiler, 350 kW
- no coal consumption

Environmental

- Significant direct emission reductions: CO_2 4.55 tpy, SO_2 2.58 tpy, NO_x 0.039 tpy, Dust 1.37 tpy (where tpy = tonne per year).
- Avoided methane emission reduction from anaerobic digestion of previously dumped green residues, calculated at 23 tpy.
- Total emission reduction over the project lifetime (ten years) is calculated at 15,500 CO_2-equivalents

Financial

Annual savings include:

- EUR 19,800 in hard coal procurement (220 tpy @ € 90)
- EUR 16,600 in avoided waste wood disposal costs (388 tpy @ € 43)
- Avoided environmental charges and penalties

Socio-economic

- Use of a local energy source: waste wood from urban green areas which otherwise had to be disposed off.
- Use of ash (combustion residue) for fertilising city green areas (returning part of the organic waste to the ecosystem)
- Eliminating transport of coal to the heating plant and wood waste transport to the landfill
- Know-how transfer (the establishment of a biomass energy centre was included in the project)

6.7.6 Potential for replication

The project has been already a replication of a similar investment made in the town of Otwock near Warsaw where waste biomass from cuttings in parks and other green municipal areas was turned into a fuel in a modern heating installation. Apart from some technical innovations applied in the project in Jelenia Góra an interesting solution of the JI mechanism was used for financing the investment, which has prepared now the background and allowed testing some practices with using various methods for the calculations of baselines for next larger biomass JI projects to be implemented in Poland. In general, the project has been successful and can be replicated both from technical and financing point of view in other countries as well as inside Poland itself.

6.7.7 For more information

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ANNEXES

A. DEFINITIONS AND TERMINOLOGY

ACRONYMS AND ABBREVIATIONS

General

AC	Ash content (of biomass)
CHP	Combined Heat and Power (also known as co-generation)
CFB	Circulating fluid bed
DB	Dry basis
FC	Fixed carbon
GCV	Gross calorific value (equivalent to HHV, Higher heating value)
IC	Internal combustion
LCV	Lower calorific value
MC	Moisture content (of biomass)
NCV	Net calorific value (equivalent to the LHV, Lower heating value)
PAH	Polycyclic Aromatic Hydrocarbons
PPM	Parts per million
WB	Wet basis
ε	coefficient of performance, COP
λ	air factor
η	efficiency
φ	mass flow

Subscripts

ar	as received
b	with m^3: bulk volume (equal to the specific volume times [1-porosity])
dry	on a dry basis
daf	on a dry and ash-free basis
e	electric
m	with %: mass percent
s	with m^3: specific (or true) volume of solid material
th	thermal
tot	total
vol	with %: volume percent
w	on a wet basis
wt:	with %: weight percent

Units and prefixes

Length, Area, Volume

m	meter
m^2	square meter
m^3	cubic meter
Nm^3	cubic meter gas under normal conditions: 273.15 ^{0}C and 101325 Pa.
N_Em^3	similar as Nm^3, but including only 11vol% O_2, and 0% H_2O.

Weight

g	gram
kg	kilogram
kg_x	kilogram of matter at a reference moisture content (MC_w) of x $\%_m$
t	tonne (1000 kg)
t_x	tonne of matter at a reference moisture content (MC_w) of x $\%_m$

Time

d	day (24 hours)
h	hour
s	second
yr	year (365 days

Energy and Power

J	Joule
kWh	kilo-Watt-hour (3,600,000 J or 3.6 MJ)
Btu	British Thermal Unit, 1055.06 J
W	Watt (=J/s)

Temperature

^0C	degrees Celsius
K	degrees Kelvin, $^0C + 273.15$

Pressure

Pa	Pascal
bara	bar, absolute pressure, 10^5 Pa
barg	bar gauge, relative to ambient pressure, 10^5 Pa
atm	atmosphere, 101325 Pa

Prefixes

µ	micro	(10^{-6})
m	milli	(10^{-3})
c	centi	(10^{-2})
k	kilo	(10^{3})
M	Mega	(10^{6})
G	Giga	(10^{9})
T	Tera	$(10^{12)}$

Definitions

General

Air factor (λ)	Actual quantity of air per stoichiometric quantity of air for complete combustion of the fuel, volume or molar ratio (mole/mole)
Availability factor	Number of actual operating hours per total number of hours during operational period
Bulk density	Weight per unit volume of material as packed in a container.
Burnout	(or Carbon burnout) Degree of carbon removal by burning or gasifying the remainder of a solid fuel
Dryness fraction	Ratio of the mass of vapour to the total mass of a vapour/liquid mixture.
Equivalence ratio	Same as air factor (used in literature on gasification)
Excess air ratio	Air factor minus 1
Producer gas	Gas and vapours produced as the product of the gasification process.
Pyrolysis gas	Gas and vapours produced as a by-product of the carbonisation process
Specific density	Weight of material divided by the volume occupied by the solid material.

On material analysis

Ash content	The quantity of inert matter remaining after combustion in a prescribed manner (proximate analysis). It is formed from minerals present in the fuel and oxygen supplied during combustion
Charcoal	Carbonised biomass, mainly consisting of carbon and ash
Dried biomass	Biomass material on which a drying operation has been performed; moisture content is not specified
Dry matter	Weight of matter, excluding any water present
Fixed carbon	Carbon remaining after heating (see proximate analysis)
Proximate analysis	The determination, by given prescribed methods, of the moisture content (ISO 331), volatile matter content (ISO 562), ash content (ISO 1171) and fixed carbon content (ISO 609) of a fuel
Ultimate analysis	The determination, by given prescribed methods, of the elemental composition of a fuel
Moisture content	In this report defined on wet basis, unless indicated differently

On the calorific value

Gross calorific value	The amount of heat released per mass of fuel after its combustion in oxygen under standard conditions (a.o. constant volume); combustion products are the off-gas (containing oxygen, carbon dioxide, sulphur dioxide, nitrogen, water and other oxygenated compounds), liquid water in equilibrium with its vapour and saturated with carbon dioxide, and ash. Note that the GCV includes the heat of evaporation of nearly all water present in the product gas. Determination is prescribed in ISO 1928.
Net calorific value	The amount of heat released per mass of fuel after its combustion in oxygen under standard conditions (a.o. constant volume, all water present in the gas phase); combustion products are a.o. carbon dioxide, sulphur dioxide, water vapour and ash, all at 25°C. Note that the NCV does not include the heat of condensation of the water present in the combustion gas. Determination is prescribed in ISO 1928.

On efficiencies

Cold gas efficiency	The ratio between the NCV of the gas and the NCV of the original solid fuel
Hot gas efficiency	The cold gas efficiency plus the amount of sensible heat of the gas at reference conditions (20°C, 101325 Pa)
Gross efficiency	Ratio of energy output and fuel input, whereas the latter is based on GCV
Net efficiency	Ratio of energy output and fuel input, whereas the latter is based on NCV

B. POLLUTANTS FROM BIOMASS COMBUSTION AND THEIR IMPACTS.

Pollutants from biomass combustion and their impacts on climate, environment, & health.

Component	**Biomass sources**	**Climate, environmental and health impact**
Carbon dioxide (CO_2)	Major combustion product from all biomass fuels	**Climate:** Direct GHG. However, biomass is a CO_2-neutral fuel
Carbon monoxide (CO)	Incomplete combustion of all biomass fuels	**Climate:** Indirect GHG through ozone formation. **Health:** Reduced oxygen uptake especially influences people with asthma, and embryos. Suffocation in extreme cases.
Methane (CH_4)	Incomplete combustion of all biomass fuels	**Climate:** Direct GHG. Indirect GHG through ozone formation.
Non Methane Volatile Organic Components (NMVOC)	Incomplete combustion of all biomass fuels	**Climate:** Indirect GHG through ozone formation. **Health:** Negative effect on human respiratory system
Polycyclic Aromatic Hydrocarbons (PAH)	Incomplete combustion of all biomass fuels	**Environment:** Smog formation **Health:** Carcinogenic effects
Particles	Soot, char and condensed heavy hydrocarbons (tar) from incomplete combustion of all biomass fuels. Fly ash and salts	**Climate and environment:** Reversed greenhouse effect through aerosol formation. Indirect effects of heavy-metal concentrations in deposited particles. **Health:** Negative effect on the human respiratory system. Carcinogenic effects
Nitric oxides ($NO_x = NO$ and NO_2)	Minor combustion product from all biomass fuels containing nitrogen. Additional NO_x may be formed from nitrogen in the air under certain conditions	**Climate and environment:** Indirect greenhouse effect through ozone formation. Reversed greenhouse effect through aerosol formation. Acid precipitation. Vegetation damage. Smog formation. Corrosion and material damage. **Health:** Negative effect on the human respiratory system. NO_2 is toxic
Nitrous oxide (N_2O)	Minor combustion product from all biomass fuels containing nitrogen	**Climate:** Direct GHG. **Health:** Indirect effect through ozone depletion in the stratosphere
Ammonia (NH_3)	Small amounts may be emitted as a result of incomplete conversion of NH_3 from pyrolysis/ gasification	**Environment:** Acid precipitation. Vegetation damage. Corrosion and material damage. **Health:** Negative effect on the human respiratory system.
Sulphur oxides ($SO_x = SO_2$ and SO_3	Minor combustion product from all biomass fuels containing sulphur.	**Climate and environment:** Reversed greenhouse effect through aerosol formation. Acid precipitation. Vegetation damage. Smog formation. Corrosion and

		material damage. **Health:** Negative effect on the human respiratory system., asthmatic effect
Heavy metals	All biomass fuels contain heavy metals to some degree, which will remain in the ash or evaporate	**Health:** Accumulate in the food chain. Some are toxic and some have carcinogenic effects
(Ground level) Ozone (O_3)	Secondary combustion product from atmospheric reactions, including CO, CH_4, NMVOC and NO_x	**Climate and environment:** Direct GHG. Vegetation damage. Smog formation. Material damage. **Health:** Indirect effect through ozone depletion in the stratosphere. Negative effect on the human respiratory system, asthmatic effect
Hydrogen Chloride	Minor combustion product from all biomass fuels containing chlorine	**Environment:** Acid precipitation. Vegetation damage. Corrosion and material damage. **Health:** Negative effect on the human respiratory system. Toxic
Dioxins and Furans PCDD/PCDF	Small amounts may be emitted as a result of reactions including carbon, chlorine, and oxygen in the presence of catalysts (Cu)	**Health:** Highly toxic. Liver damage. Central nervous system damage. Reduced immunity defence. Accumulate in the food chain