KUNGL
TEKNISKA
HÖGSKOLAN

HPT/KTH **08/2002**

HYBRID COMBINED CYCLES WITH BIOMASS AND WASTE FIRED BOTTOMING CYCLE

- Literature Study

Miroslav P. Petrov

Literature Study Report
2002
SNEA Project P12477-1

Department of Energy Technology
Division of Heat and Power
Royal Institute of Technology
Stockholm, Sweden

KUNGL TEKNISKA HÖGSKOLAN

Literature Study Report
Dept. of Energy Technology
Div. of Heat and Power
Prof: Torsten H. Fransson

Title: Hybrid Combined Cycles with Biomass and Wastes Fired Bottoming Cycle. A Literature Study.	**Keywords:** Biomass, MSW, Topping and Bottoming Cycle, Gas Turbines, Internal Combustion Engines, Steam Boilers, Hybrid Combined Cycle.
Author: Miroslav P. Petrov	**Report nr:** KTH/HPT 08/2002

Project title: "Biomass and Natural Gas" SNEA P12477-1	**Pages:** 145	**Figures:** 65
Supervisors at KTH: Laszlo Hunyadi, Torsten H. Fransson, Jinyue Yan	**Date:** 2002-02-13	**Appendices:** 3

Overall responsible at KTH: Prof. Torsten H. Fransson

Approved at KTH by:	**Signature:**
Overall responsible at industry:	
Industrial partners:	
Approved by industrial partners:	**Signature:**

Abstract

Biomass is one of the main natural resources in Sweden. The present low-CO2 emission characteristics of the Swedish electricity production system (hydro and nuclear) can be retained only by expansion of biofuel applications for energy purposes. Domestic Swedish biomass resources are vast and renewable, but not infinite. They must be utilized as efficiently as possible, in order to make sure that they meet the conditions for sustainability in the future. Application of efficient power generation cycles at low costs is essential for meeting this challenge. This applies also to municipal solid waste incineration with energy extraction, which should be preferred to its dumping in landfills.
Hybrid dual-fuel combined cycle units are a simple and affordable way to increase the electric efficiency of biofuel energy utilization, without big investments, uncertainties or loss of reliability arising from complicated technologies. Configurations of such power cycles are very flexible and reliable. Their potential for high electric efficiency in condensing mode, high total efficiency in combined heat and power mode and unrivalled load flexibility is explored in this project.

The present report is a Literature Study that concentrates on certain biomass utilization technologies, in particular the design and performance of hybrid combined cycle power units of various configurations, with gas turbines and internal combustion engines as topping cycles. An overview of published literature and general development trends on the relevant topic is presented. The study is extended to encompass a short overview of biomass utilization as an energy source (focusing on Sweden), history of combined cycles development with reference especially to combined cycles with supplementary firing and coal-fired hybrid combined cycles, repowering of old steam units into hybrid ones and combined cycles for internal combustion engines. The hybrid combined cycle concept for municipal solid waste incinerators is probably the option with the greatest efficiency improvement potential, within the reasonable cost and scale limits.
Furthermore, a State-of-Art report is included in the study as a separate chapter. Descriptions of existing hybrid combined cycle installations with biofuel-fired bottoming cycle in Sweden and its surrounding countries are compiled in it. The presentation shows that hybrid combined cycles are a standard technology in many respects. These specific configurations have been chosen as the most rewarding ones out of various alternatives and have proved their advantages in commercial operation.

The major research project following this Literature Study will focus on investigation of possible efficiency improvement of biomass energy utilization by application of hybrid configurations with natural gas fired gas turbine and internal combustion engines as topping cycles.

(Svensk titel: Hybridcykler med gasturbiner/kolvmotorer och biobränsle- eller avfallseldade pannor.)

Approved for distribution:
❑ **Open**

Name/company	Address	Locality

CONTENTS:

ACKNOWLEDGEMENTS

The entire project, part of which is this Literature Study, is financially supported by the Swedish National Energy Administration (Statens Energimyndighet), within the programme framework Termiska Processer för Elproduktion. Their contribution is gratefully acknowledged.

I am extremely grateful to Prof. Torsten Fransson and Assoc. Prof. Laszlo Hunyadi at the Division of Heat and Power Technology in KTH, for the chance to work on this exciting project. Gratitude is also due to all my supervisors in the Advisory Group, whose help, remarks and suggestions cleared the way for this report and the whole project work.

Last but not least, I would like to thank my fellow students and other colleagues from the Division of Heat & Power Technology and the whole Department of Energy Technology at KTH, for their support and for the nice atmosphere.

ABBREVIATIONS

BC	Bottoming Cycle
BFB	Bubbling Fluidised Bed
CC	Combined Cycle
CFB	Circulating Fluidised Bed
CHP	Combined Heat & Power
DFCC	Dual-Fuel Combined Cycle
FB	Fluidized Bed
FD	Forced Draught (fan)
GT	Gas Turbine
GTCC	Gas Turbine Combined Cycle
ha	hectare (a unit for land area = 10’000 m^2)
HCC	Hybrid Combined Cycle
HFO	Heavy Fuel Oil
HHV	Higher Heating Value (of fuels)
HRSG	Heat Recovery Steam Generator
HP	High Pressure
ICE	Internal Combustion Engine
ICECC	Internal Combustion Engine Combined Cycle
ID	Induced Draught (fan)
IGCC	Integrated Gasification Combined Cycle
LFO	Light Fuel Oil
LHV	Lower Heating Value (of fuels)
LP	Low Pressure
MP	Medium Pressure
MSW	Municipal Solid Waste
NG	Natural Gas
NOx	Nitric Oxides (generalised formulation)
PFBC	Pressurised Fluidised Bed Combustion
PM	Particulate Matter
ppm	parts-per-million (= mg/Nm^3)
ppmv	parts-per-million on volume basis (= ml/Nm^3)
RDF	Refuse Derived Fuel
SOx	Sulphuric Oxides (generalised formulation)
ST	Steam Turbine
TC	Topping Cycle
UHC	Unburned Hydro Carbons
VOC	Volatile Organic Compounds
%wt	% by weight (mass percent)
%v	% by volume (volumetric percent)

1. INTRODUCTION

1.1. Background

Biomass has been used as fuel since the dawn of humankind. In the last two centuries however, it has been gradually replaced by fossil fuels, which are used in ever-rising quantities (although in many developing countries biomass is still the most important local energy resource). Industrialization, large-scale centralized power generation and production of fuels for transportation have ruled out the use of biomass as a major energy resource. It now contributes a very small fraction of the primary energy mix of the industrialized nations, although still being important for developing countries.

Currently, interest in biomass energy utilization is rising again. Several major factors for this can be mentioned: decreasing pollutant emissions (especially carbon dioxide), sustainable energy resource management and lowering dependence on imported fossil fuels. Due to the increasing necessity for CO_2-neutral power generation and sustainability of fuel supplies, increased use of biomass for energy purposes is inevitable if the world is going to switch away from fossil fuels.

Because of their inherently small sizes, biofuel-fired power cycles with steam boilers and steam turbines have comparatively low electric efficiency. High specific investment costs do not allow raised steam parameters in small scales, while fuel availability is limiting the possibilities for economy of scale.
There is plenty of room for electric efficiency enhancement of traditional biomass power units, either by raising steam parameters and applying modern steam turbines, or by alternative options, such as externally fired gas turbines for example.

One easy way to utilize biofuels with higher efficiencies, while keeping the process simple and reliable and using only standard equipment, is the incorporation of a biofuel-fired power cycle as bottoming cycle in a hybrid combined cycle (HCC) with a topping high-grade fuel fired heat engine. A small amount of fossil fuel in the topping cycle can provide higher efficiency of biofuel utilization in the bottoming cycle. To reach full sustainability and higher CO_2 emissions reduction, the topping cycle can be powered by biomass-derived high-grade fuels.

Natural gas is the least polluting of all fossil fuels, allowing power generation with very high efficiencies in modern gas turbines and internal combustion engines.
Natural gas and biofuel powered HCC units are a simple and affordable way to increase the electric efficiency of biofuel energy utilization, without big investments, uncertainties or loss of reliability arising from complicated technologies.
Configurations of such power cycles are very flexible and reliable, and provide high total efficiency in CHP mode.

1.2. Objectives

1.2.1. Objectives of the main research project

The main research project, titled "Biomass and Natural Gas", will focus on modelling, calculation and analysis of hybrid combined cycles with biomass-fired bottoming cycle. The topping cycle is fired with natural gas. Various arrangements and heat engines will be considered and evaluated in design mode and part-load operating conditions. The effect of topping cycle to bottoming cycle power ratio (natural gas to biofuel ratio) on the electrical efficiency of the various configurations will be investigated, together with its effect on the efficiency of biofuel utilization itself. The purpose is to prove that the efficiency of energy utilization of the bottoming fuel (biomass or wastes) will increase when the cycle is combined with high-grade fuel fired high-efficiency topping cycle. This increase in efficiency may not be very high, but can be achieved with simple technology, using standard equipment and showing excellent flexibility and part-load performance.

The different cycle configurations will be optimized for highest electrical efficiency, analysed and compared to each-other in terms of performance, economy, fields of application and difficulties in construction, operation and maintenance. The most promising cycles will be modelled also in CHP mode and promoted for practical installation as cogeneration units in Sweden and other countries.

1.2.2. Objectives of the Literature Study

The main purpose of this literature study is to serve as an introduction to the project that it precedes, along with providing the needed investigation of published scientific work that has been performed by others on the relevant topic. The author sincerely hopes that the present report will help to promote the general understanding, appreciation and knowledge on the importance of biomass and biofuels for sustainable energy production, facing the environmental challenges that are inherent to the economic growth of our human civilisation.

Among all types of biofuel energy utilization technologies, hybrid combined cycles are presented as probably the simplest and the most technically and economically viable option for widespread application. The present literature study will show that HCC's are based on well-developed and proven technology. Power units of this type are in operation since long time. All installed units have shown very good performance, availability and reliability.

1.3. Definitions

Some points should first be made on terminology issues. As with many other fields of science, the terminology of this engineering field is subject to some debate. On the contrary, technical terms denoting all details of the separate power cycles in the English language are well accepted and well spread world-wide. Consequently, no misinterpretations or misunderstandings are possible when talking about steam cycles, gas turbine cycles or pure gas and steam combined cycles.
However, this is not the case with the hybrid power cycles.

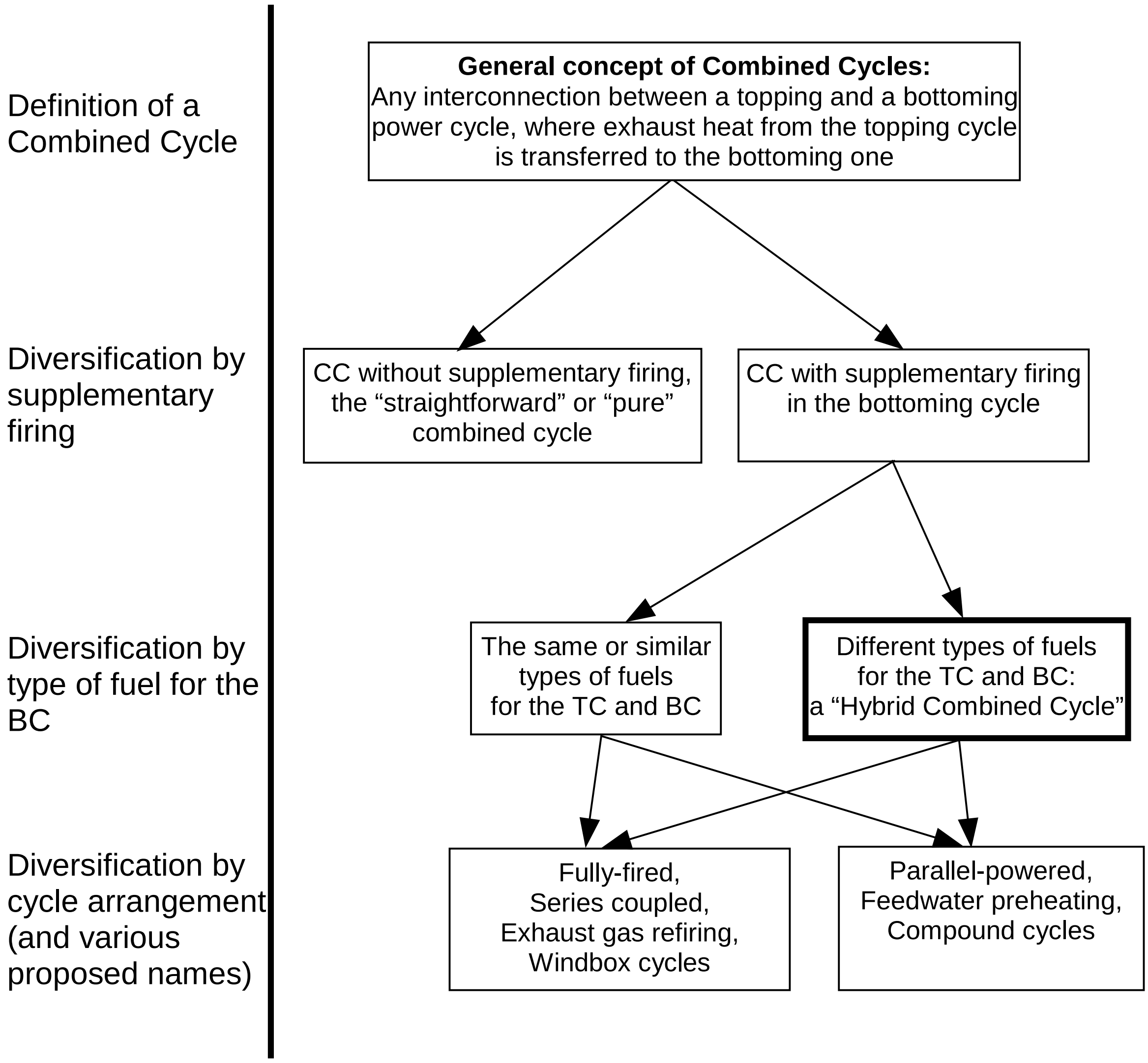
In the English literature, there seems to exist no standardised technical term for addressing the power cycles, which we call here "hybrid combined cycles" (HCC). Instead, terminology as "parallel-powered", "unfired" and "fully-fired" are mostly used. As long as these names feature different types of HCC and address certain properties of these types of cycles that differentiate them from each-other, they are widely used and will be used also in this report. The term "compound cycle" is also used for HCC, but usually denoting parallel-powered HCC (it probably comes from the German word "Verbundkraftwerk" [4.4]). The term "feedwater heating system", meaning exactly parallel-powered HCC, can also be met in the literature. Alternative name for fully-fired HCC is "exhaust gas refiring system" or "series coupled CC".
The term "supplementary firing" is also very popular and usually means additional firing of the same fuel in the HRSG, as in the gas turbine itself. Supplementary firing is often used in combined cycles for various reasons and can transform a pure (unfired) CC into a fired or fully-fired HCC.

The term "hybrid cycle" has slowly gained popularity, as it depicts in a generalised way the distinctive features of both parallel powered and fully-fired cycles and unifies them under one name. Its corresponding word in Swedish is "hybridcykel". As compared to the supplementary-fired cycles, the term "hybrid cycles" denotes specifically "dual-fuel" combined power cycles, which means combined cycles in which different fuels are used for the topping and bottoming cycle. This is one of the big advantages of the HCC – the possibility to utilize low-grade fuels (solid fuels) in the bottoming cycle, together with high-grade fuels (gaseous or liquid) on a high temperature level in the topping turbine or topping engine. Exactly this "dual-fuel" feature gives us the right to use the term "hybrid".

It must be pointed out that the word "hybrid" itself can actually be met in various technical applications, as it is very general and can be applied to different issues. The most common usage of the term "hybrid" in the technical literature is connected to the hybrid traction of vehicles – a combination of an engine (or a small gas turbine) with batteries and electric motor. In the field of power engineering, "hybrid" is usually used to denote any combination of two different power cycles (for example a fuel cell combined with a gas turbine), or a single engine able to utilize different fuels.

In this report and in the project that follows, the term "HCC" is used to address dual-fuel hybrid combined cycles, with high-grade fuel fired topping cycle and low-grade fuel fired bottoming cycle, as explained above.

In order to clarify further the term "hybrid combined cycle" and its position within the "family" of combined cycles, a chart is presented below (suggested by the initiator of this project, Laszlo Hunyadi).



1.4. Methodology

This Literature Study is the first step in starting the work on the main project "Biomass and Natural Gas". It serves as an introduction to the subject of the project and as an investigation of published work on the relevant topic.

The broad scope of the project has provoked the author to try to encompass in this report all aspects that go together with the main core of the study. This is of course a very demanding job, which cannot be done in its full extent. It is virtually impossible to collect and review every published paper on the major subject of this study. In this connection, the author has done his best to present the history, present situation and future trends in the development of hybrid combined cycles in general and hybrid combined cycles with biofuel-fired bottoming cycle - the main topic of the present investigation and the project work that follows. Together with this, a considerable amount of information and statistical data on the broader issues of biomass and wastes utilization and handling is provided.

The Literature Study Report is organised in seven chapters:

Chapter 2 "Biomass and MSW as Fuels" presents an outline of biofuel availability and importance of its use as an energy source, together with its major characteristics and utilization methods. The accent is put on Sweden.

Chapters 3, 4, 5 and 6 are the core of the Literature Study.
Chapter 3 "History and Development of Hybrid Combined Cycles (HCC). HCC with Coal-Fired Bottoming Cycle" is an overview of the history and development of supplementary fired CC and HCC with coal-fired BC.

Chapter 4 "Hybrid Combined Cycles with Biomass and MSW as Bottoming Fuel" is a presentation of research and publications on biofuel-powered HCC.

Chapter 5 "Special Attention to Internal Combustion Engine Combined Cycles" is an overview of pure combined cycles and hybrid cycles with ICE as TC.

Chapter 6 "State-of-Art of Biomass HCC in Sweden and its neighbouring countries" is a description of HCC units with biomass–fired BC in operation in Sweden, Finland, Denmark and some other countries.

Chapter 7 presents some concluding remarks on this Literature Study and on the research project that follows.

The subsequent main project will deal with thorough modelling, calculation and thermodynamic analysis of natural gas and biofuel powered HCC, as explained above. Various cycle configurations of all basic types will be modelled and heat-balance-calculated with the help of a computer program "ProSim". Each cycle configuration will be optimised for highest electrical efficiency in condensing mode, in order for comparison to be performed. Highest total efficiency in CHP mode will also be investigated. Comparison of the different cycle configurations will follow, taking into account their thermodynamic performance and complexity.

The effect of different topping cycle to bottoming cycle power ratios on the efficiency of biomass and topping fuel utilization (efficiencies attributable to the specific fuel), for every different cycle configuration, will be examined. Comprehensive part-load characteristics for every cycle configuration will also be a topic of investigation.
Finally, an attempt will be made for economic calculations of a chosen promising cycle configuration in CHP mode for practical installation in a Swedish municipality. This will require a case study to be performed and the results will be very dependent on the amount of exact information available.

Power unit sizes, which will be considered in the simulations after this literature study, are in general less than 100 MW total useful energy output. More introductory information on the types of cycles and fuels that are under consideration is presented throughout this report.

2. BIOMASS AND MUNICIPAL SOLID WASTE (MSW) AS FUELS

2.1. Utilization of Biomass and MSW world-wide and in Sweden

2.1.1. Importance of Biofuels as sources of energy

The mere existence of Life on Earth is supported by solar energy. The most important process in nature, the photosynthesis, captures the energy of sun's radiation and synthesises organic substances out of inorganic ones. These organic substances are the energy source for all other life forms in the food chain. In other words, photosynthesis, or the ability of plants to fix atmospheric carbon, is the most essential process that provides our food and most of our energy. Since the times when life has started to exist on Earth, photosynthesis has created and continues to create huge storage of hydrocarbons, which we see today as growing plants or fossil fuel deposits.

The energy invested by nature in the production of photosynthetic products is enormous. Incident on the top of the atmosphere is a continuous radiant power of over 10^{17} W. Green plants collect and utilize around 0.02% of this, producing a total annual energy storage of 10^{21} J [2.29]. Plants' photosynthesis also serves as the principal generator of atmospheric oxygen, critical to the respiration of animals and plants, as well as for all important combustion reactions which humans use for converting energy to sustain their life. Prior to human industrialization, total energy stored in biomass has been well in excess of human needs. Currently, this is only a little more than three times the total human non-food energy consumption including all energy forms – fossil, nuclear, geothermal, gravitational and direct solar [2.29].
Views differ about how much of the total world energy needs are provided by biomass, probably depending on estimation methods, forms of energy included in the analysis and of course on the information available. According to Jenkins et al. [2.29] and Gustavsson et al. [2.22] (who refer to the UN statistics), biomass now contributes around 6% of global non-food primary energy consumption, much of this through primitive low efficient and polluting combustion in poorly controlled heating and cooking fires, which support the major share of the worlds population. According to other estimates, cited by Bain et al. [2.3] and Gustavsson et al. [2.22], biomass provides 14%-15% of world's energy needs (the share in developing countries being much larger), which ranks it fourth as a primary energy resource.

Biomass is still a traditional and sometimes the only locally available source of energy for many regions in the world. Under favourable circumstances, biofuels can contribute significantly to the energy mix in industrialized countries as well. However, if biomass is to play a major role in the world's energy mix in the longer term, crops will need to be grown specifically for energy. Studies performed by a number of investigators for example, have suggested that within 10 years the United States could produce large quantities of energy crops (more than 1400 TWh energy value), potentially competitive with coal in many locations. Initial installations will need to be profitable in order for the concept of dedicated energy feedstock supplies to gain broader acceptance. [2.3]
In all industrialized countries, the last two decades have shown a dramatic upswing in bioenergy use. All kinds of biofuels have received increasing attention. This growth

has been stimulated by favourable tax policies and regulatory actions both for utilities and industries.

A very small share of the worlds' electrical energy comes from biofuels. The figures differ for the different countries. On average for the European Union, 2% of all electricity production can be estimated to be based on biofuels (biomass residues and MSW). In Finland however, 13% of all electricity is based on biofuels. [2.53]

It is well known that the biggest advantage of using biofuels as sources of energy is their CO_2 neutrality. The carbon stored in biofuels in the form of carbohydrates has been freshly withdrawn from the atmosphere during the formation of these organic substances. Their combustion releases the carbon back to the atmosphere in the form of CO_2 again, but does not contribute to its accumulation because it has been captured in the plant tissues a "short time" before that and is part of the natural carbon circulation routes.
The situation with fossil fuels is very different. They are also organic substances, but are produced millions of years ago and the carbon storage in the form of fossil fuel deposits has contributed to the present atmosphere elemental structure and present climate on Earth. The continuing trend of unrestricted usage of fossil fuels slowly but steadily releases CO_2 to the atmosphere, which accumulates there because its amount is bigger than the amount of carbon in the normal carbon circulation in nature. Part of the excess CO_2 is absorbed by the world oceans (water absorbs CO_2 much better than other common gases), but still the amount of CO_2 in the atmosphere increases. This adds to the notorious greenhouse effect and in general brings our environment out of its established equilibrium, which may lead to unpredictable consequences for the whole planet.

Another significant advantage of biofuels can be their local production. This helps to generate revenues locally and decreases the country's or regional dependence on imported fuels. In the case of MSW and other organic wastes, their utilization as fuels saves the environment from dangerous effluents and eases the ever-growing problem of waste disposal.

Well-controlled incineration is the preferred method for handling MSW, compared to landfilling for example. It reduces the space volume needed for waste disposal in landfills by an order of magnitude. Ashes left after combustion and residues from flue gas treatment account for 20-25% of MSW weight or, due to higher densities, about 10% of MSW volume. Sometimes wastes are incinerated without any energy extraction. Wide-spread electricity generation in conventional MSW incinerators (with typically 20% electric efficiency) would cover about 3% of total electricity consumption in industrialized countries. High performance systems for better MSW energy utilization, such as HCC, could bring this share to more than 5% of total electricity demand [4.10].

MSW is not totally CO_2-neutral. Although most of it has biological origin, some of its constituents with the highest energy value, the plastics, originate from fossil fuels. Plastics constitute a very small %wt in the total unsorted MSW mass, but a quite substantial percentage of its LHV. Measured on energy value, MSW combustion emits around 8 grams fossil Carbon per MJ thermal energy. Still, these emissions are approximately two times less that those from natural gas [2.21].

It must be pointed out that utilization of all biofuels usually involves consumption of some fossil fuels for harvesting/collecting, handling and transport, so that CO_2-neutrality is generally below 100%. A carefully made Life Cycle Analysis, taking into account all fossil fuels used in one form or another to aid the energy extraction from biofuels, helps to evaluate and show the real CO_2 emissions.
More on Life Cycle Analysis and some results for Swedish conditions are presented in the last subsection 2.4.2 “Life Cycle Analysis” of section 2.4.

2.1.2. Biomass utilization in Sweden

Biomass has always been one of the main natural resources in Sweden. The role of biofuels in the Swedish primary energy mix is constantly increasing. Between 1970 and 1998, their share in the total energy consumption increased from 9 to 15 percent. In absolute values this means that the use of biofuels for energy purposes has more than doubled during this period. [2.25]
Utilization of biofuels in various applications has been continuously promoted in Sweden by both ecological and governmental organisations. Considerable progress has been made in developing infrastructure, research on technologies and planning. Modern small-scale heating units for single family houses as well as medium-scale district heating or CHP plants and industrial CHP plants have been operating since many years.

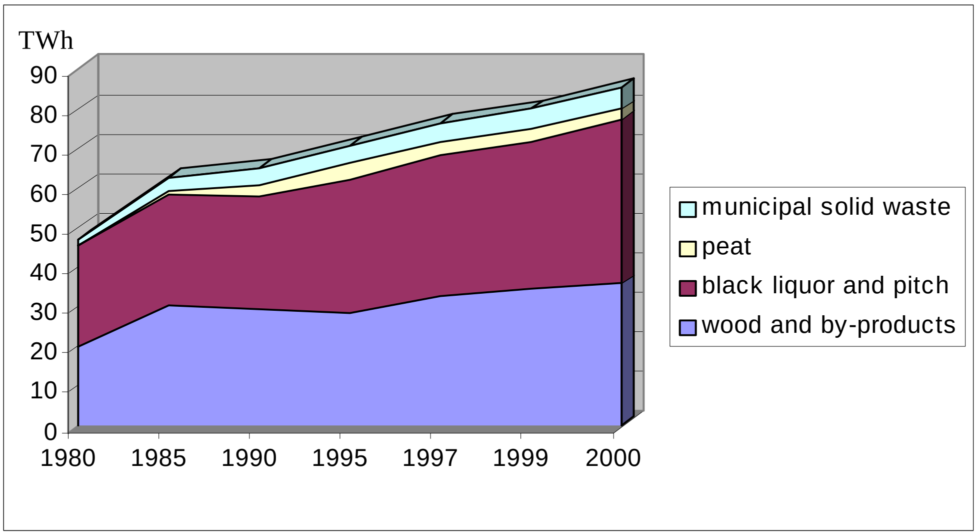


Fig. 2.1: *Total use of biofuels in Sweden, in TWh, excluding chopped wood for traditional heating of detached houses in the countryside. (data from [2.51] and [2.53])*

The increase in total use of biofuels in Sweden in the last two decades has been remarkable (see **Fig. 2.1**). It has been fostered by the energy and environmental policies applied over the past 25 years. In energy values, the figure for 1998 reached 326 PJ (90.6 TWh)[1)] [2.25] (other references state 92 TWh for 1998 [2.51]), for 1999 - 94 TWh [2.53], [2.51] and for 2000 - close to 97 TWh [2.53]. This corresponds to 16%

[1)] Energy units: 1 TWh = 3600 TJ = 3.6 PJ; 1 PJ = 10^{15} J.

of all primary energy consumption in Sweden in year 2000 [2.53]. Included in the numbers is also the traditional use of fuel wood for heating private houses in the Swedish countryside, about 12 to 15 TWh per year [2.53], [2.51], [2.22]. The utilization of wood fuels has grown faster than that of other types of biofuels. [2.25]

The largest users of biomass as a raw material in Sweden are the huge and well-established forest industries – pulp & paper and timber industries. Accordingly, the largest amounts of biofuels utilized for energy purposes come as residues from these industrial processes or from the logging and handling operations (see **Fig. 2.2**). The forest industries are themselves the largest users of biofuel-based heat and electricity.

Different types of biofuels are used in the Swedish energy system, including digester liquors from pulp mills, wood residues such as logging residues, sawdust and bark.
Other important biofuels, although in much smaller amounts, come as agricultural residues, straw, wastes of biological origin, energy crops (willow and grass) and peat. Peat has also been considered as a biofuel and is exempt from the CO_2 tax, but is levied with the sulphur tax.

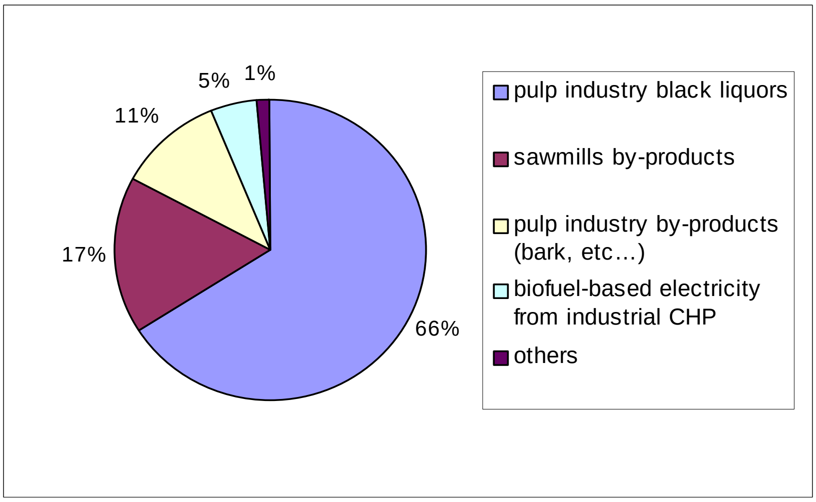


Fig. 2.2: *Types of biofuels used by Swedish industry in 2000, 59 TWh in total. (data from [2.51] and [2.53])*

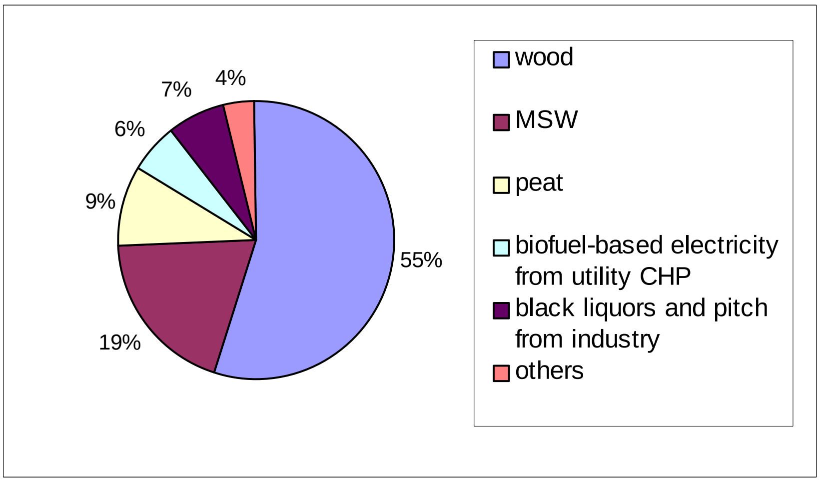


Fig. 2.3: *Types of biofuels used for district heating in Sweden in 2000, 26.6 TWh in total. (data from [2.51] and [2.53])*

Biofuels are particularly used within forest industry and for district heating, but their application for electricity generation in CHP units is also steadily growing. As of year 2000, 56 TWh were used within the industry, 25 TWh for district heating (including 2.6 TWh of peat) and 4.5 TWh for electricity production (1 TWh more than in 1999). Out of the biofuels for electricity production, 1.6 TWh were used in utility CHP plants and 2.9 TWh in industrial cogeneration. [2.53]

In 1999, about 70% of all district heat in Sweden was based on biofuels, compared to 5% in 1980 [2.51]. In the district heating sector, the use of biofuels has been growing by approximately 5% annually during the past 5 years [2.25]. See **Fig. 2.3** and **2.4**.

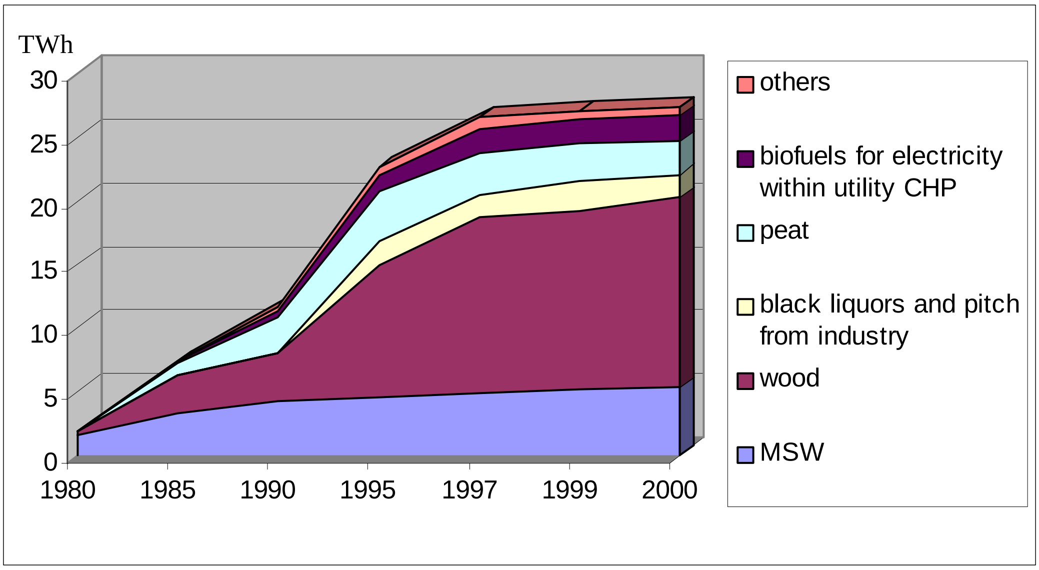


Fig. 2.4: *Rise in biofuel application in the district heating sector in Sweden, in TWh. (data from [2.53])*

Part of the wood fuel originates from recycled wooden materials. Their disadvantage is that they may contain harmful contaminants from former processing, for example from gluing, painting and impregnation, as well as pieces of metal. Combustion of such materials produces pollutants both in the flue gases and in the ash. This means that flue gas cleaning equipment must be added to the boiler and the ash cannot be returned to nature but must be dumped in a landfill.

The electricity generation capacity from biomass has increased as a result of governmental investment support for CHP plants. In a five-year period starting from 1991, 45 different projects totalling an installed electrical capacity of 326 MW received governmental support. A later version of this support scheme ending in 2000 is expected to result in further 164 MW of electricity generation capacity in 10 different projects. These investments are all in new boiler capacities connected to municipal district heating networks. The investment support involved the obligation to use biomass in the new plants for 5 years, at a rate of up to 85% of total fuel input. [2.25]
In a typical year, 90% of the harvested biomass from forests in Sweden will be used in the forest products industry and 10% will be used as fuel wood. Of the 90%, about half is used as timber wood and half as pulpwood. Around 40% of the timber and pulp

wood end up as residues and are used for energy purposes. This means that about 45% of the harvested forests in Sweden are eventually used for energy purposes. [2.25]

A commercial market for wood fuels has evolved during the last decade in the district heating system that comprises approximately 100 wood fuel-fired plants. The ten largest wood fuel consumers account for 45% of this market. There is also a small commercial market for wood fuels in the forest products industry and among households, the latter mainly in the form of wood pellets.
According to the Swedish Biofuel Association, there are 57 biofuel production companies on the market. The 10 largest wood fuel producers account for more than 58% of the market. In real terms, the market has increased more than eight-fold in 13 years. Nevertheless, there is still substantial potential for increasing the use of biomass in district heating. [2.25]

2.1.3. Biomass and MSW resource base in Sweden

More than 50%, or 23 million ha of Sweden's total land area is productive forestland. The annual increment is about 100 million cubic meters total stem volume including bark, out of which about 70 million cubic meters are harvested every year by the timber and pulp and paper industries [2.25], [2.51]. The sustainable annual resource base of wood fuels from the Swedish forests has been estimated in various studies. When incorporating into the calculation certain restrictions of technical, ecological and economical nature, wood fuels with an energy value of the order of 130 PJ (36.1 TWh) are believed to be available for the energy market in the next ten years at prices below present levels. [2.25], [2.21]
There are about 20 large suppliers of wood fuels in Sweden. Most of them are associated with large forest companies or forest product companies. Some municipalities and power companies are also owners of large wood fuel supply companies.
In addition, substantial amount of biofuels is regularly imported commercially to Sweden. Imported biofuel (in the form of wood residues) is at present cheaper than domestically produced biofuels. In 1997, imports accounted to around 30-40% of the total biofuel consumption of the Swedish district heating plants, coming mainly from Germany, the Baltic States and North America. [2.25], [2.18], [2.53]

The industrial use of wood fuels is highly dependent on the prices of competing fossil fuels and is also affected by policy instruments. Fuels are traded on the international market, while energy policies have so far been mainly national. The expected common energy policy of the European Union is likely to affect biofuel trade. [2.25]

Productive farmland in Sweden covers 3 million ha, or about 7% of the country's total area. The agricultural sector produces significant amounts of biomass that can be used as fuel. The main sources are by-products of cultivation of various crops, manure from livestock, wastes from agricultural processing industries, as well as energy crops. A typical example of a by-product from cultivation is straw. Straw is currently used in three district heating plants in southern Sweden [2.25], [2.21] and corresponds to 0.5 TWh thermal energy [2.51].

Liquid manure from cattle farms can be either dried and directly burned, or anaerobically digested to produce biogas, which is a valuable gaseous fuel.
Agricultural and food processing industries produce large amounts of by-products, such as waste from slaughterhouses, sugar mills and breweries. Most of these wastes are valuable additives to the anaerobic digesters for production of biogas.

The excess production of food crops made it possible to produce energy crops on agricultural land. The energy crops (short rotation coppice) have undergone rapid development in Sweden during the last 10 years. Now the area planted with fast growing trees (willow - *Salix*) has stabilised to about 14'500 ha, all in the southern third of the country [2.60]. A wide research program covering most aspects of *Salix* production and utilization began in the 1970s. Alongside the research, different types of support have been given to make energy crops an attractive and competitive option for farmers with redundant farmland. The willow trees are harvested once in several years, directly cut into wood chips and sold to the district heating sector. The potential for energy crops is great and continuing increase is expected in the future. In 2000, only 0.1 TWh were used. *Salix* cannot grow in northern Sweden, but there are herbaceous species (grasses, harvested annually), which are very suitable and well adapted to the northern climate. However, the economy of short rotation forestry depends strongly on agricultural policies, economic support and prices for competitive biofuels. [2.25], [2.53], [2.60]

The average amount of MSW (household waste) in Sweden is around 300 kg per person per year, with average energy content of 10 MJ/kg (LHV). 50% of it is used as fuel, which is approximately 1'350'000 tons per year [2.25]. A small part of the waste is processed into RDF. Recycling of materials is given priority over energy extraction from MSW in Sweden, but incentives for decreasing the landfill operations together with better separation and use of combustible waste as fuel are created, which will lead to continuous increase in the power generation from waste.

Other sources show that 1'450'000 tons of MSW together with 809'000 tons industrial wastes are combusted in Sweden every year [2.51]. There are 22 major waste incineration plants, in which the size of the furnaces varies from 1 to 60 MW_{th}. The total installed thermal capacity is 740 MW (as of 1998). District heating based on wastes incineration supplies around 10-12% of the country's needs. Energy extraction from MSW in 1999 was 5.2 TWh [2.53], [2.51] and in 2000 - 5.3 TWh [2.53].

In the Stockholm area, the household waste is estimated at 600'000 tons annually. With an energy value of around 3 kWh/kg (10.8 MJ/kg LHV), it can provide 1.8 TWh thermal energy per year. Taking into account the fact that a great deal of paper and plastics are sorted out of the waste, the available thermal energy is around 1.3 TWh per year [2.13]. Furthermore, there are about 550'000 tons of industrial waste (garbage from shops, factories and construction sites), which can provide another 0.8 TWh thermal energy. In total, the possible thermal energy extractable from MSW in the Stockholm area lies around 2.1 TWh (the total district heating supply in Stockholm's county is around 11 TWh). [2.13]

Biogas supplies a quite small amount of the energy in Sweden. In 1998, 27 GWh were used for electricity production and 298 GWh for heat production [2.25]. The corresponding figures for 1999 are 33 GWh for electricity and 447 GWh for heating (yet an increase can be observed) [2.53]. This contribution is expected to rise substantially within the next decade. In general, the economic performance of present biogas facilities has not been particularly good. Nevertheless, driving forces for increased biogas production do exist. The resources for biogas production (cattle manure and rich organic residues from the food processing industries) are generally underutilized. Part of the MSW can also be diverted to biogas digesters and utilized as valuable fuel in the form of biogas, although anaerobic digestion of MSW is much more difficult to implement and control (and the yield is far less) compared to manure and food processing wastes (see subsection 2.3.2.2.). The utilization of the biogas itself can be made in a much more profitable way, for example as an automobile fuel.

In general, consumption of biofuels in Sweden is strongly dependent on their prices, which at the moment are competitive with prices of conventional fuels only due to the heavy taxation of all fossil fuels. Tax incentives and pressure from society remain the major impetus for continuing increase of biofuel use as energy source.

Table 2.1: *Comparison of fuel prices in Sweden, including all taxes and VAT, in SEK/kWh for year 2000. (data from [2.53])*

	Industrial Users SEK/kWh	**Households** SEK/kWh
Gasoline 95	-	0.87
Diesel	0.297	0.77
LFO	0.318	0.56
HFO	0.23	-
Coal	0.128	-
Woodchips, 50% wet	0.112	0.14
Electric heating	0.25	0.73

2.2. Combustion Properties of Biofuels

Efforts for technical enhancement in the contribution of biomass to commercial energy forms are focused on improving both the efficiency and environmental impact of biomass conversion. Still there is a need for further investigation of the properties of biomass, in regard to its conversion into thermal energy by combustion processes.
A better understanding of not only how the conversion technology can be adapted to fit the properties of the biomass fuel, but how the properties of the fuel might be varied to suit the conversion technology of choice, is necessary.

Plants rely on certain fundamental processes for growth and reproduction, yet have evolved to accommodate a great diversity of ecosystems and environmental conditions. Accordingly, they exhibit certain gross similarities in properties, yet with substantial specific variation. Many of these properties are critical for proper design and operation of energy conversion facilities, although not all properties are equally important in every conversion technique. As the biomass-fuelled power generation industry has expanded in recent decades, the diversity of fuel types utilized has expanded as well, often with unanticipated and undesirable impact on facility operation. Although the fundamental combustion behaviour of biomass fuels has received increasing attention and has been well understood, there is still space to be filled with knowledge about combustion characteristics and the importance of certain properties for the design, control, operation and maintenance of energy conversion facilities. [2.29]
Combustion is a complex phenomenon, involving simultaneous heat and mass transfer, governed by chemical reactions and fluid flow. Its prediction for the purposes of design and control requires knowledge of fuel properties and the manner in which these properties influence the outcome of the combustion process.

In general, combustion is a chemical reaction with oxygen (oxidation) of certain chemical elements from the fuel into their oxides. As a natural phenomenon, combustion is a common process for "downgrading" chemical elements from their unstable forms to their most stable forms – the oxides, in which forms they naturally occur in the environment. These chemical reactions, being exothermic ones, release energy in the form of heat. In order for the fuel to be formed (produced), energy must be involved (dumped) to bring the elements from their stable forms into "high-grade" unstable forms, or fuels. Biomass has performed this "upgrading" process, using the sun as an energy provider and involving sophisticated chemical conversions (still not understood completely by human science) to form complex carbohydrates out of carbon dioxide and water as raw materials. Carbon and hydrogen are exactly the main ingredients of biological matter, whose conversion back into oxides in the combustion process provides the release of energy in the form of heat.

2.2.1. Elemental structure of biomass. Ash composition.

In the process of forming hydrocarbons, plants need many "supporting" chemical elements in small quantities, which must exist in the environment in order for the plant to thrive. Some other chemical elements are not crucial for the plants' survival, but are entrained and stored into the plants' growing matter in variable quantities. In these two groups of additional elements present in the biological matter fall almost all known chemical elements that exist in nature. They all have influence on the combustion properties of biomass and on the combustion process itself and may or may not undergo conversion.

A general reaction formulation for the combustion of biomass fuel in air might take the following form [2.29]:

$$C_{x1}H_{x2}O_{x3}N_{x4}S_{x5}Cl_{x6}Si_{x7}K_{x8}Ca_{x9}Mg_{x10}Na_{x11}P_{x12}Fe_{x13}Al_{x14}Ti_{x15} + n_1H_2O + n_2(1+e)(O+$$

$$3.76N_2) = n_3CO_2 + n_4H_2O + n_5O_2 + n_6N_2 + n_7CO + n_8CH_4 + n_9NO + n_{10}NO_2 +$$

$$+ n_{11}SO_2 + n_{12}HCl + n_{13}KCl + n_{14}K_2SO_4 + n_{15}C + \ldots\ldots\ldots$$

The first reactant compound is the biomass fuel (generalised empirical formula). The second reactant term expresses the moisture in the fuel, which can be extremely variable. The third term represents air (although this again is a simplification) as a simple binary mixture of oxygen and nitrogen in the ratio of 21 %v to 79 %v.
The product side of the reaction is complex. The main products are those appearing first. It is difficult to identify all of the reaction products, some of them have short lives within the combustion chamber and are not present in the cooled exhaust gases. The detailed chemistry of the reaction is far from understood. Making standardizations and engineering recommendations concerning the design of biomass conversion energy systems is very difficult.

Table 2.2: *Results of chemical analysis, identifying the elemental coefficients from the above formula for two major types of biomass – woody biomass (Hybrid Poplar species) and herbaceous (grassy) biomass (Rice straw). (from [2.29])*

		Hybrid Poplar	***Rice straw***
C	x_1 =	4.1916	3.2072
H	x_2 =	6.0322	5.1973
O	x_3 =	2.5828	2.8148
N	x_4 =	0.0430	0.0625
S	x_5 =	0.0006	0.0057
Cl	x_6 =	0.0003	0.0165
Si	x_7 =	0.0057	0.5000
K	x_8 =	0.0067	0.0592
Ca	x_9 =	0.0337	0.0141
Mg	x_{10} =	0.0205	0.0135
Na	x_{11} =	0.0002	0.0079
P	x_{12} =	0.0012	0.0086
Fe	x_{13} =	0.0007	0.0029
Al	x_{14} =	0.0008	0.0073
Ti	x_{15} =	0.0002	0.0004

From **Table 2.2** it can be clearly seen that the main chemical elements, building the plant tissue, are C, H, and O. Expressed in percentages, carbon constitutes around 40 to 55 %wt, hydrogen around 6 %wt and oxygen around 35 to 42 %wt of the biomass matter.
Other main group of elements represented in the organic tissue are the non-metals most commonly met in nature – N, S, Cl and P. Nitrogen is very important for the plant

nutrition and critical to its growth. Its presence in the biomass matter is unwanted however, because (although inert at normal conditions) in combustion reactions at high temperatures nitrogen reacts with oxygen to form several nitric oxides, generally represented by the formula NOx. Sulphur, Cl and P are also unwanted, as long as they inevitably form hazardous gaseous oxides or other polluting compounds during combustion.
The third major group is the ash-forming elements. They are metals, representatives of the alkali and alkaline-earth groups of chemical elements, also other metals, aluminium, silicon and others. They are important and well-spread constituents of the mineral part of the soil and are entrained in the plant tissue with the water as oxides or as positive ions from dissolved salts. Some of them are very important for the plant nutrition and for the ion exchange processes in every living organism. They are present in the plant tissue both as part of the organic matter and as separate inorganic substances.

Ash in biofuels has a very crucial importance for the design, performance and maintenance of combustion chambers and boiler heat-exchange surfaces. During combustion, most of the ash-forming elements do not undergo chemical reactions of big significance for the heat release process, but they all are actively involved in the general chemical transformations taking place in the combustion zone and emerge eventually as oxides, hydroxides and salts. Boiler ash consists almost entirely of solid oxides and salts of alkali, alkaline-earth elements, aluminium and silicon.
The physical transformations (such as melting, fusion, sublimation and vaporising) that ash-forming substances undergo in the combustion chamber also have an enormous impact on the boiler design, performance and maintenance. The combination of high oxygen content and high organic volatile matter in biomass indicates a potential for creating large amounts of inorganic vapours during combustion. Alkali elements in particular are directly vaporised at normal furnace operating temperatures. Alkali salts and sulphates harm thermochemical conversion systems by fouling and corroding heat-exchange surfaces, gas turbine blades and other components. [2.29], [2.3]
Chlorine is a major factor in ash formation. It facilitates the mobility of many inorganic compounds. Chlorine concentration often dictates the amount of alkali vaporised during combustion as strongly as does the alkali concentration. Potassium chloride is among the most stable high-temperature, gas-phase, alkali-containing species. In most cases, chlorine appears to play a shuttle role, facilitating the transport of alkali from the fuel to surfaces. In the absence of chlorine, alkali hydroxides are the major stable alkali-containing gas-phase species in combustion gases. [2.29]

The inclusion of 15 elements in the empirical formula of the fuel above is incomplete. There are many more, some of which are also important to the issue of biomass combustion. Heavy metals, for example, have a strong influence on ash disposal, but are contained in infinitesimal quantities and cannot be easily taken into account in the elemental structure above.
The amount at which every element is present in the plant tissue is strongly dependent on the plant type, soil type and conditions of growth.

Inorganic materials such as dirt, soil and stony particles are usually incorporated in the biofuels during their harvest, transport and storage. Such additional material can be a major part of the ash in the fuel.

Compared to coal, biomass has lower ash content, much higher oxygen content, lower nitrogen content and lower sulphur content. The low amounts of S, N and ash-forming elements in the biomass give it a big advantage to the solid fossil fuels. The very high oxygen content, decreases the calorific value per unit weight of fuel and can be considered as a drawback. However, oxygen plays a role in another important property of the biomass as compared to some coals for example – its high volatility. The high volatility helps a fuel to be easily ignited at lower temperatures and more thoroughly combusted.

Compared to each other, the two main types of biomass, woody and herbaceous biomass, have also some quite important differences. In general, all biomass with herbaceous origin (grass, straw, agricultural by-products such as shells, hulls and pits, also leaves and green parts of woody plants) have lower contents of the important elements (carbon and hydrogen) and higher contents of all unwanted elements as oxygen, nitrogen, sulphur, phosphorus, chlorine and especially ash-forming elements. The content of alkaline-earth elements is slightly lower in herbaceous biomass, but the alkali elements content is much higher. The very high silica content in the herbaceous biomass (up to 10 – 15 % SiO_2 in dry matter) is very curious. [2.29], [4.4]

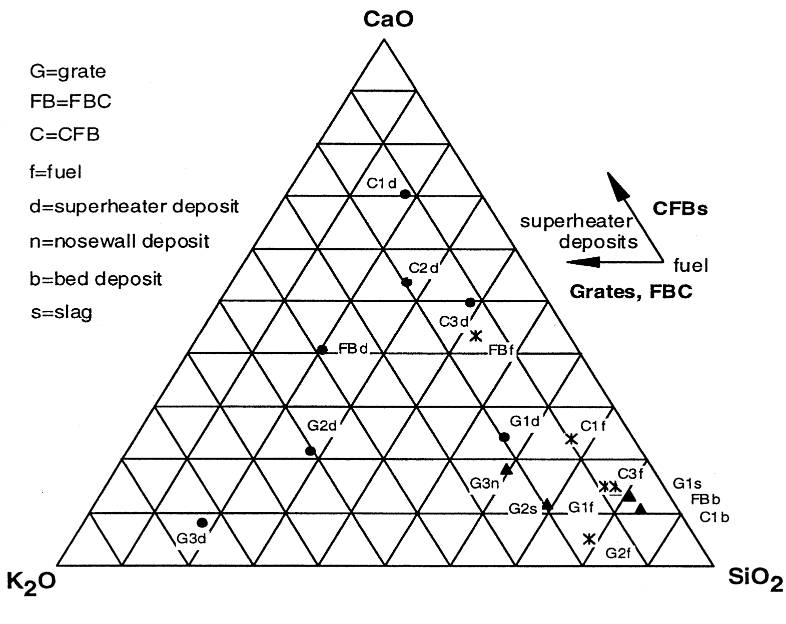


Fig. 2.5: *Normalized ash compositions, various deposits and slags from commercial biomass power plants in the three component oxide system Si-K-Ca. Each corner of the diagram represents a composition consisting entirely of the compound shown at that corner. Intermediate values are mass concentrations of the oxides. Grid spacing is 10%. The first letter of the designation refers to the type of combustion unit, the number following it refers to individual units. The lower case letter refers to the type of material as shown in the legend. (from [2.29]).*

This difference between the two types of biomass (woody and grassy biomass) has a significant impact on their energy utilization in combustion chambers. The higher ash content and the high silica and alkali content in the ash from herbaceous biomass makes burning and ash handling much more difficult. This has a particular effect on

boiler design and maintenance. For example, burning straw in a boiler designed for wood, causes rapid sintering, slagging and fouling.
Although the detailed chemistry of ash slagging and fouling is not yet fully developed, removal of alkali and other elements is known to increase the fusion temperature of the ash. In general, lower alkali content improves ash behaviour substantially.

Exact values of the elemental composition of biomass are presented in **Appendix A.1.**

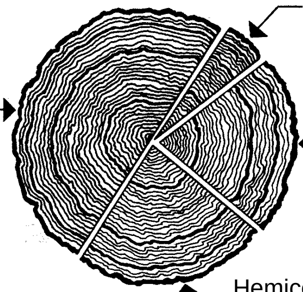


Fig. 2.6: *Classification of organic substances in woody biomass. The three major organic materials in biomass structure are Cellulose, Hemicellulose and Lignin. Many precious organic substances constitute the Extractives, depending on the plant species.*
(from [2.62])

2.2.2. The energy value of biomass. Combustion kinetics.

As with every fuel, the standard measure for the energy content of biomass is its heating value (calorific value), expressed as lower or higher heating value.
Ash content and moisture content however, decrease significantly the heating value of all solid fuels, including biomass. Heating value of every fuel is directly proportional to the amount of carbon and hydrogen in it and inversely proportional to the amount of all other elements and moisture.
On dry ash-free basis, biomass fuels have heating values comparable with those for mid-quality coal, around 20 MJ/kg. The difference between HHV and LHV for biomass is quite small, due to the low amount of hydrogen (not more than 6%wt). Ash content in biomass is comparatively low and it does not have a profound effect on heating values.
The high content of oxygen in biomass leads to less air needed for complete combustion of the fuel. The less air needed for stoichiometric combustion, the less diluent in the form of atmospheric nitrogen must be heated along with the combustion products to achieve the adiabatic flame temperature. Stoichiometric air-fuel ratios for hydrocarbon fuels are typically between 14 and 17, for biomass they are 4 to 7. Adiabatic flame temperatures for biomass (dry basis) lie typically well above 2000 K, up to 2700 K. For comparison, the adiabatic flame temperature of methane (CH_4) in air is approximately 2300 K, while the higher heating value of methane is almost three times that of wood. [2.29]

In addition to the energy released by combustion, the rate of combustion is also important in the design of combustion systems. The rate at which a biomass fuel burns depends on a number of physical phenomena. Two predominant factors are the rate of heat transfer and the kinetic rate of reaction. Particle size dominates the influence of heat transfer, with small particles heating more rapidly than coarser, thicker particles.

Combustion occurs both in the gas phase with the burning of volatile materials released through pyrolysis of the fuel upon heating, and heterogeneously in the solid phase as char oxidation. The burning of volatiles is generally quite rapid and follows as fast as volatiles are released [2.29]. The oxidation of char occurs much more slowly, so it is the major governing factor for the velocity of the combustion process as a whole. The residence time of the particle in the furnace is important for the complete conversion and for the emissions from the combustor.

Fuel pyrolysis and char oxidation can be classified as the main sub-processes during combustion of a wood particle. After entering the furnace, the following processes are observed to be experienced by a wood particle, starting from room temperature:

1. Up to about 150^oC - drying with a small weight loss;
2. Between 200 and 400^oC – a very rapid loss in weight due to the evolution of volatile material, which will ignite and burn;
3. Following the release of volatiles, there is a slow loss of weight as the residual char gradually decomposes and oxidizes. Carbon first oxidizes to carbon monoxide (CO) on the surface of the char particle, then to carbon dioxide (CO_2) in the volume surrounding the char particle. The speed of the overall process depends strongly on the speed of oxygen delivery to the surface of the char particle and on the conditions for carrying away the products from the particle's surface.

Metals in biomass are known to have an effect on reaction rates and are thought to be catalytic to pyrolysis. [2.29]

Moisture is the factor that is of uppermost significance due to its direct effect on heating value of biomass and on its combustion behaviour. Freshly harvested biomass contains usually more than 50%wt of moisture. The evaporation of the water in the fuel is a strongly endothermic process, which dumps much of the heat released by the combustion. High amounts of water in a fuel can lead to difficulties for this fuel to be readily combusted. The autothermal limit (self-supporting combustion) for most biofuels is around 65% moisture content on wet basis (mass of water per mass of moist fuel). Above this point, the energy liberated by combustion is insufficient to satisfy the heat requirements for evaporation of moisture and heating of the reactants to the ignition point, which means that combustion cannot support itself. Moisture in biomass has also an impact on pollutant emissions from boilers, for example CO – a product of incomplete combustion.
The effect of moisture content on the heating value of biomass can be easily understood with the help of graphs, such as that shown in **Appendix A.2.**

Biomass with moisture content of around 50 %wt (freshly harvested wood for example) has a LHV of approximately 8 MJ/kg, more than two times less than on dry basis. Air-dried biomass (for example wood stored for a long time under a shelter) has moisture content of around 15-20%wt and a LHV of around 15 MJ/kg.

Drying of biofuels prior to combustion leads to a more stable combustion process, lower emissions from incomplete combustion (less CO, UHC and VOC), improved radiative heat transfer in the combustion chamber, lower volume of flue gases and less heat carried away by non-condensed water in flue gases. Effects of high moisture content in biofuels will be shortly discussed in section 2.4. "Biofuel Handling and

Preparation”. A separate work within the present project, a Master of Science Thesis titled “Drying of Biofuels for Energy Purposes”, takes a closer look at the problems associated with moist biofuels and the possibilities for drying them before combustion.

2.2.3. Pollutant emissions

Although being CO_2-free, combustion of biomass is not pollutant-free. As is the case with every fuel, the combustion of biomass emits numerous hazardous compounds.

Primary pollutants formed during biomass combustion are PM, CO, UHC, VOC, NOx (principally NO and NO_2), and SOx (principally as SO_2). Other acidic gases such as HCl may also be emitted, as well as heavy metals.
CO and UHC, including VOC and polycyclic aromatic hydrocarbons, are products of incomplete combustion. These species are largely controlled by stoichiometry, fuel moisture, combustion temperatures and residence time in the combustion chamber. Heavy metals can be present in high concentrations in the exhaust gases from the combustion of RDF and painted or treated woods.
PM includes soot, microscopic ash particles, condensed fumes (tars, oils) and sorbed materials as VOC and aromatic hydrocarbons. PM released in heavily populated areas causes breathing hazards and in humid atmosphere facilitates the formation of smog. PM can also cause cancer.
Emissions of SOx arise from the sulphur content of the fuel, which in the case of biomass luckily is not high and no flue gas desulphurisation equipment is needed.
Emissions of NOx arise predominantly from nitrogen in the biomass. Many commercial biomass combustors (among them all FB combustors) operate at temperatures low enough that thermal NOx contributes only a small fraction of the total. Nitrogen from the fuel however is easily oxidised. The hazardous effects of NOx are numerous and there is no need to discuss them here.
NOx emissions also depend partly on stoichiometry. Fuel-lean conditions produce higher amounts of NOx, to the point where combustion temperatures are decreased much by the additional air and NOx formation rapidly decreases. Fuel-rich conditions produce inherently very low amounts of NOx. Moreover, understoichiometric combustion is known to be able to destroy NOx to a great extent with the help of hydrocarbon radicals, which reduce the NOx back to molecular nitrogen.

Apart from the already mentioned pollutants resultant from the combustion of any solid fuel, incineration of MSW produces also some toxic organic substances (such as dioxins, furans, aromatics and others), which can be classified as VOC and are very harmful. This is aggravated by the fact that temperatures in the MSW combustion chamber are kept comparatively low in order to diminish the possibility of molten ash deposits on the boiler surfaces and the corrosion caused by hydrogen chloride. Lower temperatures increase the production of toxic organic pollutants and prevent their destruction (they are easily destroyed at higher temperatures). Furthermore, the high amount of acid-forming elements, alkali elements and heavy metals also results in poisonous emissions. [2.38]
Because of this, MSW incinerators must always be equipped with flue gas scrubbing systems, to ensure that emissions are within limits. Longer retention time of the combustion products at higher temperatures in an isolated zone can also help destroying the toxic organic substances.

2.3. Technologies for Biomass and MSW Energy Utilization

The technologies for the primary conversion of biofuels to energy can be divided in three main groups: direct combustion, upgrading to gaseous fuels and upgrading to liquid fuels.

2.3.1. Direct combustion

Traditional open heating and cooking fires, cofiring of biofuels in fossil-fired boilers and combustion in boilers designed specifically for biofuels, are the utilization methods that fall into the direct combustion group.
Direct combustion is the least complicated technology that can readily utilize any type of fuel.

2.3.1.1. Pile burner

Pile burners represent the historic industrial method of wood combustion and typically consist of a two-stage combustion chamber with a separate furnace and boiler located above the secondary combustion chamber. The combustion chamber is separated into a lower pile section for primary combustion and an upper secondary combustion section. Wood is piled on a grate in the bottom section and combustion air is fed upwards through the grate and inwards from the walls. Combustion is completed in the secondary combustion zone using overfire air. Wood is introduced either on top of the pile or through an underfeed arrangement. Ash is removed manually by dumping it from the grate after it's cooled. [2.3]
Pile burners typically have low efficiencies, cyclic operating characteristics (because of cyclic fuel feed and ash removal) and the combustion is erratic and difficult to control. They have slow response and are not suitable for load following operations. Their only advantage is their simplicity and the ability to handle wet, dirty fuels.

2.3.1.2. Stoker grate

Stoker combustors feature a moving grate, which permits continuous and easy ash collection and wood feeding. In addition, the fuel is spread more evenly and in a thinner layer in the combustion zone, giving more efficient combustion. The bottom of the furnace is the moving grate, which is cooled by underfire air. Underfire air defines the maximum temperature of the grate and the allowable moisture content of the fuel. Combustion is completed by the use of overfire air.
Modern designs employ a sloping reciprocating water-cooled grate. Low NOx levels can be achieved in grate combustors by using staged combustion – fuel burns on the grate in slightly understoichiometric conditions and half of the air is supplied as overfire air. In this way temperatures in the furnace are maintained at lower levels, which has a positive impact not only on the NOx formation but also on ash behaviour (furnace temperature is below ash deformation temperature). [2.3]
Grate boilers of various designs are commonly used for MSW incineration, **Fig. 2.3**.

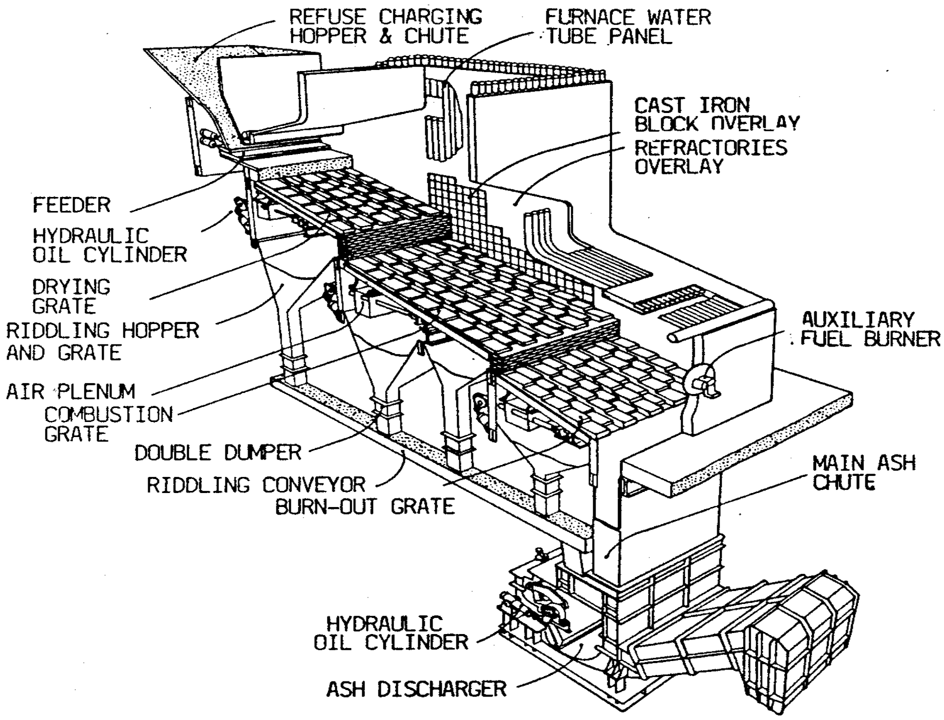


Fig. 2.7: *A reciprocating grate MSW incinerator. Cross sectional schematic of the combustion zone. (from [2.36])*

2.3.1.3. Pulverized wood burners

The combustion of pulverized wood in high heat load blast furnaces is derived from the similar technology for coal. It is also called "suspension firing". It requires a feed moisture content of less than 15% and a particle size less than 1.5 mm. The high heat load results in smaller furnace size. High boiler efficiencies are achievable, up to 80%. Offsetting the higher efficiency is the cost for drying and pulverizing of the feedstock. In addition, special burners need to be used. Burners developed for suspension firing include scroll cyclonic burners and vertical cylindrical burners. [2.3]

2.3.1.4. Whole tree technology

A company in USA has developed a special innovative technology for burning wood. As the name suggests, the concept involves burning whole trees in a staged combustion system with the use of low temperature exhaust gas to dry the trees.
Trees are transported to the power plant, cut into pieces of desired length and stored in the form of a large loosely stacked pile, under which air heated by the flue gas is ducted to dry the trees for several weeks. The moisture content falls to about 25%, resulting in nearly 10% increase in furnace efficiency [2.46]. Then the feed is introduced into the primary combustion chamber through a ram charger. The primary combustion chamber is envisioned as a deep bed, where the whole trees burn, supported by a water-cooled grate, in substoichiometric conditions. The gases leaving

the primary combustion chamber will be burned with overfire air under excess air conditions. Char falling through the grate will also be burned.
The advantages of such a technology are the reduced operating costs. Higher efficiency is also expected. [2.3], [2.46]

2.3.1.5. Fluidized bed combustors

The fluidized bed technology is well developed and many installations are already in use throughout the world. FB combustors have the advantage of extremely good mixing of the reactants and high heat transfer rates, resulting in very uniform combustion conditions and low overall bed temperatures. The combination of good mixing with low combustion temperatures in the FB system results in extremely efficient combustion (almost no CO, UHC and VOC, with typically 99-100% carbon burnout) and very low NOx emissions.

FB boilers can handle almost any kind of fuel and are very suitable for combustion of biomass (wood, MSW, waste sludge or others) with low emissions. Ensuring a dwell time of several seconds in an isolated zone at slightly higher temperatures than in the primary combustion zone (with special temperature profiling to avoid recombination) helps meet the requirement for low emissions of toxic organic pollutants [2.45].
One other advantage of the FB boiler as a BC in a fully-fired HCC is the fact that a significant part of the NOx emissions in the TC exhaust can be destroyed in the combustion process in the FB boiler. NOx generated in the TC is decomposed (deoxidised) under high temperatures and locally substoichiometric conditions in the boiler. The result is that NOx concentration of the boiler flue gases tends to be much lower than that obtained by simply adding the NOx generation in the TC to that of the BC [3.26]. Moreover, final NOx concentration after the FB boiler can be lower than that of the TC itself, if the generation of NOx in the TC is very high (for example when an ICE is used as TC). Interesting experimental results from such a research project undertaken in 1988 at Chalmers Institute of Technology, Sweden, have been presented by Wingård et al. [5.34]. A commercial fully-fired HCC configuration (gas turbine and fluidized bed boiler) at a utility CHP plant in Eskilstuna, Sweden, has confirmed this effect, reported by Björklund and Bohman, 1992, in an internal report for the utility Tekniska Verken (presently Eskilstuna Energi & Miljö) in Eskilstuna.

A concern for FB technology is the danger of bed agglomeration. High alkali contents in fuels cause particles in the bed to agglomerate, grow in size and defluidize, eventually plugging the whole system.

a) Bubbling Fluidized Bed (BFB):

In a BFB a stream of gas (air) passes through a bed of granular material, with the fuel particles dispersed in it, and the gas velocity is just enough for the solid particles to be suspended and to circulate freely throughout the bed without being carried away. The BFB looks like a boiling liquid and has the physical properties of a fluid.
In combustion of biomass, the fluidizing medium is air and the bed is composed usually of sand or limestone particles. Air velocity is around 1.5 – 3.5 m/s. Overfire air is normally introduced over the bed. BFB boilers are usually designed for complete ash carryover, necessitating the addition of cyclones and/or bag houses for particulate

control. A cyclone is used to either return fines to the bed or to remove ash-rich fines from the system. Biomass is introduced either through a feed chute to the top of the bed or through an auger right into the bed. In-bed introduction of feedstock is to be preferred, as it provides longer residence time for fines, which would otherwise be directly entrained in the fluidizing medium.

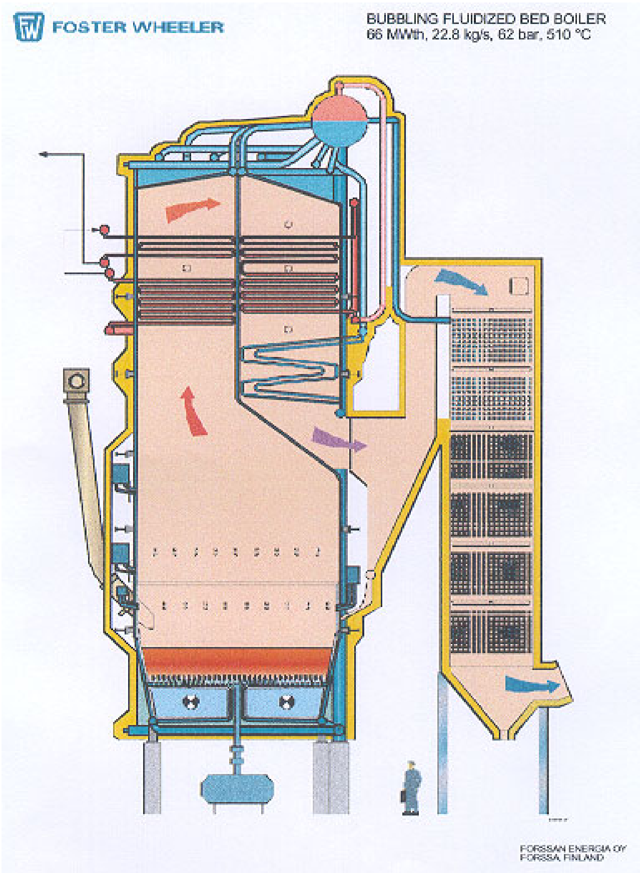


Fig. 2.8: *Schematic of a BFB steam generator. (from [2.1])*

Upon start-up, the bed is preheated to the fuel ignition temperature using an external burner fired by natural gas, propane or fuel oil. For biomass this temperature is around 540^{o}C. At this point, biomass is slowly introduced into the bed and the temperature gradually climbs up to the normal operating point of around 850^{o}C. During operation, the combustion process is completed in the freeboard space above the bed, resulting in temperatures above the bed reaching 980^{o}C. [2.3]

b) Circulating Fluidized Bed (CFB):

If the velocity of the airflow in a BFB is increased, the bubbles become larger, forming large voids in the bed and entraining substantial amounts of solids. This type of bed is referred to as a “turbulent FB” [2.3]. If the air velocity is further increased, the solids in the bed start to flow together with the fluidizing medium and there is no distinct separation between the dense solid zone and the dilute zone above it, the flow forms a large circulating system – a CFB.

In a CFB, the solids are separated from the gas in cyclones, collected and returned to the bed, forming a solid circulation loop. CFB has lower bed densities than BFB, while the air velocity is as high as 9 to 10 m/s. The residence time for the solids in a CFB is determined by the solids circulation rate, the friction characteristics of the solids and the collection efficiency of the separation device. The fuel concentration in the bed is usually very low. CFB temperatures are maintained at about 860-870^{o}C. [2.3]

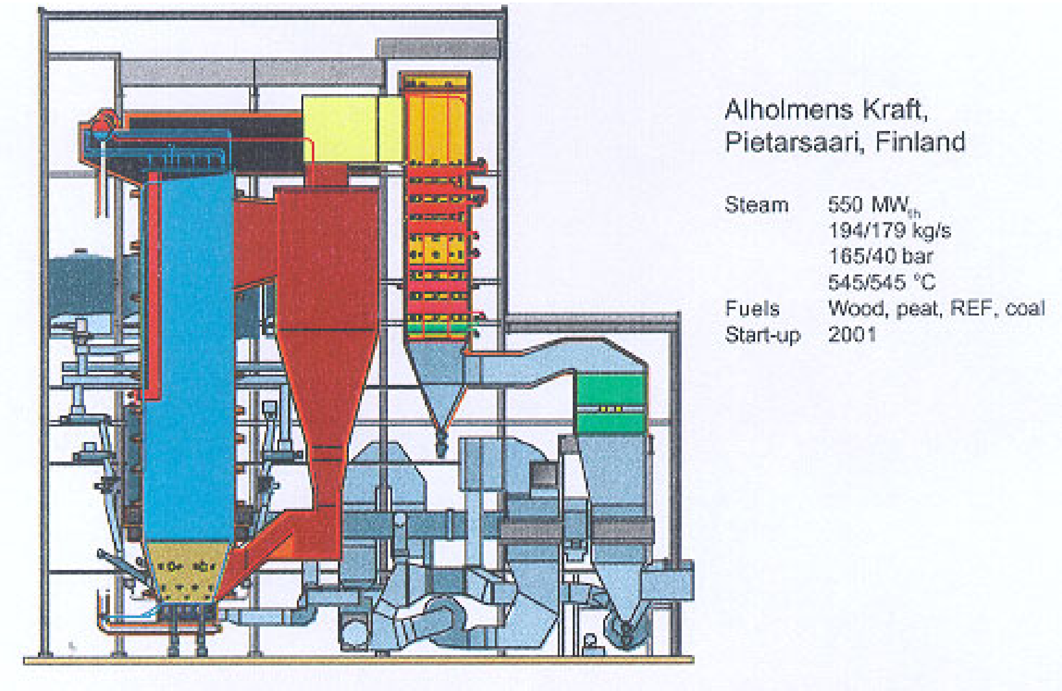


Fig. 2.9: *Schematic of a CFB steam generator. This particular boiler is the newest and one of the largest in the world. Power generation from the ST is 240 MW. Another 100 MW$_{th}$ are supplied as process steam and 60 MW$_{th}$ as district heat. Fuel input is 580 MW. (from [2.1])*

A major advantage of a CFB is its ability to handle varying feedstock with different compositions and moisture content, also the possibility for SOx capture with very high efficiency by adding limestone directly in the furnace.

2.3.1.6. Cofiring

Cofiring means simple co-combustion of biofuels as a supplementary fuel to coal in existing coal fired boilers. The interest in biofuel cofiring varies in different countries. Usually interest in this method of biomass utilization exists in countries with well established coal fired power generation. A perfect example is USA. Research and development work on cofiring has been given almost total priority in USA among all other biofuel energy utilization technologies for both biomass and MSW.

Cofiring in coal fired boilers is the simplest method of biofuel energy utilization without investments in new facilities. This is its main attractiveness. Direct reduction of CO_2, SOx and NOx emissions can be achieved in existing power generation facilities, using

the CO_2 neutrality features of biomass and its much lower content of fuel-bound sulphur and nitrogen. Moreover, cofiring in large-scale high-efficiency coal-fired steam cycles increases the efficiency of biomass energy utilization itself [2.26].
However, caution must be taken to verify the limit to which coal can be substituted with biomass in a boiler designed specifically for coal. This has been one of the main topics of research and experimental work done on cofiring.

In USA, a number of utilities have instituted commercial cofiring operations or have performed commercial scale cofiring tests in utility boilers. These operations have covered the majority of types of utility boilers that are expecting to be candidates for cofiring – different pulverized coal units and FB units. Some US utilities routinely co-fire several hundred thousand tons of biomass every year (wood, grass and agricultural by-products). Tests have been performed mostly at a 10%wt coal substitution by biomass. The significant technical, economical and environmental benefits associated with wood cofiring have been demonstrated, achieving 100% boiler capacity during cofiring with minimum impact on boiler efficiency. Most of the tests have been performed at low levels of cofiring – 4-5%wt biomass to coal. [2.3], [2.57]
Some general conclusions have been reached:
For low levels of cofiring (5-8% biomass), wood can be combined with coal prior to the pulverizers. At medium levels (10-15% biomass), a separate wood preparation and delivery system should be provided, while boiler operation is based on coal characteristics. At moderately high levels (25-50% biomass), a multifuel system such as FB should be used. At very high levels (50% biomass and more), a boiler designed specifically for biomass operation should be used [2.3].

Cofiring of unprocessed MSW is not desired in a coal-fired boiler. MSW has been cofired in some extent only in the form of RDF [2.50]. Of course, cofiring of biomass or coal (also gaseous and liquid fuels) in a boiler designed for MSW is always possible.

Straw can also be co-fired with coal. However, changes must be made to prevent fouling and quick erosion of the boiler surfaces, due to the specific ash composition of straw. After solving these problems, a CHP plant in Denmark has started to substitute more than 40% of its coal fuel with straw and other agricultural residues [2.1], [2.57].

Cofiring of natural gas as a supplementary fuel in MSW incinerators is a routine process in many installations. It has been recognized, that firing of natural gas with wastes is one easy way to achieve combustion control, lower levels of organic toxins from MSW incineration, higher boiler performance and load flexibility. Two basic concepts have been devised, gas cofiring and gas reburning. Both aim mainly at reducing the generation of toxic compounds and other pollutant emissions in MSW incinerators by providing suitable conditions in the boiler. Gas cofiring has been extensively employed in Europe and is in fact required in Germany as part of the permit conditions for MSW incineration. [2.36], [2.37], [2.50]

Cofiring has one major disadvantage. Biomass ash is mixed with ash from coal or wastes and cannot be recirculated back to nature. Returning biomass ash back to the fields and forests is one important prerequisite for reaching sustainability of biomass production as a renewable energy source.

2.3.2. Upgrading to gaseous fuels

The methods for producing gaseous fuels out of biomass can be summarized as: Gasification, Biogas production (anaerobic digestion) and Methane Synthesis.

2.3.2.1. Gasification

Production of a gaseous fuel out of a solid fuel is a well-known technology. An enormous amount of research and development work is still being invested in gasification process improvement and component design for all kinds of solid fuels. The technology is ready for commercial implementation and many gasification plants of all scales already exist all over the world. [9.9]

Biomass gasification has sparked ever-increasing interest as a way to produce gaseous fuel out of CO_2-neutral and readily available raw material, both in developed and developing countries. The technology is put into service mainly in small scale and various types of gasifier designs have been developed. Its advantages are many. First of all, the possibility to utilize biomass with high efficiency in gas turbines and internal combustion engines must be mentioned. Utilization of coal and biomass at very high electrical and total efficiencies with IGCC is expected, where a combined cycle is powered solely by a solid fuel.

There are many books, publications and materials on gasification. However, it is not a topic for the present Literature Study and for the research project that follows. Moreover, one of the main purposes of the present research project is to investigate an alternative to gasification by showing that biomass energy can be converted to power in an easy way with a satisfactory efficiency without the need for a complex gasification process.
The gasification process uses the partial oxidation of carbon to produce low-heating value carbon monoxide (CO) gas. Partial oxidation of carbon to CO occurs at high temperatures in extreme substoichiometric conditions. The combustible part of the product gas consists of carbon monoxide, hydrogen and small amount of methane. During the process, a certain quantity of the carbon in the raw fuel is fully oxidised to CO_2 and lost. The heat of full oxidation to CO_2 supports the process (or can be recovered if it's in excess), but a part of the fuel is not upgraded and its energy cannot be utilized in a combined cycle at high efficiency. The exact composition of the product gas and its heating value depend on what kind of oxidizing medium is used – air, pure oxygen or steam. Air gasification is the cheapest and the most common way to gasify biomass. Because of the large amount of nitrogen in air, the product gas is diluted and has a very low heating value, of the order of 5 MJ/kg. Pure oxygen or steam gasification produces a gas with a heating value of around or more than 10 MJ/kg.

Gasification of biomass features several significant problems that have not been overcome to a full extent or require large investments for additional equipment. At typical gasifier conditions, tars and volatile alkali are generated. They are harmful for the hot-gas-path equipment and especially for gas turbine blading. Tars must be either directly burned in gas turbine combustors without cooling the product gas below their dew point (typically about 538^oC [2.3]), or hot-gas cleanup systems should be installed. Hot-gas cleanup is expensive and unreliable, while direct burning causes soot formation and combustion instabilities in turbine combustors. If the product gas is generated at low pressures and needs to be compressed, care must be taken not to allow condensation in the compressor. Some ingredients of the product gas may be harmful also to the lubrication oil. Ammonia is also formed in considerable quantities. It causes problems with handling and combustion of the product gas. [9.9]

Gasification of MSW produces a whole cocktail of dangerous substances, both organic and inorganic, that are part of the product gas. Many of them are toxic. Reason for this is the inevitable presence of certain chemical elements in MSW (no matter how well it is sorted) either as poisonous chemical wastes or as ingredients of plastics. Sophisticated gas-scrubbing equipment must be used if a MSW product gas is to be burned in internal combustion engines or gas turbines. An additional problem is the variation in MSW characteristics with time.
Direct combustion of MSW is to be preferred to its gasification. Nevertheless, new technologies for energy utilization from MSW, incorporating pyrolysis and gasification processes have been developed. Two such technologies have been implemented in commercial plants in Germany [4.14].

In general, the gasification process is expensive, difficult to control and unstable at part-loads. The expected very high efficiencies of the IGCC proved to be technically or economically hard to achieve.

2.3.2.2. Biogas

Biogas consists mostly of methane (CH_4) and CO_2. Water vapour and some other organic and inorganic gaseous substances are also present in variable quantities. CH_4 content reaches 60%v, which makes biogas valuable for all kinds of gaseous fuel applications, with a heating value about half that of natural gas.

The production of biogas is a natural biological process. It is a result of the so-called “anaerobic digestion” of organic substances by certain microorganisms. Anaerobic digestion occurs in an oxygen-free environment.
A controlled digestion of organic residues in a reaction containment of various designs (digester tank) is the core of biogas production. The process maintenance itself is extremely simple – the only requirement is that a suitable temperature is kept in the digester vessel so that the microorganisms can thrive and produce at the highest possible rate. The technology is easily applicable at all scales. Biogas production units have been used since long time both in developed and developing countries.
Raw materials for biogas production are in general all kinds of organic wastes, such as animal manure, residues from slaughterhouses and food processing industries, organic sludge and municipal sewage waters. Best production rates give residues from fish processing industry and chicken or cow manure. [2.55]

Hard wood can also be used for direct biogas production (biological gasification), although with longer retention times in the digester vessel and smaller yields [2.11].

The most commonly used disposal method for MSW today, dumping in landfills, is in fact a method for biogas production. A natural anaerobic digestion of the organic part of the wastes starts inside the encapsulated landfill, which produces so-called “landfill gas", which has CH_4 content close to that of digester-produced biogas. Gas production in landfills lasts usually around 20 years [2.23]. Furthermore, utilization of the landfill gas is to be preferred to just letting the gas escape in the atmosphere, which always happens slowly but inevitably, if the gas is not collected. Its high content of methane adds significantly to the greenhouse effect and the presence of other organic substances causes bad odour and toxicity in the region. [2.19], [2.24]

Attempts have been made biogas from MSW to be directly produced in anaerobic digesters. The process is technically and economically viable, but has proved to be unstable and the production has been comparatively low. [2.6]
In Sweden, it has been estimated that sorting and handling organic MSW for direct biogas production involves higher expenses than revenues from acquired products, so the efforts are not justified [2.13].

More information on biogas, landfill gas and natural gas composition and properties can be found in Svenskt Gastekniskt Center's reports and the references thereafter (for example [2.52]).

2.3.2.3. Methane Synthesis

The synthesis of methane involves the so called "methanation reaction" – a chemical reaction between carbon monoxide and hydrogen (with the help of a catalyst) to form gaseous methane.
The overall process starts with oxygen-blown gasification of the raw material (the solid fuel). The product gas is a high energy value mixture of CO, CO_2, H_2, water vapours and small traces of other elements. The use of pure oxygen for gasification ensures no presence of nitrogen. Then the product gas must be enriched with hydrogen to the required molar ratio CO/H_2 for the methanation reaction to proceed. One easy way of producing hydrogen out of CO is the standard "water-shift" reaction – a chemical reaction between CO and water vapours at certain conditions, which produce CO_2 and pure H_2. A certain part of the CO in the product gas is used to produce H_2.

water-shift reaction: $CO + H_2O \Rightarrow CO_2 + H_2$

The gas after the water-shift reaction is stripped out of the CO_2 and fed to the methanation reactor vessel, where on the surface of the catalyst the CO and H_2 react to form CH_4 and water. The CH_4 is scrubbed out, dried and fed to a pipeline or storage. It is almost pure methane (with only small impurities) and its LHV is very close to that of natural gas.

methanation reaction: $CO + 3H_2 \Rightarrow CH_4 + H_2O + \text{heat}$

The process is complicated, but the equipment and reactions involved are well-studied and developed and are technically feasible in industrial scales. The economics of such a process are uncertain, but if a cheap raw material is used and a stable market with good prices for the final product is available, the overall process can be very well economically justified. The overall efficiency of conversion is low, but the resultant final product is a high-grade fuel, which can be utilized with a high efficiency for all energy purposes. A large-scale pilot plant producing synthetic methane from coal has been build and operated with good economy in USA [9.8].

2.3.3. Upgrading to liquid fuels

Producing liquid fuels out of solid ones has always been an attractive research area.
Liquid fuels are perfect for all combustion applications, but most of all as an energy source for transportation (better energy-per-volume and energy-per-mass ratio, easy handling and storage) and consequently are the most expensive ones. Methods for upgrading biomass or coal to liquid fuels do exist and are technically feasible, although in general not economically feasible yet. These methods are regarded as the future way for humankind to supply itself with high-grade fuels from biomass and coal, after resources of oil and natural gas are exhausted and economics of solid fuel liquefaction are more favourable.
The basic research and development work on coal and biomass liquefaction methods has been performed in the first half of the 20th century, between and during the two world wars. That was the time of big advances and expansion in industrial chemistry and the conditions were ripe for the first quest for alternative fuels to take place. The second peak of interest in liquefied coal and biomass occurred after the oil prices sharply soared in the 1970-ies. Nowadays, apart from the necessity of oil substitution, environmental considerations play a leading role in the attempts to utilize biomass in the form of liquid transportation fuel.
Biomass liquefaction processes can be generally divided into thermochemical and biological. The thermochemical processes described briefly in this subsection (or parts of them) are widely used in the chemical industry for production of certain organic gases and liquids. They are part of the standard organic chemical synthesis.

2.3.3.1. Fast pyrolysis

Fast pyrolysis can be regarded as the simplest of all biomass liquefaction methods.
The process resembles the well-known wood pyrolysis for production of charcoal. Fast pyrolysis incorporates heating of the raw feedstock in the absence of air at a very quick rate (as the name suggests) and under a special temperature regime up to around 500^{o}C. The stress during fast pyrolysis is put on extraction of organic substances from biomass in volatile form. The residues from the process are char and non-condensable gases, which can still be utilized as fuels and are usually used within the pyrolysis process for heating.
In the common slow pyrolysis process (production of charcoal), the temperature regime is such that it maximises the char production by simply devolatilizing the wood.
During fast pyrolysis, the long chains of carbon molecules in the biomass structure crack down and certain part of the organic substances undergo chemical changes, are volatilised and entrained into the product gas. A very small quantity of the feedstock ends up as residual char. The main product, called “bio-oil” or “pyrolysis oil”, is obtained in yields of up to 80%wt on dry feed basis.
The product gas after the pyrolysis is rapidly cooled and condensed. The final product, condensed pyrolysis-oil, is a dark brown liquid with viscosity close to that of medium fuel oil and less than half of its LHV. It is composed of organic substances with complex molecules, which are in liquid phase at normal temperatures, but non-stable

at higher temperatures (it cannot be distilled). Pyrolysis-oil does not readily mix with fossil hydrocarbon fuels. Its elemental structure and LHV are close to that of the raw biomass, with oxygen content around 40%. [2.8] [2.15] [2.33]

Fast pyrolysis is easy to implement at any scale and is one of the most efficient of all biomass conversion processes. The quality of the bio-oil however makes it unsuitable for transportation fuel applications.
Pyrolysis oil can be directly utilized in stationary diesels or as boiler and gas turbine fuel. Gas turbine combustors have been tested on such bio-oils and the results have been encouraging, however special arrangements are necessary to accommodate the very high viscosity of the fuel [2.32].

Post-treatment of pyrolysis oil can be implemented for its transformation into high-quality hydrocarbons. One such process is the hydrodeoxygenation / hydrocracking with hydrogen gas (on catalyst surface) for oxygen removal and molecular weight reduction, which yields gasoline-range hydrocarbons. [2.17]
Another option for direct production of high-quality hydrocarbons from pyrolysis is to perform the pyrolysis process under pressure in hydrogen atmosphere with dispersed catalyst (hydropyrolysis). [2.49]
Experiments have been made also with a mixed gasification-pyrolysis process for the production of high energy value low molecular weight gases and liquids, together with activated carbon as char residue. The process is performed in a reactive atmosphere of steam and carbon dioxide or in inert argon atmosphere [2.41].

2.3.3.2. Solvolysis

Solvolysis, followed by catalytic hydrotreatment, is another thermochemical method, aiming at the direct production of liquid hydrocarbons of higher quality than the pyrolysis oil. The solvolysis process comprises simple dissolving of the wood in acidified organic solvents, at certain temperature and pressure. After that the solvent is evaporated under vacuum and recycled. Water generated during the solvolysis is also evaporated. The solvolysis product is very viscous and has oxygen content of about 22%. It is then subjected to catalytic hydrotreatment (hydrodeoxygenation) at about 350^{o}-$370^{o}C$ under pressurised hydrogen gas atmosphere. The final product is composed of a mixture of hydrocarbons with very low oxygen content, the heavy fractions prevailing. [2.48]

Wood solvolysis is used also for production of chemical feedstock and as a novel pulp extraction method. Lignin remains as a residue. Hydrocracking of residual lignin from solvolysis and pulp mills is possible, which can produce valuable hydrocarbons from residues of large-scale industries [2.56].

2.3.3.3. Methanol production

Methanol can be produced from biomass with a thermochemical process similar to that of methane synthesis, mentioned above. Methanol is the lightest alcohol, poisonous for the organic tissues. It is an important raw material for many industrial

chemical processes and its production from fossil fuels or biomass is a widely employed method.
If biomass is gasified in an oxygen-blown reactor and then part of the CO is converted to H_2 by water-shift reaction, the resultant mixture (after scrubbing away impurities) can be forced under certain conditions (and with the help of a catalyst) to undergo a conversion to methanol, CH_3OH. This is another typical example of organic synthesis.

methanol synthesis: $CO + 2H_2 \Rightarrow CH_3OH + heat$
$CO_2 + 3H_2 \Rightarrow CH_3OH + H_2O + heat$

If hydrogen from additional source is added to the process, so that carbon is not sacrificed to produce hydrogen, the methanol yield will be more than doubled. [2.16]

Methanol is readily applicable as fuel for ICE and in any other combustion processes. Because of the presence of an oxygen atom in its molecule, its LHV per mass and per volume is lower than that of the common fossil liquid fuels. However, the combustion of alcohols in general (methanol and ethanol) emits fewer pollutants than any other liquid fuel. The thermal efficiency of engines burning alcohols is slightly higher than for standard fuels. Combustion of pure alcohol requires modification of the engine, but mixture of gasoline with small percentage of alcohol can be readily utilized in standard car engines.

Furthermore, methanol molecules can be forced to react with each other, forming ethers, out of which synthetic gasoline and any hydrocarbons can be produced. The process is complicated, but its chemistry is well-studied and ready to be implemented on industrial scale. In fact, a large-scale commercial plant for synthetic gasoline production out of natural gas has been constructed in New Zealand. An offshore field supplies raw natural gas to the plant, which is steam-reformed to produce CO and H_2, from which methanol is synthesised. Then gasoline is synthesised out of the methanol. Another smaller plant has been built in Germany. [9.3]

2.3.3.4. Ethanol production

Methane and methanol synthesis involve forced chemical reactions to convert organic products into inorganic ones (gasification) and then back into new organic ones through complex conversion processes. Ethanol can also be synthesised out of inorganic substances, theoretically.

In sharp contrast with synthetic methods, ethanol can be produced by a purely natural biological process – fermentation. Just like the anaerobic digestion, fermentation is one of the basic biological processes in nature, which is the result from the life cycle of tiny microorganisms. In water medium and presence of air (aerobic process), these microorgaisms convert the sugars or starch (complex sugars) contained in certain plants into alcohols, dissolved in the water. In the mixture of alcohols, ethanol (C_2H_5OH) is the main product, but small amount of methanol (and possibly also higher alcohols) is also present.
The process of fermentation is well-known to humankind since the dawn of civilization. Not any organic matter however can undergo fermentation. It must contain sugars. After fermentation is complete (after all sugar is consumed by the microorganisms) the

product must be separated out of its water solution by certain methods. Alcohols mix with water freely in any ratio and a distillation process must be used to separate them. Distillation is also a well-known process. It is highly energy consuming however, and decreases the overall efficiency of ethanol energy conversion.

Apart from being the important constituent of all alcoholic beverages, ethanol is also important material for the chemical industry. Large commercial plants for production of industrial ethanol exist in many places. The raw materials are usually residues from the sugar extraction industry or any kind of cereals (grain).
Like methanol, ethanol is a high-quality fuel. Mixed with gasoline in small ratios, it can be utilized in gasoline engines without any modifications (so called “gasohol” fuel), leading to less pollutant emissions. It also increases the octane number of gasoline.
Alcohols can be easily burned in gas turbine combustors. Dry low-NOx combustion chambers can probably be designed for alcohols with fewer difficulties than for any other liquid fuel.

A point worth mentioning is the ethanol fuel program in Brazil. Since many years, a large part of harvested sugar cane in Brazil is used for ethanol production through fermentation. Of course, the process economy depends on international sugar prices and certain governmental subsidies. The overall result is very promising, Brazilian ethanol fuel program has been highly successful. Its main incentives have been reducing the dependence on imported oil and finding an alternative market for sugar cane overproduction, but it has proved that a renewable domestic fuel can be produced and used on large scale in an economically viable way. A considerable percentage of the entire light-vehicle fleet in Brazil runs on ethanol. Consequently, this has resulted in a huge positive impact on the environment and is the only example for a widespread use of biofuels for transportation in the world today. [2.43]

Ethanol can be produced also from hardwood biomass, either with a thermochemical process (acid hydrolysis) or with a biological process (enzymatic hydrolysis). Hydrolysis has the purpose to break down the complex molecules of cellulose in biomass (decomposition into fragments) and hydrolyse them into sugars (sugar syrup), which can be subjected subsequently to fermentation and ethanol extraction, as described above. Acid hydrolysis is a well-developed method, but produces polluting effluents and has low efficiency and high costs.
Enzymatic hydrolysis (bacterial hydrolysis) is a purely biological process and is a comparatively new method, still under development. The process starts with pre-treatment of the biomass particles, in order to facilitate easier access of the microorganisms to the cellulose. Pre-treatment methods can be summarized under the following names: alkaline maceration, steam explosion, ammonia fiber explosion and autohydrolysis. Lignin and hemicellulose-derived sugars are not fermentable and can be burned or sold as chemical feedstock. MSW can also be liquefied in this way. Pre-treatment and especially the production of enzymes (bacterial cultures) are the most energy consuming and expensive sub-processes, which hinder the application of the method other than on experimental scale. Finding effective and cheaper pre-treatment and enzyme production methods will make microbiological liquefaction of biomass a viable alternative. [2.62], [2.54], [2.14], [2.20]

In Sweden, the costs for ethanol production with any method and raw feedstock are estimated to be higher than that for methanol. The reduction of CO_2 emissions by using ethanol as transportation fuel substitute is lower than that obtainable from methanol [2.22] (see also section 2.4.).

2.3.3.5. Plant Oils (Vegetable Oils)

Plant oils (vegetable oils) are important part of our food supplies. They are naturally produced by many plants and simple extraction methods (pressing or leaching with solvents) are sufficient for their production.

Plant oils are perfect fuels. Their heating value is only slightly less than that of fossil liquid fuels. They can be burned without any modifications in boilers or gas turbine combustors designed for heavy fossil oil fuels. With slight modifications (to accommodate the higher viscosity of plant oils), diesel ICE can directly utilize them as transportation fuels. The ability of diesel engines to operate on vegetable oils was suggested by Rudolf Diesel himself [2.12].
After certain processing (trans-esterification with the help of methanol, to lower the viscosity and push combustion characteristics closer to that of diesel fuel), plant oils can be transformed into methyl-esters, which can directly be used as diesel fuel without any engine modifications. Plant oils mix with hydrocarbons in any ratio.
Production of plant oils for fuel use is expensive. They are also a raw material for the food processing, cosmetic and other chemical industries. However, if regarded as agricultural residues whose production is higher than the food and industrial market consumes, they can be cheap enough to be used as engine fuels. Such is the case with rapeseed oil in Northern Europe, which in many cases is considered a residue from the production of protein-rich fodder for livestock [2.61].

2.3.3.6. Slurries (suspensions) of solid fuels in water or oil

Attempts to transform solid fuels into liquid fuels by simple pulverizing and mixing with water date back to the end of the 19th century. Rudolf Diesel himself has stated that coal-water slurry could be a possible alternative for use in his newly-invented engine.
A lot of research work and experimental tests have been performed on coal-water slurries as fuels for boilers and diesel engines. Results have been generally quite positive [5.22], [5.15]. Combustion tests with biomass-water emulsions as fuels however have not been reported. Wood is more difficult to pulverize and handle due to its fibrous structure and high volatility (easy flammability). Also, biomass heating value is too low to ensure stable combustion if humidity is high, but has the advantage of lower ash and lower sulphur content than coal. Slurries of agricultural products in water have been suggested [2.44]. Slurries of char in water may have good combustion characteristics, but no such experiments are reported.

Suspension of wood powder in oil is a more viable alternative. For example, a mixture of wood particles with diesel or kerosene has been suggested [2.4]. The wood particles have been pre-treated in a special process for washing, autohydrolysis and drying. Care must be taken to prevent settling of the particles if slurries are to be stored for long time, which requires constant stirring or stabilization with additives. Slurry fuels of residual lignin from pulp mills in fuel oil and water have also been suggested [2.47]. Suspension of char in pyrolysis oil (both products of biomass pyrolysis process), with presence of heavy fossil fuel oil, has been tested in boiler combustors. The production of slurry fuel from RDF has been devised and combustion experiments with it have been performed as cofiring in coal-fired boilers [2.39].

2.4. Biofuel Handling and Preparation. Life Cycle Analysis.

One of the main issues in biomass energy technologies is fuel handling. This is connected to some typical features of biomass production and utilization – non-concentrated resource base, intermittent production, transportation distances, fuel processing difficulties and problems with fuel storage.

Biomass resources are distributed over large land areas and their collection for fuel use is labour and power demanding. Transportation is a critical part of the fuel management. The same applies also to MSW. Although being collected only in urban areas, MSW collection vehicles travel long distance in order to deliver a certain amount of energy, compared to bulk transportation of coal for example.
Having in mind also the low power density of biomass fuels (low calorific value per unit mass and per unit volume) and long haulage distances (especially for forest residue-supplied large power plants), the power input for transportation and the problems and uncertainties associated with fuel delivery have been addressed in various studies, both economic and technical. Extending the analysis to accommodate all energy input for power plant construction and maintenance (with possible demolition in the end of service) and waste disposal leads to the so-called "life cycle analysis" or "life cycle approach" or "life cycle assessment". The main stress in such analysis is on the fact that all power input for biomass fuel delivery and utility construction is based on fossil fuels. This of course asks for a thorough investigation of the actual extent to which, for example, biomass utilization as energy source reduces pollutant emissions (net emissions reduction). Or in simple words, investigation is necessary in order to find out how much of the biomass-based energy production is really based on biomass.

This has been the topic of numerous studies, reports and publications. Several of them are included in the reference list. In general, the studies have lead to the conclusion that biomass utilization for energy purposes cuts CO_2 emissions close to 100%, although estimations of fossil fuel input for component production and utility construction are very difficult to perform and data is rarely accurate. Fossil fuel input for transportation has been found to be a negligible part of the energy value of biomass, of the order of several percent, though an important part of biomass cost. In precise figures, fossil energy input for biofuel transportation is generally around or less than 3% of the energy value of the biofuel, while transportation costs correspond to around 25% of the cost of the biofuel, delivered at the end-user site.
One interesting object of profound studies are the energy crops, inspired by growing expectations that crops grown specifically for fuel-use will increase in the future. It is expected that yields will grow and the input of energy will be lowered with time. Currently, the input of fossil fuels and energy in general into the production and harvesting of energy crops is high (depending on the type of crops), higher than the transportation energy (unless distances are long), but still in the order of only several percent of the energy available in the biomass produced. Most of the life-cycle and transportation studies are devoted to energy crops, including also forest residues and agricultural residues as a comparison. Energy input is expected to steadily decrease

with increasing efficiency of vehicle engines and better solution to logistic challenges. [2.9], [2.10], [2.22], [2.18], [2.2], [2.5], [2.40], [2.51], [2.58], [9.6].

Table 2.3: *Estimated energy yields, energy inputs and net energy yields for energy crops and logging residues in Sweden (adapted from [2.22], 1995)*

Biomass resource	Energy yield MWh/ha,year	Energy input MWh/ha, year	Net energy yield MWh/ha, year
Rapeseed	18	3.5	15
Winter wheat, grain	21	3.4	18
Reed canary-grass	30	2.3	28
Lucerne	32	1.8	30
Salix (short rotation coppice)	42	1.6	40
Straw from food production	2	0.17	1.8
Logging residues	1.1	0.04	1.1

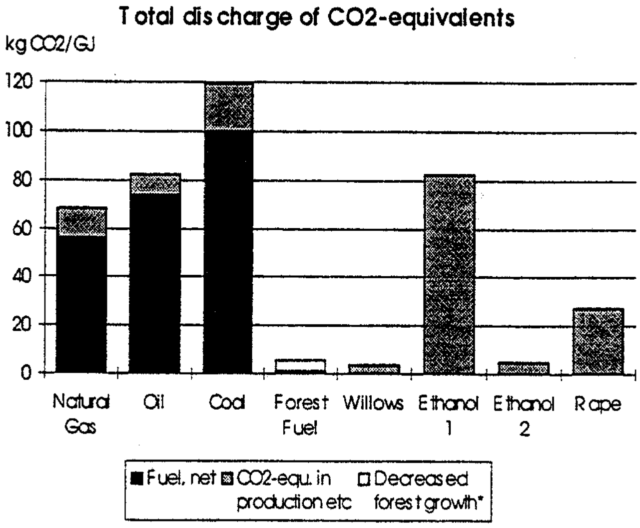


Fig. 2.10: *Total discharge of CO_2 per unit energy from several biomass-derived fuels, compared to three most common fossil fuels. Biomass fuels have no net CO_2 emissions, but energy input for their production can be high. Energy input for fossil fuel production is also high.*
Ethanol 1 is produced from grain, with fossil fuel input. Ethanol 2 is derived from forest residues, with energy input only in the form of waste heat.
**Decreased forest growth can be counteracted by fertilization. Return of ashes is also recommended. (from [2.5])*

It has been estimated that fossil fuel use for biomass production, transportation and facilities construction contributes around 5% of the total CO_2 emissions from a given biomass power unit. This leads to the conclusion that biomass utilization for energy purposes provides 95% net reduction of CO_2 emissions [2.40].
Energy input for biomass transportation by truck in Swedish conditions is estimated to be around 1.4 MJ $ton^{-1}km^{-1}$ [2.9], [2.10] (1996).

Other studies have been devoted to the so-called "social costs" of increased biomass use, or to the necessary resource base management, if larger biomass energy utilization have to be implemented in the future. Examples are [2.42], [2.63]. Careful estimation of the practical feasibility of biomass fuel delivery is often overlooked. This is a very important issue, especially in the case of large centralized power plants. Underestimating the problems can result in wrong conclusions and false planning.

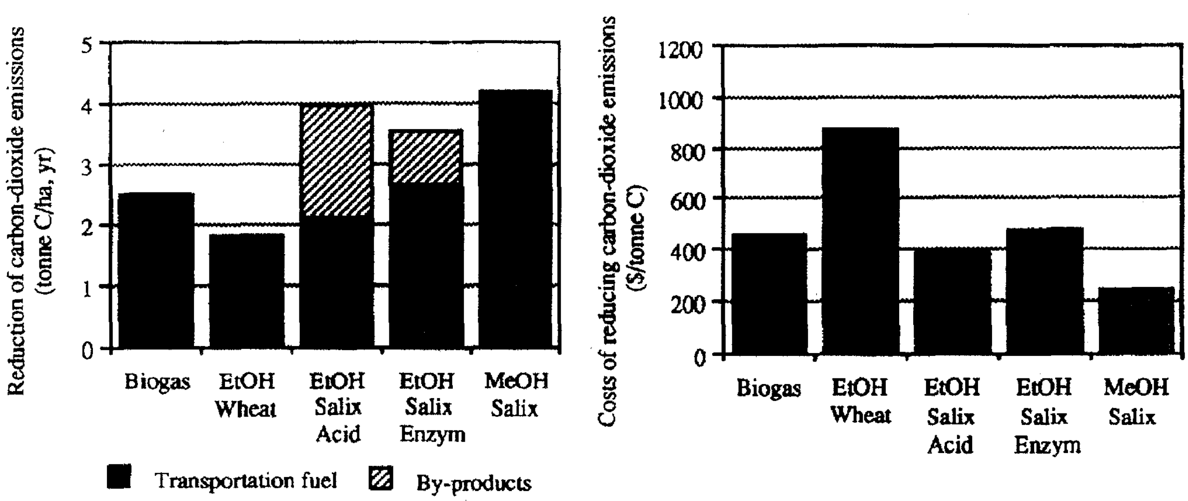


Fig. 2.11: *Reduction of CO_2 emissions and costs for this reduction when gasoline in light-duty vehicles is replaced by biomass-derived transportation fuels in Sweden. The by-products are also used for energy purposes, together with straw from wheat. (from [2.22], 1995)*

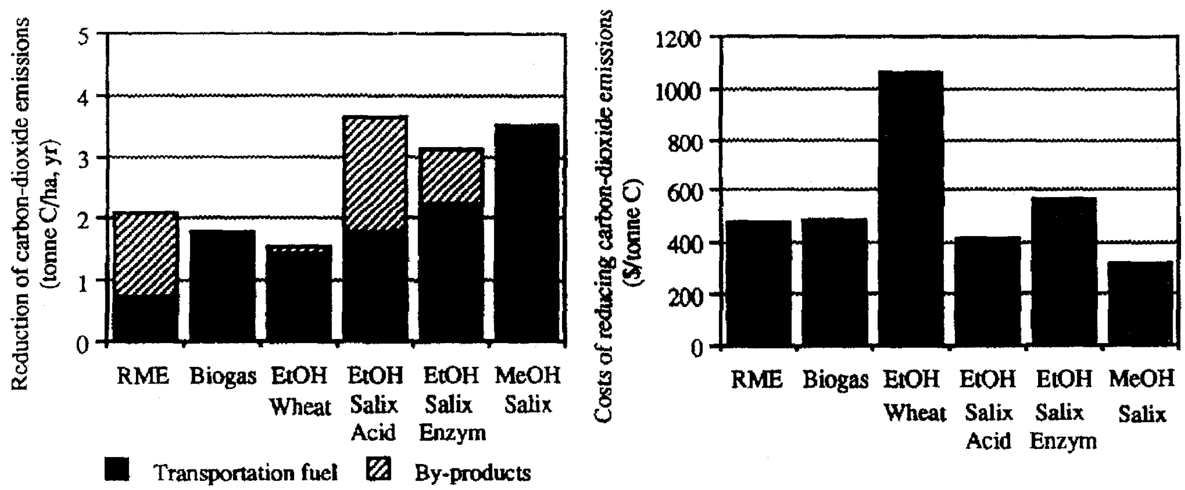


Fig. 2.12: *Reduction of CO_2 emissions and costs for this reduction when diesel in heavy-duty vehicles is replaced by biomass-derived transportation fuels in Sweden. The by-products are also used for energy purposes, together with straw from wheat. (from [2.22], 1995)*

Storage of biomass is another critical issue. Usually biofuels have to be stored for certain periods before burning them, sometimes quite long. Biomass properties can change considerably with time, depending on the conditions and the initial stage. Two important properties of biomass can change during storage – its moisture content and its heating value. As was already pointed out in previous sections, moisture adversely affects the heating value of fuels. Storage of biomass for a certain time leads to a decrease of its moisture content, even if it is placed out in the open. Decreasing moisture content means increasing LHV of the stored fuel.
However, placing wet biomass in a pile causes another problem. Wood, being a raw organic material, supports fungal growth on its surface and emits sponges in the

atmosphere, which deteriorate its properties and can cause health hazards. The microbiological activity dissipates heat (or in other words, transforms some of the energy of the wood into low-temperature unusable heat). This heating process proceeds together with gradual slow devolatilization of wood, which causes a risk of fire in the pile. Self-ignition in stored piles of biomass (also coal) is possible.

Many studies have been devoted to change of characteristics and to problems arising during wood fuel storage, especially in bulk piles. In general, microbiological activity in the pile increases its temperature and wastes the energy content of the biomass, while moisture decreases with time to a certain steady value. These processes depend on the type of biomass and on the size of the particles – the smaller the particles, the higher the rate. The combined effect of these processes leads to increase in the LHV of a unit mass of fuel (due to drying out), but the total energy content of the stored pile decreases (due to the loss of mass and energy by microbiological conversion and devolatilization). If these effects can be neglected and risk of fire is diminished, outside storage of biomass is possible and is often attempted. [2.59], [2.30], [2.35], [2.34]

MSW must not be stored for long time in the open. Apart from emitting bad odour, it sustains microbiological processes, similar to that in biomass piles, but much more pronounced. Conversion of the available organic material in the MSW pile (by aerobic rotting or anaerobic digestion) proceeds with a high rate and energy “escapes” in the form of dissipated heat and released gases.

In order to reduce the risks and losses of energy during storage, biomass fuels can be processed in a certain way. This reduces their moisture content (increases the LHV per unit mass) and makes their handling and long-term storage easier, without any degradation of the fuel.
Processed biomass fuels are quite common for heating purposes, these are the wood pellets and wood briquettes. Pellets and briquettes are produced by chipping and compressing the wood particles at slightly elevated temperature. The wood is first dried to very low moisture content. During the compression process, the wood particles partially devolatilize, part of the organics melt and the result is that the particles glue together steadily. Comparatively small part of the energy in the biomass is used for this process. Wood pellets or briquettes are dry upgraded solid biomass fuels, with no restrictions about storage and handling.
Producing RDF out of MSW is a similar way to upgrade the wastes, so that storage, handling and combustion are easier. Usually MSW is chopped, metal particles and other inert materials are separated as much as possible and lime particles are mixed in, then the whole mass is compressed and pelletized.

A good overview of the problems connected with storage and processing of wood fuel can be found in the book by Lehtikangas [2.34].

Long term outside storage of biomass is possible, but often biofuels (especially MSW) are burned soon after they have been supplied. In the case of fresh forest residues, energy crops or certain agricultural residues, chipped and transported right after they have been harvested (including MSW), the fuel contains large amount of moisture. Moisture content of 50-60% is a usual value for raw wood. MSW usually has moisture contents of around 20-40%.

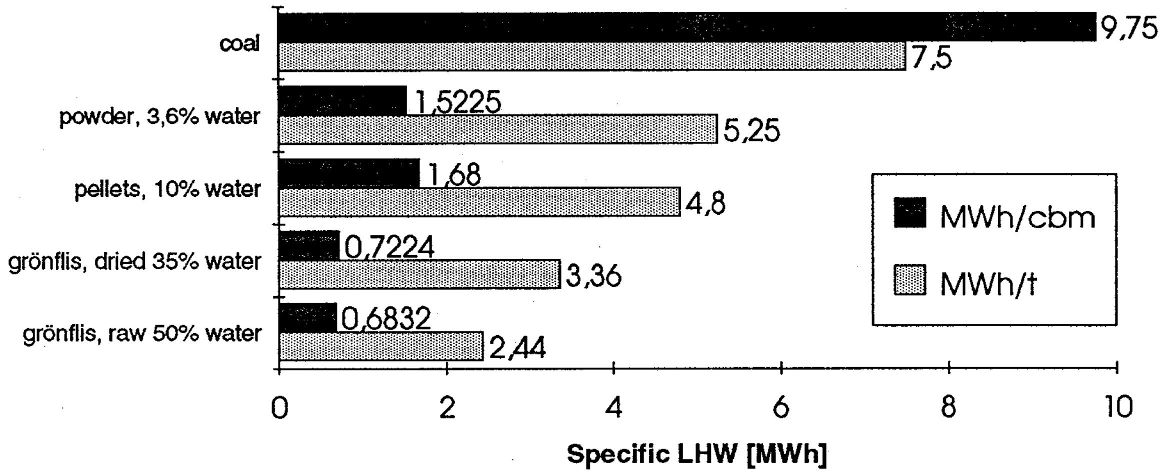


Fig. 2.13: *A comparison between LHV per m^3 and per ton of several processed biomass fuels (wood powder, pellets, woodchips with two different moisture contents) and coal. (from [4.4])*

Moist biofuels are often directly introduced into the combustion chamber of the boiler. This has a negative effect on the combustion process, boiler efficiency and efficiency of the whole power cycle. In simple small-scale wood combustion, boiler efficiency is believed to improve with 1% for every 1% drop in moisture content of the fuel, starting from the raw moisture values. When wood moisture content falls to 40%, boiler efficiency improves with 0.5% for every 1% further loss of moisture of the fuel [2.35].

Drying of biofuels before combustion leads to a more stable combustion process, less emissions from incomplete combustion (less CO, UHC and VOC), improved radiative heat transfer in the combustion chamber, lower volume of flue gases and less heat carried away by non-condensed water in flue gases. The thermal energy involved in evaporating the moisture entering the combustion chamber with the fuel is not lost, it is contained in the form of condensable water vapour in the flue gases. However, this thermal energy can be utilized only at low temperature levels (by flue gas condensation), provided also that suitable heat sinks are at all available.
A separate work within the present project, a Master of Science Thesis titled "Drying of Biofuels for Energy Purposes", takes a closer look at the problems associated with moist biofuels and the possibilities for drying them before combustion.

3. HISTORY AND DEVELOPMENT OF HYBRID COMBINED CYCLES. HCC WITH COAL-FIRED BOTTOMING CYCLE.

3.1. Topping cycles and fuels

The term "Topping Cycle" (TC) addresses the power cycle of any heat engine, which accepts heat at very high temperature level and whose remaining exhaust heat is utilized by another cycle at a lower temperature level. Typical examples for TC heat engines are the gas turbines and the internal combustion engines (ICE). Both are well developed today and are widely used for all kinds of prime movers' applications, including power generation.

TC engines can easily utilize any gaseous or liquid fuel. Solid fuels are much more problematic for GT and ICE. The presence of ash particles and unstable combustion are the most common problems that impede application of solid fuels in GT or ICE. Many attempts are made to utilize solid fuels in TC cycles, and several technologies are ready for (or close to) commercialisation. These novel technologies include IGCC, PFBC, pressurised cyclone combustion, externally-fired GT, coal-water slurry for ICE and various methods for producing liquid fuels out of solid ones.
Nevertheless, common gaseous and liquid fuels are still the ones mostly used for TC.

Open cycle GT and ICE will be considered as TC engines in the cycle simulations for the present research project. The most common gaseous fuel for power applications, the natural gas, will be the one under consideration. Natural gas is also the most "environmentally-friendly" fuel among the fossil ones, with lowest CO_2 emissions and close to zero SOx emissions. It is comparatively easy to handle and burn, with very low power consumption for fuel preparation. Modern combustors for natural gas give extremely low emissions of NOx, CO and UHC.
Biogas can replace the natural gas as TC fuel, if total CO_2 neutrality is sought. The composition of biogas is close to that of natural gas in terms of elements present, the only major difference is that the CH_4 content is about half that of natural gas. Some modifications are required for switching from natural gas to biogas in GT and ICE combustion chambers. The volumetric fuel flow will be doubled.

3.2. Bottoming cycles and fuels

The term "Bottoming Cycle" (BC) refers to any power cycle, whose heat addition is being implemented by heat rejected from another power cycle through a heat exchanger. The temperature of heat input to the BC (when viewed as a simple cycle) is limited by the thermal qualities of materials available for construction of the heat exchanger. The BC itself rejects heat at the lowest possible temperature level.
A typical example of a BC is the well-known Rankine cycle, working with steam or any other two-phase fluid. Another possible BC is the Air Bottoming Cycle – an air-driven expander and compressor. Externally fired piston engines (Stirling engines) can also work in bottoming cycles. Any fuel can be utilized in these cycles.
In the present research project, steam Rankine cycle and Air Bottoming Cycle will be modelled as BC, fuelled solely by biomass (wood) or MSW.

3.3. Combined Cycles

"TC" and "BC" terms make any sense only if we talk about a combination of power cycles, one of which is the topping, another one the bottoming. The most common combination of power cycles is the GTCC – a GT as TC and a steam Rankine cycle as BC. It can be also addressed as "pure" or "unfired" GTCC, as long as the only fuel input is at a high temperature level in the GT combustion chamber, while the steam cycle utilizes the remaining heat in the GT exhaust. The unfired GTCC is the most energy efficient power cycle today. Many such combined cycles are in operation around the world, with units ranging from several MW up to more than 200 MW.

In general, with present technology, the output of the bottoming ST is about half that of the topping GT in the CC, which means that adding a BC to the GT will increase the total output 1.5 times. The costs for the construction of a Rankine BC however are much higher than the costs for the GT.
A simplified example of an unfired GTCC is presented on **Fig. 3.1**.

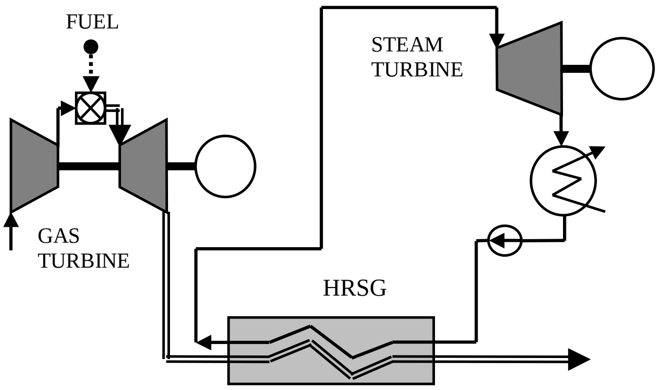


Fig. 3.1: *Simplified chart of an unfired GTCC.*

The heat input to the BC depends on the amount of heat rejected by the TC, the exhaust temperature of the TC and on the heat transfer effectiveness and irreversibilities in the HRSG. If higher heat input temperatures are needed for the BC, supplementary firing in the NRSG can be performed.
Supplementary firing in the HRSG with the same fuel as for the GT is a common practice and can be employed for various reasons. Such reasons can be the need for increased power output for peak-load covering, offsetting the GT loss of power at high ambient temperatures, improving part-load efficiency, or simply the need for higher steam superheat/reheat temperatures, cycle flexibility and possibility to run as a pure steam cycle during GT outage.

Supplementary firing, even if at a very small extent and with the same fuel, converts the unfired GTCC into a fired GTCC. If all the remaining oxygen in the GT exhaust is used for supplementary firing, the cycle is transformed into a fully-fired CC. Studies about the effects of supplementary firing on the overall cycle output and efficiency have been performed by many authors, targeting different issues. Cycle performance and especially part-load behaviour and part-load efficiency have been largely investigated. Examples for such studies are [3.2], [3.10], [3.15], [3.12], [3.17].

If another fuel is used to provide additional heat input for the BC, the GTCC will be transformed into a HCC. Two basic arrangements are possible:

1) Fired or fully-fired HCC, where the TC exhaust is used as combustion air for firing the other type of fuel. This additional firing can be performed in an adapted HRSG, or (in the case of solid fuels) in a standard boiler for the BC.
2) Parallel-powered HCC, where the heat of the TC exhaust is utilized for feedwater preheating and/or additional steam generation for the BC. The BC has a separate steam generator, in which the additional fuel is fired.

Simplified examples of fully-fired and parallel-powered HCC are given on **Fig. 3.2** and **Fig. 3.3** respectively. All figures of HCC later in this report feature different modifications and layouts of these two basic arrangements.

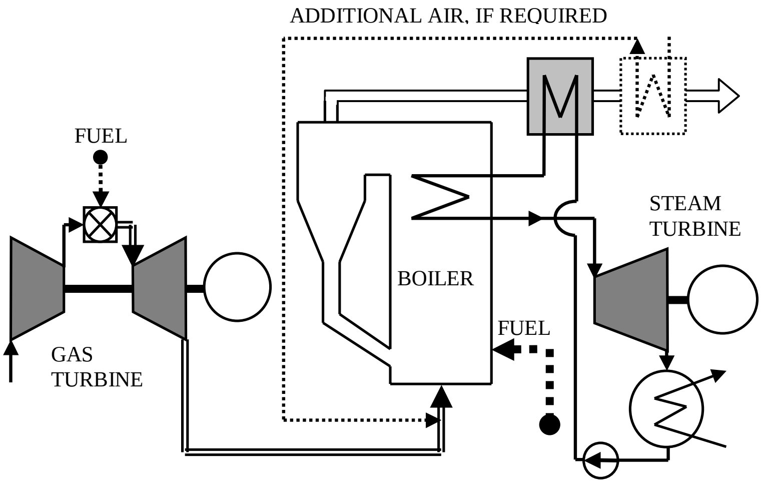


Fig. 3.2: *Simplified chart of a fully-fired HCC.*

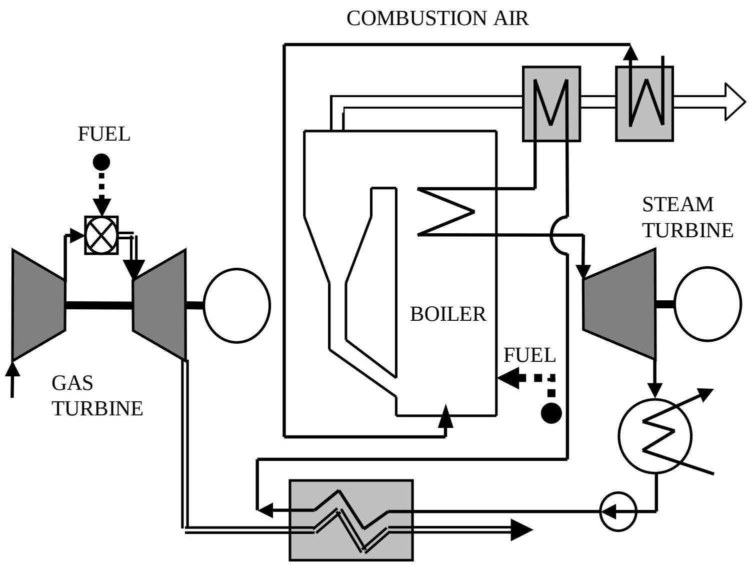


Fig. 3.3: *Simplified chart of a parallel-powered HCC.*

3.4. Review of supplementary fired CC and coal-fired HCC

It was not possible in this literature study to track the entire history of research and development work on supplementary-fired CC and coal-fired HCC. However, some facts depict the whole conception of hybrid combined cycles, from adolescence to maturity, and they must be presented here.

The idea for hybrid cycle development is not new. The first suggestions for such a partial combination between a TC and a BC have originated as long back as the idea for a GTCC itself. The first proposed combined cycles, well before 1960, included also supplementary firing in the HRSG. The gas turbine was just entering its first industrial applications at that time, and its performance was not as advanced and reliable as it is now, so supplementary firing was an important tool to achieve reasonable efficiencies in the steam BC and high enough total output from the combined cycle. Most of the very first commercial GTCC units had a fired HRSG or a conventional boiler with the same fuel as the GT. At that time they showed 5-6% higher efficiencies than the conventional steam plants. [9.7], [3.4]
Finckh [3.12] points out that limited supplementary firing in the HRSG of a GTCC with one steam pressure level for the BC would result in increased cycle efficiency. In the case of more advanced heat utilization in the HRSG (dual steam pressure level with reheat), efficiency would not increase with supplementary firing, but nonetheless, supplementary firing of up to 30% would not result in big efficiency loss [3.12], [3.2].

Of course, since coal is generally the most accessible and cheapest fuel, it didn't take a long time until ideas for coal utilization as a supplementary fuel in combined cycles were proposed. Serious considerations for HCC development have started with the suggestion for installing topping gas turbines on existing coal-fired steam boilers, after technical and economic studies undoubtedly showed that this would be an easy and cheap way to improve both output and efficiency of the old coal power plants. The short construction times needed for simple addition of a gas turbine cycle to the existing plant with minor modifications of the steam circuit and boiler, were also adding to the growing popularity of this method for refurbishment of old plants.
Repowering was undoubtedly the primary reason for the interest in HCC, but new-build power stations in HCC mode were also considered. Of course, the leaders in this development were countries with ageing coal-fired power plants, available modern gas turbine technology and energy efficiency at the top of the agenda of their governmental energy plans. Finally, it was the energy efficiency and economic attractiveness that was the primary impulse for actual investments. Low-investment strategies for the renewal of old steam generators have been developed in all industrialized countries, based on the ground of needed increase in installed power, increase in efficiency and refurbishment of ageing boilers with minimum financial burden. Leaders among these countries in terms of number of installed large-scale units are Germany, Holland and Japan.

Literature on the relevant issue in English language has been published in technical journals or presented at international conferences by authors from various countries. The prevailing number of authors is from several countries in continental Europe, USA and Japan. Many articles have appeared in German language by German, Austrian

and Dutch authors in technical magazines such as VGB Kraftwerkstechnik, Brennstoff-Wärme-Kraft (BWK) and Energie und Technik.
Relevant articles have also been published in Dutch, Italian, Japanese, Finnish, French and of course Swedish language in the respective countries.

US authors have largely contributed to the relatively small amount of publications on issues relevant to HCC promotion and development, but very few units have ever been constructed in the USA.
Authors from the UK have generally not been very active in this field.

IGCC, PFBC and other sophisticated coal utilization technologies, which require intensive research and development and large investments, have been given almost total priority in USA [3.39]. Nevertheless, HCC for repowering of the ageing US coal-fired steam plants has been proposed and put forward (among other technologies), for example in [3.43], [3.35], [3.7], [3.16], [3.26] and others. Brander and Chase [3.4], as well as most of the other authors, underline the potential of repowering coal-fired steam plants with gas turbines, but conclude that switching away from coal to natural gas, replacing the boiler with a HRSG and converting the old plants into unfired GTCC is a more viable option.
However, interest in coal utilization is steadily increasing in recent years and probably the HCC idea will be revived in the US.

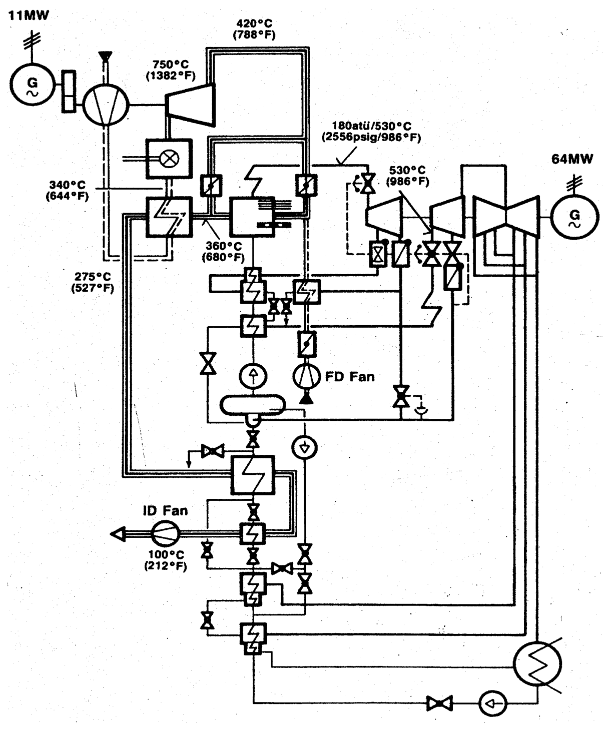


Fig. 3.4: Layout of the "Hohe Wand" HCC Power Plant with a Coal-Fired Boiler. (from [3.40])

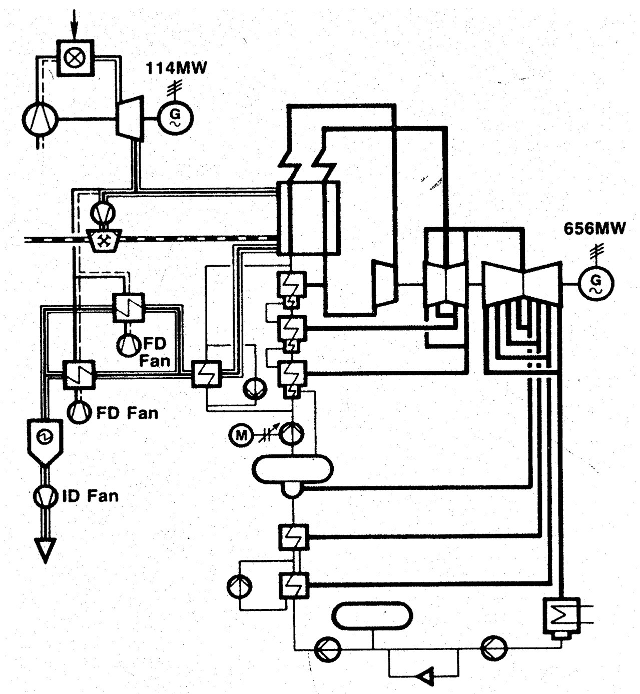


Fig. 3.5: Layout of the "Gersteinwerk Unit K" HCC with a Coal-Fired Boiler. (from [3.40])

In the early 1960's, fully-fired CC plants utilizing the high temperature gas turbine discharge as combustion air intake for a coal-fired steam generator were developed.

The first of such plants was the power station "Hohe Wand" (Austria), which went into commercial operation in 1965 [3.40], [3.5], [4.4]. The total output of this unit was 75 MW, with a gas turbine rating at 11 MW and a single reheat steam turbine rating of 64 MW. Very modest gas and steam turbine inlet conditions (GT inlet temperature of 750^{o}C, according to the then prevailing state of development) still provided a thermal efficiency of 40.5% [3.40], [3.18]. The layout of this power plant is shown on **Fig. 3.4**.

In the following years, the unfired GTCC power plants underwent a steep increase in performance and popularity. Extensive experience had already been gained with a large number of them. In fact, CC development has always proceeded in parallel with GT development. Meanwhile, many fully-fired units utilizing natural gas or oil for both the gas turbine and the steam generator were build and operated in Germany. They showed excellent performance and reliability statistics, which gave confidence in these types of combined cycles (see Table 3.1 on next page).

In 1984, a 750 MW fully-fired HCC unit with a coal-fired steam generator went into operation as the fifth unit of the 2300 MW Gersteinwerk power plant (Germany). Its flowchart is shown on **Fig. 3.5**. This plant was not optimally designed in regard to the gas/steam turbine output ratio and supplementary air was needed for the steam boiler, supplied by FD fans. Despite this feature and the moderate inlet conditions for the gas and steam turbines, the net efficiency of the entire power station (including losses for flue gas desulphurisation) reached 41%. [3.40], [4.4]

Optimal matching of the gas turbine output to that of the steam boiler for fully-fired plants of a similar scale to the one mentioned just above, with two gas turbines and one steam boiler as an example, is presented on **Fig. 3.6**.

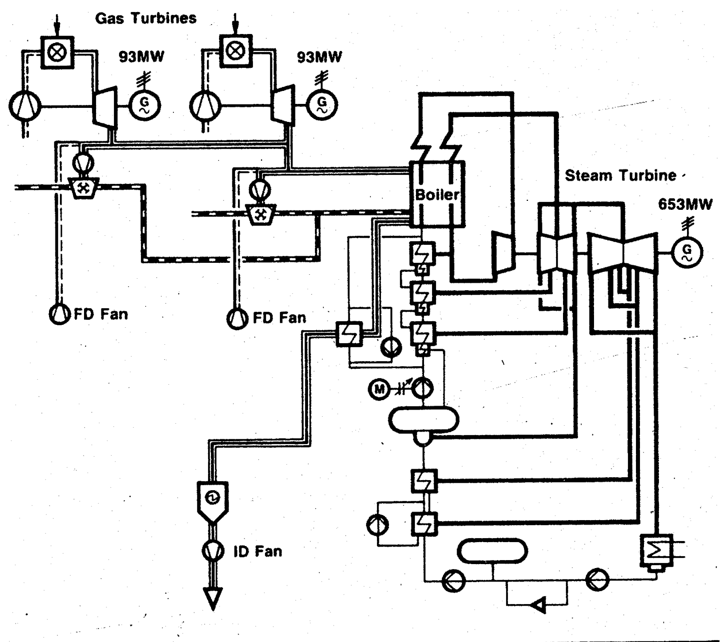


Fig. 3.6: *HCC Power Plant with two Gas Turbines and one Coal-Fired Boiler. (from [3.40])*

This optimum coordination of gas and steam turbines is reached when all but a minimum of the oxygen present in the GT exhaust is consumed during combustion in the steam generator. Nevertheless, plants requiring supplementary air because the GT is too small are also economic [3.6]. A very high flexibility is achieved if the FD fans can allow full-load operation without the gas turbine.

Kraftwerk Union (KWU) has been involved in building a large number of fully-fired CC units with a variety of fuels (many of these plants used natural gas both as gas turbine and boiler fuel). A list of such plants, which feature KWU gas turbines, is presented below in **Table 3.1**.

Table 3.1: *Fully-fired HCC power plants with KWU gas turbines. (Adapted from [3.40])*

	Output (MW)		Fuel		In service from
	Total	Gas Turbine	Gas Turbine	Steam Generator	
1) NEWAG Hohe Wand	75	11	NG	Coal / NG / DO	1965
2) Neckarwerke Altbach	251	51	NG / DO	NG / HO	1971
3) NWK Emden	452	52	NG	NG	1973
4) VEW Gersteinwerk F	417	52	NG	NG	1973
5) VEW Gersteinwerk G	417	52	NG	NG	1973
6) VEW Gersteinwerk H	417	52	NG	NG	1973
7) VEW Gersteinwerk J	417	52	NG	NG	1973
8) VEW Emsland B	417	52	NG / DO	NG	1974
9) StwD Duisburg A	172	32	NG / DO	NG	1974
10) VEW Emsland C	417	52	NG / DO	NG	1975
11) GFA Gebersdorf	403	51	NG / DO	NG / HO	1975
12) EW Wien Simmering	445	66	NG	NG	1977
13) Saarbergwerke Voelklingen	223	33	CG / DO	Coal	1983
14) VEW Gersteinwerk K	770	114	NG	Coal	1984

NG - Natural Gas; DO – Distilate Oil; HO – Heavy Oil; CG – Coke Gas.

Partial gasification of coal for running the gas turbine on product gas while the steam boiler runs on coal, thus creating a fully-fired CC with supplementary firing using only coal for both TC and BC, has also been considered and investigated. Such a scheme would avail of the decreased complexity of gasification and increased flexibility of the concept, if only part of the fuel for the cycle is gasified. A power plant of this type has been built in Germany in 1972 as a prototype for gaining experience with advanced systems. Its layout has been described by Termuehlen [3.40].

Fig. 3.7 presents an exemplifying comparison between the net electrical efficiencies of three types of power cycles – open simple gas turbine cycle, conventional steam turbine cycle with moderate steam parameters and a fully-fired CC of the same output as the basic steam cycle. The figure shows net efficiency curves over the full load range for each cycle. It clearly reveals the fully-fired HCC performance advantage over separate gas or steam cycles at any load. At part-power loads between 65% to 100%, the HCC even shows increased efficiency. [3.40], [3.5], [3.6], [3.23]

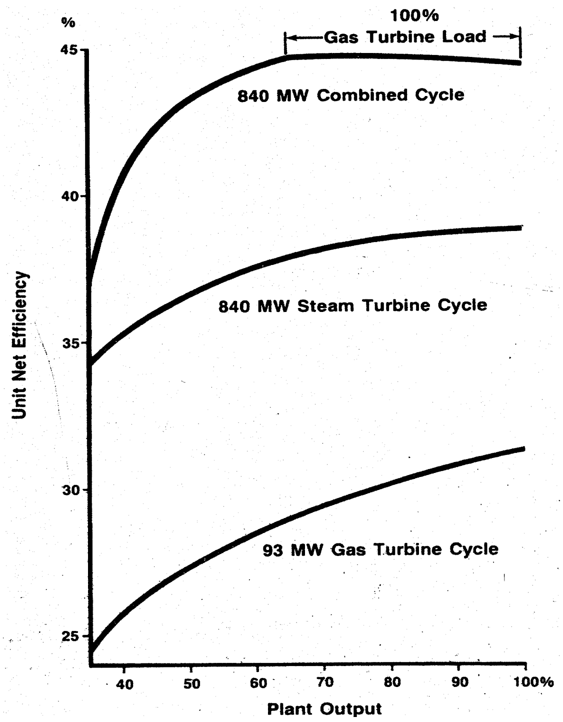


Fig. 3.7: Simple cycles and fully-fired HCC. Comparison of part-load efficiencies. (from [3.40])

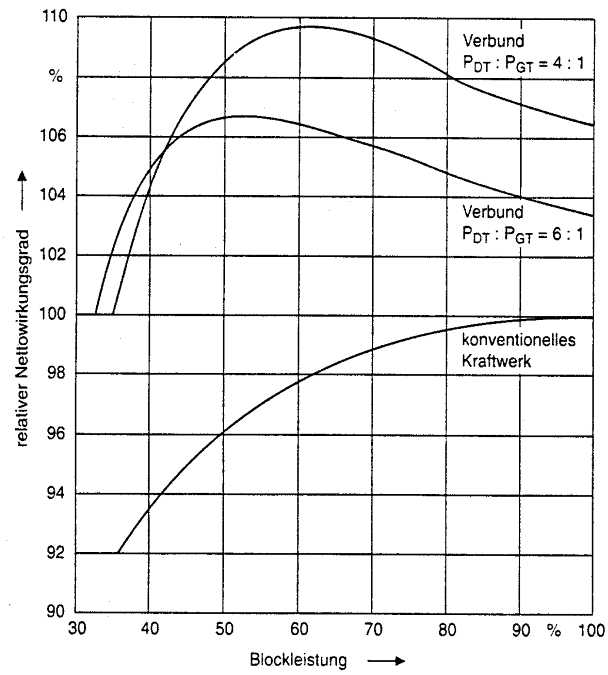


Fig. 3.8: Conventional steam cycle and two HCC with different ST to GT output ratios – comparison of efficiencies at part-load. (from [3.1]).

Another efficiency comparison at part-loads is presented on **Fig. 3.8**. The figure has appeared in the first reports on the relevant topic, published in German. A standard steam Rankine cycle is compared to a hybrid combined cycle, where two different ST to GT power output ratios are considered, showing the effect of the gas-to-coal fuel input ratio. Efficiency variations are shown relative to the value of the simple steam cycle at full load. DT is an abbreviation from "Dampfturbine", meaning "steam turbine" in German. [3.1], [4.4]

This good part-load efficiency, typical for any HCC, can be achieved by decreasing the load of the steam generator (the BC) while keeping the gas turbine (the TC) running at full power. In this way a larger part of the total heat for the cycle is provided at the high temperature level in the gas turbine, improving the prerequisites for efficient energy conversion. If the GT load is decreased, the poor part-load performance of the GT will inevitably affect the whole cycle performance and the cycle efficiency will start to decrease rapidly with decreasing power. Thus, to make the best out of the good part-load features of HCC, gas turbine load must be decreased only after the load of the steam boiler has been decreased to its minimum level.

However, when reducing steam generator output while keeping full gas turbine load (for a cycle layout such as that on **Fig. 3.6**), the partial feedwater flow through the HP feedwater preheater will be throttled to keep the partial flow economiser flow unchanged. At about 65% load all feedwater passes through the partial flow economiser and the feedwater flow through the HP feedwater heater becomes zero. A further reduction in steam generator output with full gas turbine load will lead to an increase in feedwater temperature and consequently to boiling conditions at the evaporator inlet. To stay within the temperature limit of 10°C below boiling temperature as illustrated on **Fig. 3.9** for this specific case, a lowering of the gas turbine output is necessary (since lowering of the final feedwater temperature by, for example, bypassing the partial flow economiser would result in a thermal performance loss of the power cycle). [3.40]

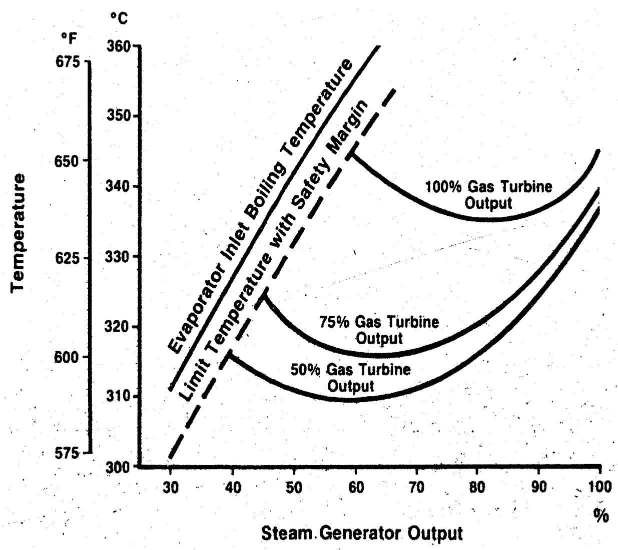


Fig. 3.9: Evaporator inlet temperatures for different steam generator and gas turbine part-load outputs for the HCC plant shown on Fig.3.6. (from [3.40])

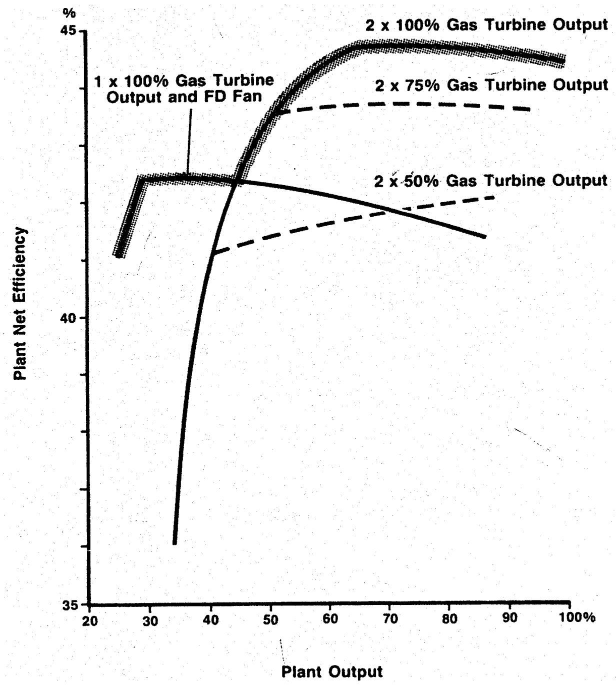


Fig. 3.10: Comparison of efficiencies at part-load operation for a HCC power cycle with one or two topping gas turbines. (from [3.40])

In a two gas turbine plant arrangement, the partial load performance can be improved by operating, at reduced plant output, only one of the two gas turbines. Plant net efficiency in this mode of operation is shown on **Fig. 3.10**. If both gas turbines are operated and the output of both of them is reduced below 50%, the loss in thermal efficiency is quite drastic. Decreasing the output of only one gas turbine while keeping the other at full load until the first one reaches zero load, provides excellent part-load performance down to around 25% of the total plant output. [3.40]

Raising cycle efficiency and increasing plant output can be easily achieved by converting an old steam turbine plant into a HCC plant by addition of a gas turbine, thus creating a fully-fired or parallel-powered HCC. Topping of such old steam plants with gas turbines is a highly economical way to achieve the improvements mentioned above, if fuel for the gas turbine is available at reasonable prices.
Repowering of old steam power plants can also be performed by transforming them into pure CC unit by addition of a gas turbine, dismantling of the steam boiler and replacing it with a HRSG. This concept may be attractive in small scales, but is quite expensive to implement in large scales (if not to say impossible). Topping a power plant with transformation of the steam boiler into a fully-fired unit can be performed in any scale and is particularly attractive in large scales, up to 700 MW and more [Term]. Of course, preliminary studies must be carefully executed in order to find out whether such modifications to the boiler are technically and economically feasible. Transformation of a steam power plant into a parallel-powered CC is particularly attractive at any scale. The changes to the steam flow path in both these cases are minimal.

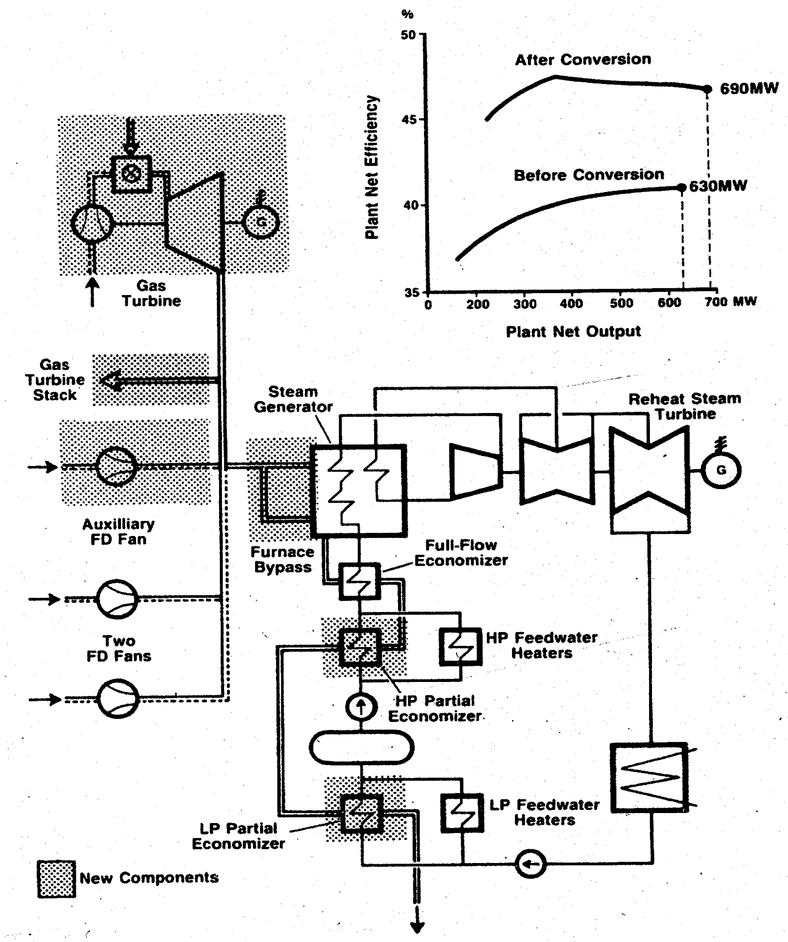


Fig. 3.11: *An example for repowering of an old steam plant into a fully-fired combined cycle – "Eemscentrale 2", The Netherlands. (from [3.40] and [3.5])*

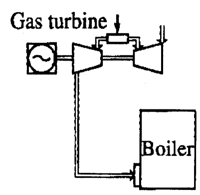
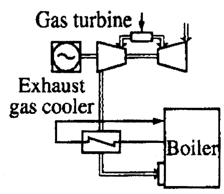
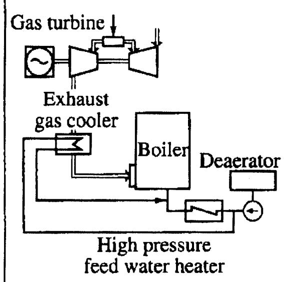
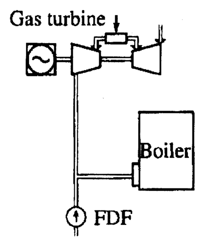
Just as in newly build fully-fired HCC plants, repowered plants can also be designed for independent steam or gas turbine operation. In fully-fired units, separate fans for combustion air must exist, if the boiler is to be operated without the gas turbine. Such concepts allow for the most flexible and efficient operation of the TC and BC together or separately, as required by load-following duties or maintenance procedures.

An example of a plant repowered into a fully-fired unit is shown on **Fig. 3.11**. A 140 MW gas turbine is topped onto the old 630 MW steam turbine plant. The GT exhaust is fed to the steam generator, but part of it is bypassed directly upstream of the first superheater section. The output of the steam generator is lowered after the conversion, so that the renewed plant output is 690 MW in total. Partial flow economisers are chosen for both HP and LP feedwater heating. The net efficiency of the plant is increased to 47% (from 40% before) and the total fuel consumption for the 690 MW power is less than that for the 630 MW previously. [3.40]

In Japan, fully-fired cycles have been seriously considered for repowering of old steam generators. A sharp increase in power demand during the last 10-15 years in Japan (mostly for peak-load electricity) has triggered a growing interest in HCC development. A comparative study among different fully-fired HCC system configurations applicable to Japanese conditions have been presented by Morikawa et.al. and Takizawa et. al. [3.27] and [3.38].

Regarding GT exhaust heat utilization, different cycle configurations are possible. **Table 3.2** shows a comparison between the different fully-fired cycle arrangements.

***Table 3.2:** Comparison of fully-fired CC configurations and features. (from [3.27] and [3.38])*

Equipment of gas turbine exhaust heat recovery	Hot wind box type	Cold wind box type		
		Gas feed water heater		FDF air use
		Boiler evaporator	Feed water system	Boiler
Configuration				
Modification scale of boilers	Large	Fairly large	Fairly small	Small
Features	• Plant efficiency is optimal because of recovering the gas turbine exhaust heat in the boiler. • Special structure is required at the connection of boiler and wind box.	• Plant efficiency in this case is nearly the same as that of the hot wind box-type. • A tube layout is needed from the boiler evaporator to the cooler.	• Plant efficiency in this case is slightly lower than the hot wind box type and gas turbine exhaust heat recovery type by the boiler evaporator. • A tube layout from the cooler to feed water system is needed.	• Plant efficiency in this case is nearly the same as the gas turbine exhaust heat recovery type by the feed water system. • Adequate FDF air flow rate to control wind box inlet gas temperature is required. Dry gas loss is large. • Special structure is needed at confluence of gas turbine exhaust gas and FDF air.

The system where GT exhaust is directly introduced in the boiler as an oxygen source for combustion is called "hot windbox". The steam boiler accepts the gas turbine

exhaust gas at temperatures around 500°C to 550°C, depending on the gas turbine type. This system configuration necessitates a larger modification of the combustion air ducts and burners of the steam boiler to withstand the very high admission temperatures [3.27]. An example of such a cycle layout is presented on **Fig. 3.12**.
Hot windbox repowering has the highest degree of technical complexity of all GT-based repowering options [3.35].

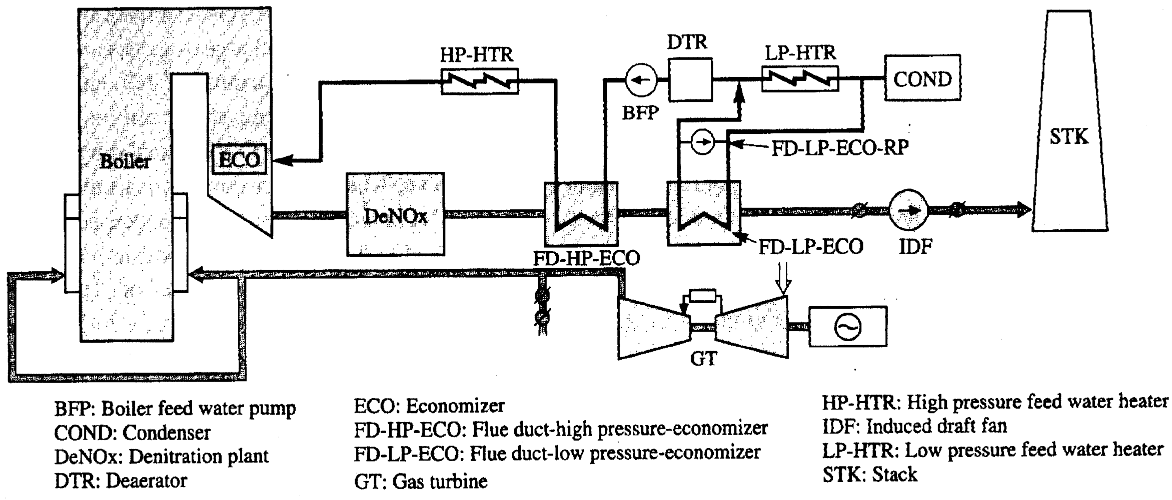


Fig. 3.12: *An example of a hot windbox fully-fired CC. The hot gas turbine exhaust is introduced directly into the boiler. (from [3.27])*

The system where gas turbine exhaust is firstly cooled to some extent before entering the steam boiler is called "cold windbox" (or "warm windbox" [3.43]). There are several options for partially cooling of the gas turbine exhaust gas, in Table 3 are mentioned feedwater preheater, boiler evaporator surface and a simple mixing with cold combustion air. The advantage of the cold windbox system is the possibility to use the existing combustion air ducting of the steam boiler without any large modifications [3.27].
Moreover, a fan can be applied on the GT exhaust flow path to boost the pressure before the boiler and to alleviate the performance degradation of the gas turbine due to exhaust pressure losses in the heat exchangers.
An example of a cold windbox type layout is presented on **Fig. 3.13**.

The hot windbox type can be expected to have a greater cycle efficiency increment than the cold windbox type, because the former results in a higher regenerative efficiency than the latter. However, this depends very much on the exact arrangement of the GT exhaust heat recovery system and whether this heat is utilized for feedwater heating (low temperature utilization) or for raising additional steam (high temperature utilization). In cold windbox type systems, where GT exhaust is cooled by additional boiler evaporator surfaces, plant efficiency rivalling that of the hot windbox type can be attained [3.27]. Furthermore, backpressure behind the gas turbine caused by pressure losses in the GT exhaust flow usually has a big effect on its efficiency and performance. In the cold windbox type these pressure losses are higher, because heat exchangers are present, but if a FD fan is installed the losses will be overcome and GT performance will be better than that of the hot windbox type system.

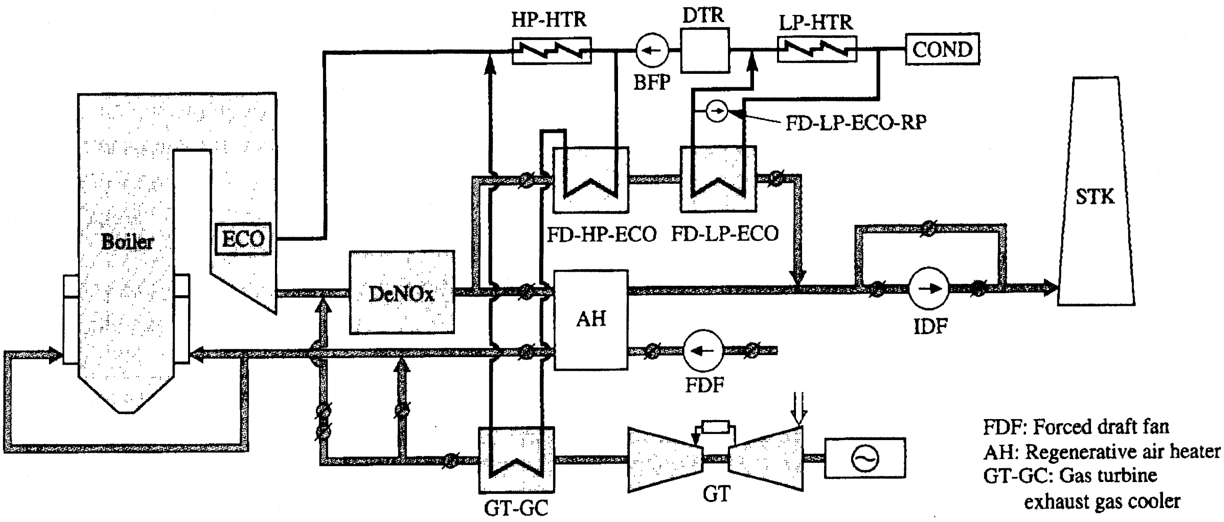


Fig. 3.13: *An example of a cold windbox (also called "warm windbox") fully-fired CC. The GT exhaust is firstly cooled to some extent before being introduced into the boiler. Main cycle components are the same as in Fig. 3.12 above. (from [3.27])*

In all repowered windbox (fully-fired) cycle arrangements, care must be taken also for maintaining steam generator efficiency at its design value. Switching from air to GT exhaust as oxygen-carrier for combustion leads to changes in flows in the boiler (both mass and volumetric flows), as well as to possible changes in adiabatic combustion temperatures, flame luminosity and percentage of radiative heat transfer to convection heat transfer. As a result, radiative superheaters may need to be replaced by convection superheaters, or other changes may be necessary.

One Japanese electric utility has decided in the beginning of the 1990-ies to convert all its existing 700 MW and 375 MW thermal power units into fully-fired HCC power plants. The addition of a gas turbine to the old units was considered the best choice due to a shorter construction period, less environmental degradation and improved plant efficiency. Basic planning for repowering has been started in the fall of 1990 and detailed design and manufacturing of part of the main equipment has been carried out by the end of 1992. This conversion has been required to increase the power supply capacity gradually from the summer of 1994 onward. [3.38]

Parallel-powered cycles have also been considered for repowering of old steam plants. The parallel-powered cycle (also called "compound cycle") is often a better (low cost) alternative for utilities in need for peaking capacity or any combination of base-load, mid-range and peaking capacities. Apart from the increase in power by addition of a gas turbine, parallel-powered cycles feature certain increase in power by closing the ST feedwater steam extractions. In a standard large-scale steam cycle, approximately 20% to 30% of the steam flow may be used for feedwater heating [3.7]. By utilizing GT exhaust for feedwater heating, additional power is produced in the ST. Parallel-powered arrangement features full flexibility. Steam generator for base-load duty can be easily augmented by a topping GT for peak-load duty. The additional power for peak-load coverage can be easily and quickly raised by the GT, while the

ST itself is in operation and can readily accept its own load increase when feedwater preheating starts to be provided by GT exhaust.

A typical example of a parallel-powered cycle is shown on **Fig. 3.14**, where a 370 MW base load is provided by a coal-fired power plant with DeNOx and DeSOx systems. Additional 85 MW of mid-range or peaking capacity is supplied by a 60 MW gas turbine with heat exchangers for feedwater preheating. The feedwater preheating provided by the gas turbine exhaust results in an output increase of the steam turbine from 370 to 395 MW due to the closing of the feedwater heater steam extractions #1 through #3 as well as #5 and #6. The heat rate improvement for the combined cycle, as compared to the heat rate of both gas and steam cycles working independently, is 5.5% or 478 Btu/kWh, see **Fig. 3.15**. Another important feature is that with such a parallel-powered arrangement, mid-range or peaking capacity is provided with a very low natural gas consumption at a roughly 50% efficiency level. In addition the specific heat rate per kWh is reduced. [3.5]

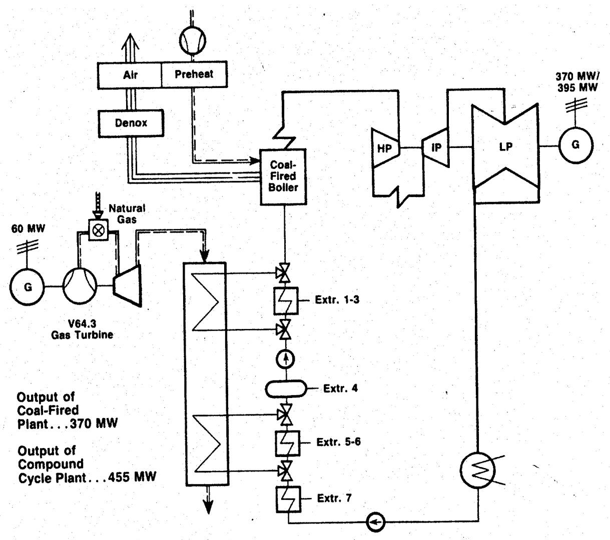


Fig. 3.14: Parallel-powered HCC with Coal-fired steam generator and heat recovery for feedwater preheating. (from [3.5])

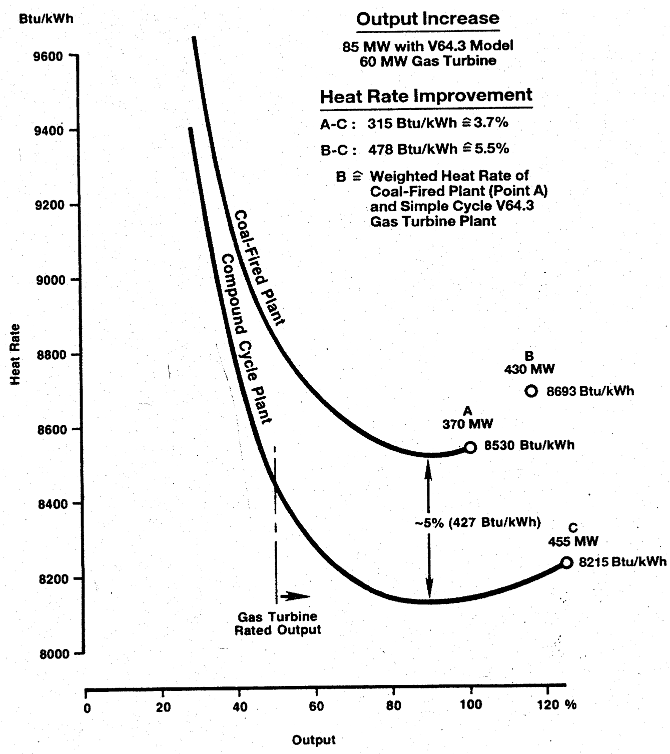


Fig. 3.15: Performance improvement of a parallel-powered (compound) HCC after topping with a gas turbine. (from [3.5])

When considering such a cycle configuration however, care must be taken to ensure that the ST can carry the increased load after closing some or all of the steam extractions for feedwater preheating. Usually the turbines are designed for operation with extractions closed, but often modifications of the LP turbine blading may be necessary, especially for the very last turbine stage. If the ST and condenser cannot accept the increased steam flow, reduction of throttle steam flow will be necessary, which will offset the power increase form closed steam extractions.

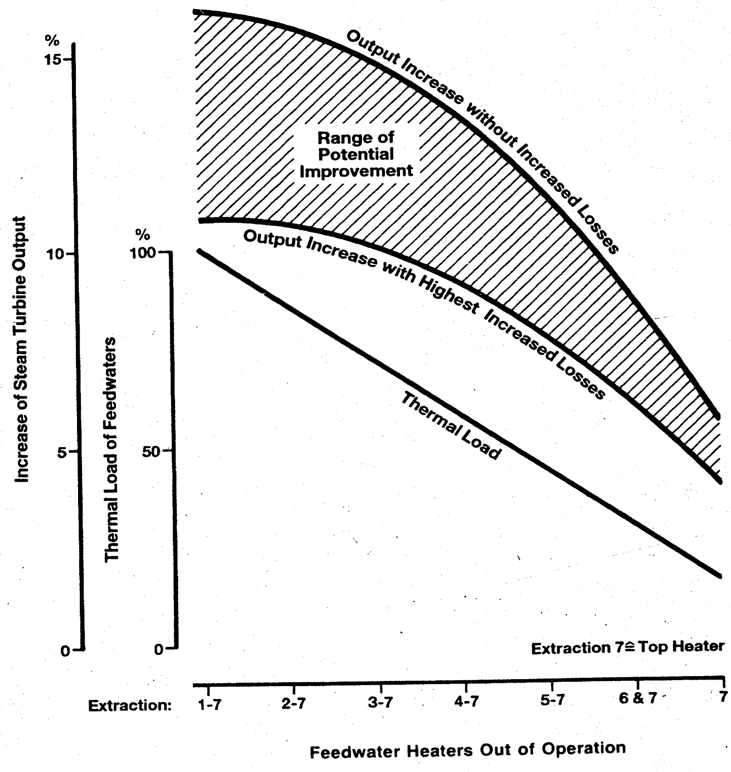


Fig. 3.16: *Effect of gas turbine topping and steam extraction closing on the performance of parallel-powered HCC. (from [3.5])*

For the selection of a parallel-powered HCC with heat exchangers for feedwater preheating, it is of utmost importance to optimise the cycle by supplying the gas turbine exhaust energy to the steam cycle at a high energy level. **Figure 3.16** reveals this important feature for a typical application of a gas turbine being topped to an approximately four times larger steam turbine plant. With equal temperature rise in each heater, the feedwater heaters' thermal load decreases by roughly 15% when switching off one feedwater heater starting with the top one. The increment in steam turbine output is largest when disconnecting the upper feedwater heaters (the HP ones). Shutting off the LP feedwater heaters has little effect on increasing the steam turbine output. For example only about 1% output gain is achieved by switching off the three LP feedwater heaters.
It is also of importance to recognise that a wide range of output gain is directly influenced by the magnitude of additional losses. The exhaust losses of an already small LP turbine section or a limited condenser capacity can especially reduce the theoretical output increase. Therefore, it is of utmost importance to select for each application the optimal cycle arrangement. [3.5]

A parallel-powered cycle can be arranged in a different way than the one described above. One possible alternative is to utilize the gas turbine exhaust in a special HRSG for supplying additional HP, MP or LP steam for the steam turbine. Another option is to perform steam reheating in the HRSG. [3.5], [3.30]

A combination of any of these two options with feedwater preheating can possibly give the best performance and efficiency for a parallel-powered HCC. One such study has been performed by Brückner et al. [3.5] in 1992. They have compared 4 different parallel-powered HCC arrangements in which one gas turbine is topped to two similar steam cycles, and the gas turbine exhaust has been used in the following ways (as of **Fig. 3.14**):

1) Feedwater heating in preheaters No.3 through No.7,
2) Feedwater heating in preheaters No.1 through No.7,
3) Hot reheat steam supply,
4) Hot reheat steam supply with feedwater heating in preheaters No.3 through No.5.

The results from the analysis are presented in **Table 3.3**.

Table 3.3: *Performance comparison of alternatives for parallel-powered repowering. (adapted from [3.5])*

		Present Reheat Steam Plants	Alternatives			
			1	**2**	**3**	**4**
Gas Turbine Output	MW	-	1x 145.9	1x 145.9	1x 145.9	1x 145.9
Steam Plant Output	MW	2x 326.8	2x 355.6	2x 345.9	2x 353.5	2x 354.8
Total Output	MW	653.6	857.1	837.7	852.9	855.5
Auxiliary Power	MW	2x 16	2x 16	2x 16	2x 17	2x 17
Plant Net Output	MW	621.6	825.1	805.7	818.9	821.5
Plant Net Heat Rate Based on HHV (el. efficiency, %)	Btu/kWh	9885 (34.53%)	9269 (36.82%)	9496 (35.94%)	9262 (36.85%)	9230 (36.98%)
Increase in Plant Net Output	MW %	Base Base	203.5 32.7	184.1 29.6	199.3 32.1	201.1 32.4
Improvement in Plant Net Heat Rate	Btu/kWh %	Base Base	616 6.2	389 3.9	623 6.3	655 6.6

The conclusions from the analysis (Table 3.3) have been the following:
All 4 alternatives are valid options for repowering in the form of a parallel-powered HCC. The two feedwater heat exchanger options seem to be easier to connect into existing plants, since only feedwater piping connects the gas turbine plant to the two steam plants. Pressure and heat losses are not a major concern. On the other hand, the heat recovery steam generator options provide a better performance improvement.
It may seem that the option with feedwater heat exchangers should be much less expensive, but the necessary addition of auxiliary stacks and dampers raises the cost.
The feedwater heat exchangers options provide greater operating flexibility, which is especially important if load cycling and two-shift operation is considered.
In regard to thermal performance, alternative 4 is the best solution.
The gas turbine performance is influenced by differences in outlet pressure loss of the heat exchangers and HRSGs, however, all calculations have been performed with a similar pressure loss value because it has been assumed that a lower pressure drop in the heat exchangers might be compensated by the use of the auxiliary stack and dampers for these alternatives.
The exhaust steam flow is the only one which has been increased above its original value. But this increase of only 7% can generally be handled if the steam turbines are

not back-end loaded or the LP blading is designed for a low mass flow limit. In such a case the LP steam turbine path can be replaced with a more advanced design. This can lead to an additional performance improvement of about 16 MW (2 to 3%) over alternative 1 for both steam turbines.
In regard to emission discharge, the specific NOx emissions per generated kWh are reduced for all alternatives (with low-NOx burners for the gas turbine). A larger improvement can be achieved with fully-fired HCC plant, especially if low-NOx burners are also fitted to the steam generator.
In cases where large relatively new existing steam plants reveal a potential for uprating, the parallel-powered cycle application is an attractive option. The specific costs for such parallel-powered topping can be as low as the expenses for building a new simple cycle gas turbine plant. Without modifying the steam turbine LP flow path, the capacity increase of existing steam turbines might be limited. [3.5]

Before the end of the 1990-ies, there have been 16 fully-fired CC or coal-fired HCC power units in operation in Germany, another 11 units in the Netherlands and several such in Austria. Some of them have been newly designed, while most have been repowered old steam boilers. Fuels range from natural gas for both the TC and BC through distillate oils (for peak-load covering) to various types of coal for the BC. Several large-scale plants are in operation also in Japan. [3.27], [3.31]

Ten steam power units in Italy have been repowered with gas turbines (one per unit) in feedwater preheating configuration. This has been the objective of the Italian Public Utility long-term plan for a significant increase in net conversion efficiency of the entire national system. [3.14], [3.25]

All Dutch repowered units have been natural gas fired steam generators combined with natural gas fired gas turbines into hot windbox fully-fired cycles of various sizes. This was a result of an extensive program for increasing cycle efficiencies and improving natural gas energy utilization in the Netherlands with minimum expenditures, carried out during the 1980-ies and 1990-ies. Eleven natural gas fired steam units have been topped by gas turbines in an effort to provide better electric efficiency while keeping the power output close to the level of the old steam cycles. Additional power has not been the incentive in the Netherlands, rather the increase in efficiency with the least possible financial burden only. The steam turbines have been operated with decreased output ever since the repowering. Provided that feedwater preheating is done to a high extent by the boiler exhaust gases, the decreased load of the steam turbines ensures stable operation without the need to refurbish the LP stages and the condensers for handling the increase in steam flow due to closed steam extractions. [3.31], [3.24], [3.23]

The efficiency improvement of the Dutch repowered stations has been 5-6 %points over a broad power range from 30% to 100% load. The average value of this improvement and corresponding fuel savings varies from 14%points at the original full load conditions to 18 %points at 50% part-load. Even at minimum load the efficiency improvement is remarkable. This is illustrated on **Fig. 3.17**. Dutch repowering project is based on GT technology from the middle of 1980-ies, with average GT simple cycle efficiency of 31-34%. If repowering is considered with the advanced recent GT technologies with efficiencies of 34-36%, additional improvement of 2 %points for these power stations is possible [3.31].

NOx emissions have been decreased in average by 40%, thanks to the lower amount of fuel burned (10-15% decrease in fuel consumption) and the changed combustion environment in the fully-fired boilers [3.23].

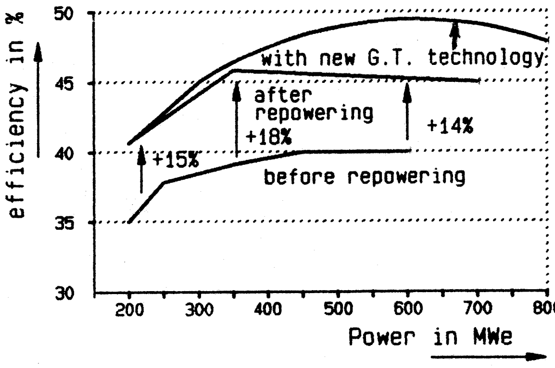


Fig. 3.17: *Efficiency improvement of one of the representative Dutch power stations before and after repowering. (from [3.31])*

In Austria, an old fully-fired CC unit (in operation since 1976, with oil and gas as steam boiler fuels) has been retrofitted by the addition of a new larger gas turbine, converting it into a mixed fully-fired & parallel-powered cycle [3.32]. The old 72 MW GT exhausts into the steam generator (total unit power 250 MW) while the exhaust gases from the new 150 MW GT are utilized for producing additional steam in a HRSG. The overall cycle efficiency is raised to a maximum of 49%, achieved at 35% part-load of the steam generator. [3.32]

In general, hybrid combined cycle power units with coal, oil or natural gas fired BC in various scales exist also in almost all countries in Western, Northern and Central Europe, as well as in Thailand, China and the USA.
Large operational and maintenance experience has already been obtained from these plants.

Almost all large industrial gas turbine manufacturers have presented reports on their products' feasibility and compatibility for repowering old steam units in any cycle configuration. Examples are [3.36], [3.41], [3.5], [3.4], [3.42].
Several case studies (not reviewed here) on design, construction and performance of specific units have also been published.

Repowering with advanced gas turbine technologies has also been proposed. The comparison between topping a steam plant with conventional GT or partial-oxidation GT in [3.21] is just one example.
Steam-injected gas turbines should also be suitable for topping applications.

Among all studies on externally fired gas turbine cycles, a coal and natural gas fired cycle with indirectly (externally) heated gas turbine working fluid has also been proposed [3.33]. A coal-fired furnace supplies heat to the compressed air for the gas turbine, after which natural gas firing lifts the temperature up to the required inlet conditions for the GT. Remaining heat in GT exhaust and coal furnace exhaust is utilized by a steam cycle in a HRSG. Coal supplies 55% of the heat required by the GT. The cycle uses a fuel mix of 65% coal and 35% natural gas. Overall efficiency of the system exceeds 47%. [3.33]

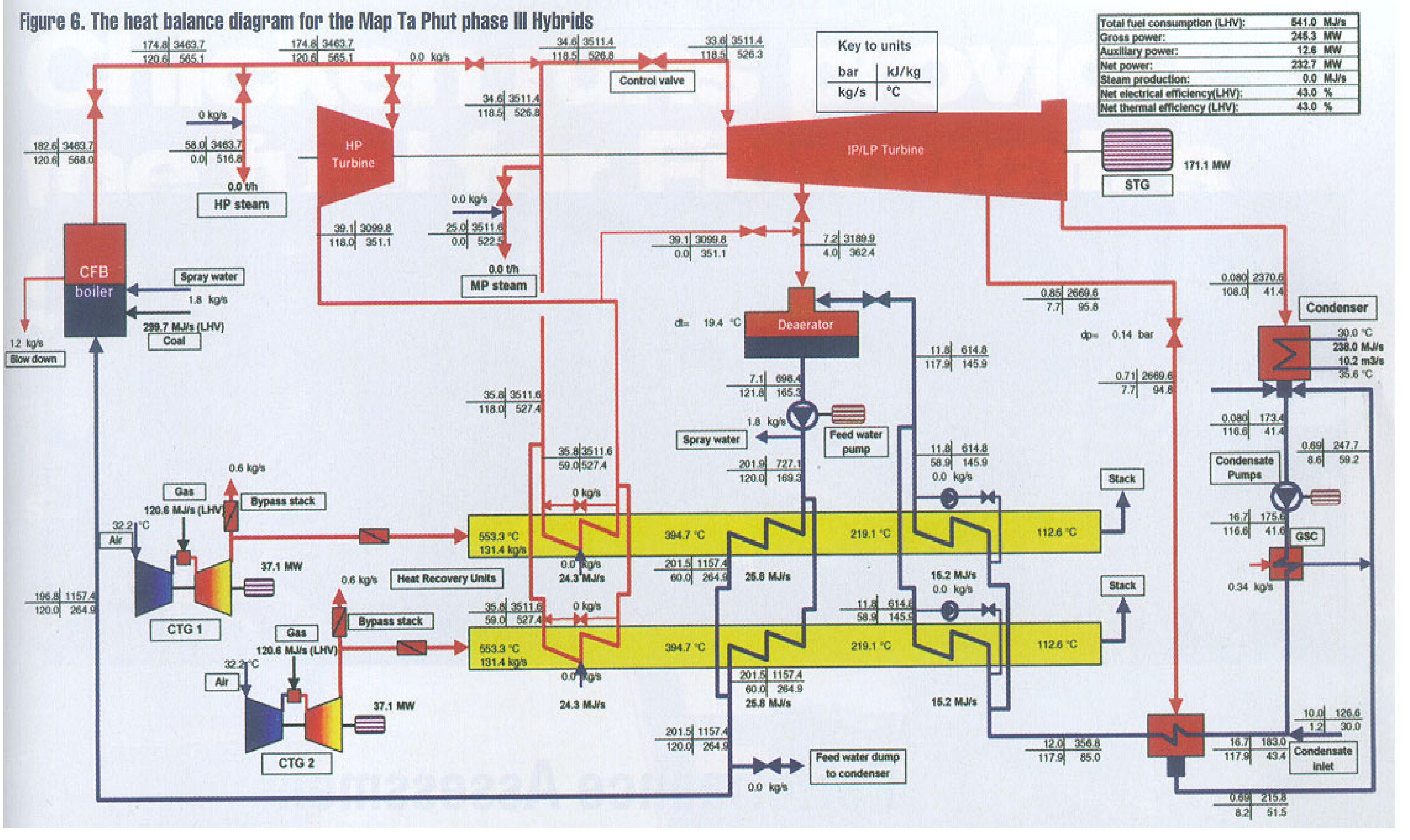
A HCC power plant composed of a natural gas fired GT as TC and a PFBC as BC has been studied by an international group of researchers from universities in UK, Italy and Germany [3.11]. They have concluded that the addition of a topping gas turbine over a coal-fired PFBC boiler can substantially improve the overall efficiency and decrease emissions. The input of some high-quality fuel (natural gas) through high-temperature TC has the same positive effect over the PFBC cycle efficiency, as over the standard steam cycles' efficiency. [3.11]

Applications of gas turbines for supplying preheated combustion air for industrial furnaces have also been suggested. Schemes for such types of cogeneration arrangements suiting various industrial settings are quite attractive. The required modifications to the existing equipment must be carefully evaluated. Certain technical problems with installation and type of the topping GT must be addressed, as well as the possible problems with controlling the turbine, the furnace and the industrial process. Investment costs can be expected to be low enough for the application to be viable. [3.8]

As a conclusion, it must be noted that any possible combination between a high temperature TC and lower temperature BC is viable and feasible. Numerous studies have been performed and papers presented on various ideas for hybrid combined cycles with nuclear, geothermal and solar powered bottoming cycles for example.
For inherently low-temperature heat sources, a combination with a topping cycle is a simple and cheap way to increase the efficiency of their energy utilization. The thermodynamic background of such improved performance is the same as for coal or biomass fired bottoming cycles with topping gas turbines.
However, the possible cycle configurations of a topping cycle combined with a nuclear, solar or geothermal powered bottoming cycle are restricted to the parallel-powered arrangement with superheat by the TC exhaust, because the BC is not a combustion cycle. Repowering nuclear, solar or geothermal power plants with topping gas turbines or even with topping steam cycles have been proposed, including feedwater preheating for steam cycles by solar concentrators, nuclear reactors or geothermal heat.

One perfect example of a newly constructed large-scale parallel-powered HCC are the "Map Ta Phut" two 230 MW_{el} units in Thailand. They have been in commercial operation since March 2000 and use natural gas/oil and coal as fuels. The plant has been specially designed to take advantage of all positive features that a parallel-powered HCC can offer. It supplies thermal energy in the form of process steam to the local industries and uses both fuels in varying quantities, according to actual market prices. Two gas turbines with two parallel HRSGs provide feedwater preheating and steam reheat for the single steam turbine in each unit. Main steam with subcritical parameters is generated in a CFB boiler in each unit. Process steam is extracted at two pressures, before the HP ST section and after the reheat. The ratio between ST output and combined GT output for each unit is 2.3. Net electrical efficiency for the units is 43%. The plant shows extreme flexibility and low capital expenditure. Large amounts of HP process steam can be supplied without the risk of overheating the reheater tubes, due to the fact that reheating is provided entirely by GT exhaust. The hybrid concept shows a running cost advantage of about 9% relative to separate gas and steam units. [3.19]
Fig. 3.18 shows the cycle layout of the power units.

Fig. 3.18: *The heat balance diagram for the new Hybrid Combined Cycle units in Thailand. Each unit comprises one coal-fired steam generator with subcritical steam parameters, one ST, two natural gas fired gas turbines with two heat recovery units, providing full feedwater preheating and reheat for the ST. (from [3.19])*



4. HYBRID COMBINED CYCLES WITH BIOMASS OR MSW AS BOTTOMING FUEL

As pointed out in the previous chapter, traditionally high percentage of fossil-fuel-based power generation, traditionally high demand for higher efficiencies (because of insufficient natural resources as compared to the energy needs) and desire for fuel diversification, as well as a good base for work on research, construction and testing were the primary incentives for any country's involvement in coal-fired HCC development and actual construction of HCC units.
These can readily apply also to the development of biomass or MSW fired HCC.
Another advantage of biofuels that must be again emphasised is the local availability of such renewable energy sources and the lower environmental impact from their utilization for energy purposes, compared to fossil fuels.

4.1. HCC with Biomass as Bottoming Fuel

Almost no publications at international forums are available as a reference, regarding combined cycles with biomass-fired bottoming cycle. No paper has been devoted entirely to hybrid biomass concepts. They have sporadically been mentioned only together with gasification options. Research work and publications on biomass energy utilization (apart from the liquefaction technologies) have mostly featured various kinds of power cycles incorporating gasification, externally (indirect) fired gas turbines or direct fired gas turbines.
In fact, biomass externally fired cycles have attracted as much attention as coal externally fired options. A considerable amount of research work resulting in a number of publications on biomass externally fired cycles and biomass gasification has been performed in the Energy Processes Division at KTH.

The solid fuel fired bottoming cycles, however, whether as a steam Rankine cycle or as an externally-heated air turbine Brayton cycle, have not been extensively and comprehensively evaluated in combination with a topping cycle. Nevertheless, interest in actual construction of such units is growing. Several natural gas and biomass HCC units exist and they will be presented and reviewed in Chapter 6: "State-of-Art of Biomass HCC in Sweden and its neighbouring countries".

In what probably is the only relevant conference paper, Spath and Overend [4.6], 1996, mention a research project undertaken at the National Renewable Energy Laboratory in USA. They present the concept of natural gas fired GT combined with a biomass combustor, together with gasification options. Both fully-fired and parallel-powered arrangements of a GT and bottoming biomass combustion have been considered viable for further evaluation. The cycle concepts are only outlined in the paper, while it is stated that further work will be done on detailed design of the cycle arrangements and on their performance and economic analyses. The authors conclude that biomass combustion combined with a natural gas fired GT is a promising concept, if utilization of biomass as energy source is to be increased in the near future. The cycle configurations, regarding biomass BC combustion (not gasification), proposed by the authors for further evaluation, are presented on **Fig. 4.1** and **Fig. 4.2**. [4.6]

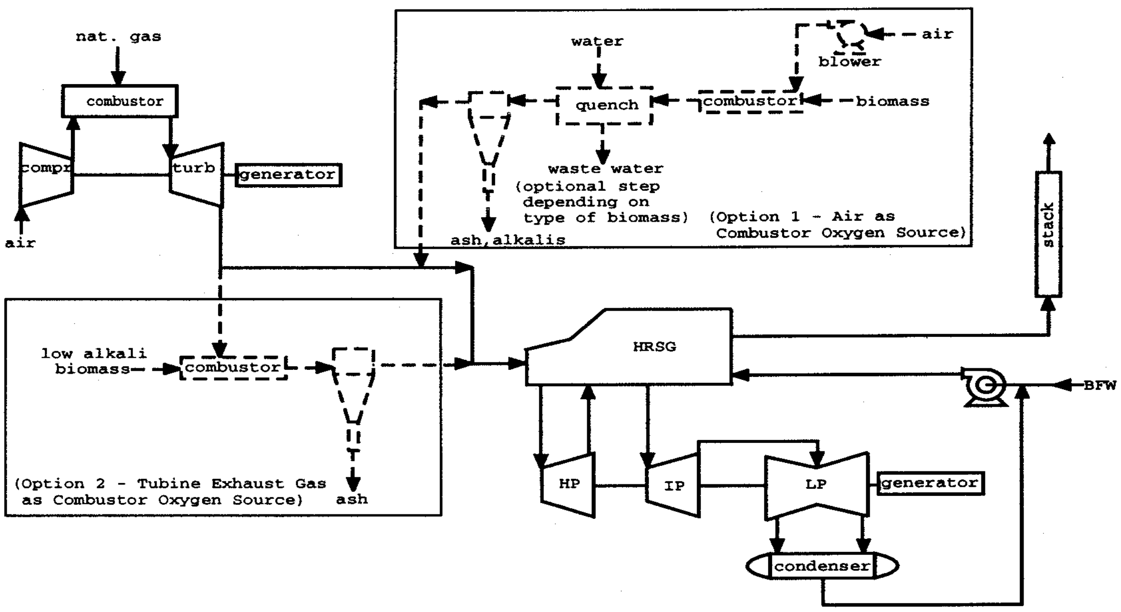


Fig. 4.1: *A suggestion for biomass and natural gas HCC. Both parallel-powered and fully-fired options are shown. Biomass is burned in a simple combustor, after which the hot gases go to the HRSG of the GT. (from [4.6])*

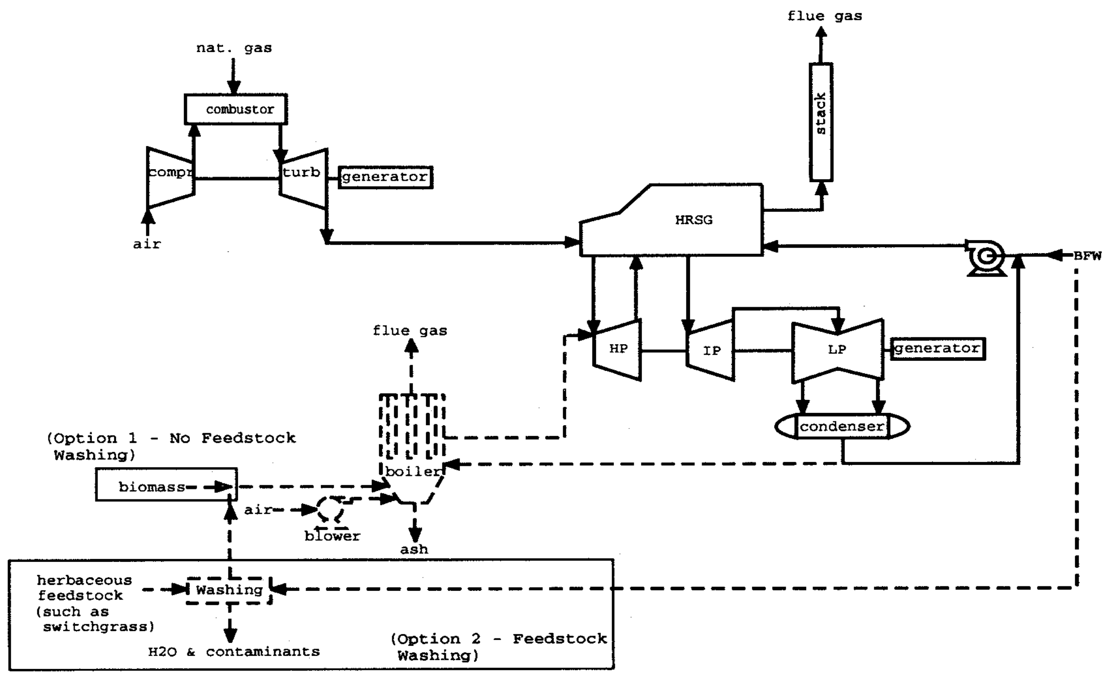


Fig. 4.2: *A suggestion for biomass and natural gas HCC. A parallel-powered arrangement with biomass-fired steam boiler. HP live steam is raised by both biomass boiler and HRSG of the GT. (from [4.6])*

The most relevant work on biomass and natural gas fired HCC systems, though not presented internationally, has been done in the form of industrial reports by Swedish power production companies and in the form of MSc theses (diploma theses) by students at Swedish technical universities.

In an article in the Swedish engineering magazine Mekanisten, Skoglund [4.5] 1996, presents the idea of converting one old oil-fired power station, owned by Stockholm Energi AB (now Birka Energi AB) and working in condensing mode, into a biomass-fired HCC with natural gas fired GT as TC. A parallel-powered cycle configuration in CHP mode with feedwater preheating by the GT exhaust is suggested and examined. Results from technical and economical calculations are presented in the article. A concise flowchart of the cycle is provided. Part-load performance is also considered. The conclusion is positive, indicating that the proposed conversion of the old power units is very attractive, provided that natural gas is available in the Stockholm area and electricity prices grow slightly above present low levels. [4.5]

In a research study at Vattenfall Utveckling AB in Sweden, Bergman [4.1] 1992, estimates the applicability of biomass HCC systems as a CHP units in Sweden. Both fully-fired (series coupled) and parallel powered configurations of both gas turbines and diesel engines as TC have been shortly evaluated, topping a biomass-fired CFB steam generator. The cycles are calculated for 60 MW district heating heat production with 70% share of biomass fuel of the total fuel input. Part-load performance has been an important part of the study. Exact cycle charts and flow values are not provided. TC fuel is light fuel oil or liquefied petroleum gas. Backpressure effects on gas turbine performance are taken into account.
The general conclusion of the study is that HCC systems present only marginal efficiency gain over separate plants with same components and are not attractive. It has been decided to abandon further studies on this concept and to concentrate on indirect-fired GT, powered entirely by biomass. [4.1]

In another more elaborate internal report at Vattenfall Utveckling AB, Westermark [4.7], 1992, presents various HCC configurations (of specific GT models) with various BC fuels, including also biofuels. Different hybrid cycles are suggested, modelled and evaluated. Several gas turbines are considered as TC, including also a short study on advanced GT with steam reforming and externally-fired GT in a HCC cycle. Several exact operational or prospective applications of these cycles are also presented. The gain in efficiency due to coupling the TC to the BC and part-load behaviour is indicated for some of the configurations. [4.7]

Kunert [4.4] in his MSc thesis at the Division of Heat & Power Technology in KTH, Stockholm, 1995, has modelled and calculated various cycle configurations of a GT combined with biomass boiler – a fully-fired HCC, a parallel-powered HCC and a fully-fired HCC with biomass dryer. All cycles are optimized for maximum electrical efficiency at similar power outputs and similar natural gas to biomass fuel ratios. More precise calculations, together with part-load performances and differing biomass to natural gas fuel ratio studies are suggested as future work.
The resulting efficiency gains by combining the GT with biomass boiler and ST are presented in **Fig. 4.3**. The work of Kunert is the closest one to the type of work that will be performed in the project whose first step is this Literature Study. [4.4]

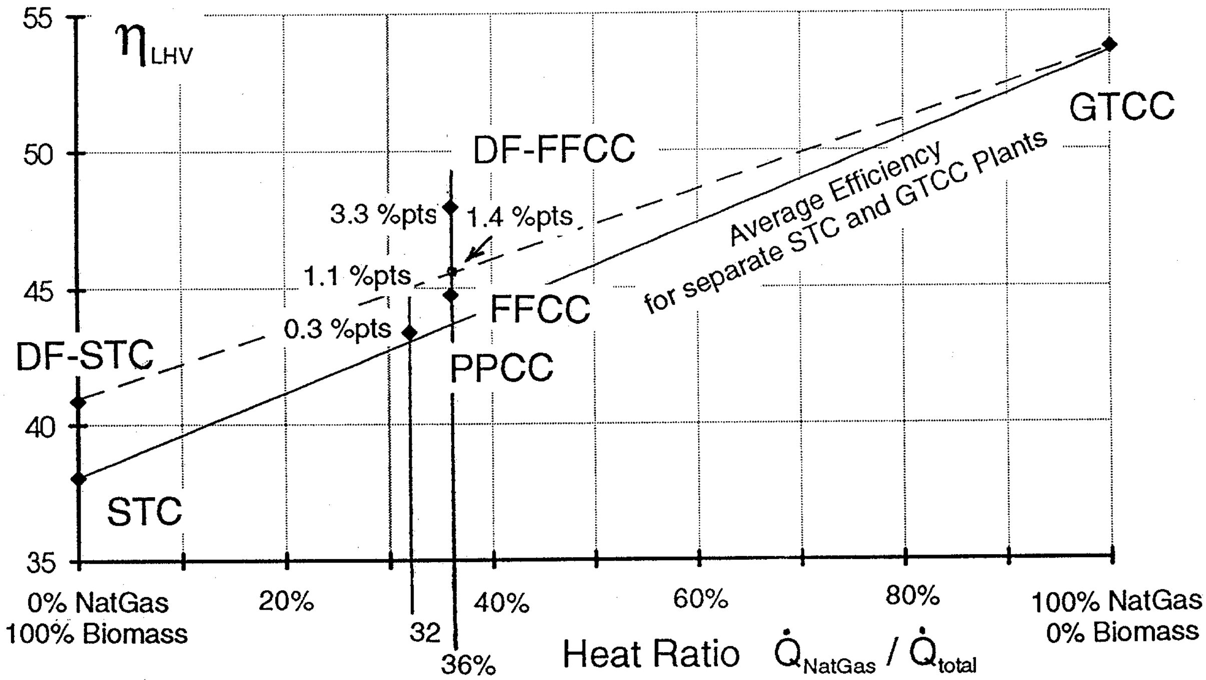


Fig. 4.3: *Efficiency gain (in percentage points) for various biomass and natural gas HCC configurations: Parallel-Powered CC (PPCC – 0.3 %pts), Fully-Fired CC (FFCC – 1.1 %pts) and Fully-Fired CC with Fuel Dryer (DF-FFCC – 3.3 %pts). The expected efficiency of the simple steam cycle with fuel dryer is denoted as DF-STC. All cycles have been modelled with similar overall outputs and similar natural gas-to-biomass fuel ratios, in condensing mode.*
The dashed line shows expected efficiencies for cycles with integrated biomass dryer for varying biomass-to-gas fuel ratios. The cycle with biomass dryer can be expected to show efficiency gain of 1.4 %pts over the mean efficiency of separate steam and GT cycles. The calculated efficiency gain for the cycle with biomass dryer has actually been higher, 3.3 %pts over the mean efficiency line. (from [4.4])

One quite comprehensive and very relevant research work, Egard et al. [4.2], has been performed under Sydkraft Konsult AB in Sweden (later renamed to Sycon Energikonsult AB). The work has resulted in a joint report by Sycon Energikonsult AB and the Department of Heat & Power Technology at Lund Institute of Technology in Lund, 1999-2000. The first half of the report (written by Egard and Steinwall) describes existing HCC plants in Sweden and Denmark, and carefully examines the possibilities and the potential for constructing HCC plants in CHP mode in Sweden. The authors concentrate on parallel-powered configurations, including the concept of repowering biomass-fired CHP plants and hot-water boilers with gas turbines. Exhaustive economic calculations with sensitivity analysis on costs are presented.
In the second half of the joint report [4.2] (written by Håkansson, Assadi and Torisson from the Technical Institute in Lund), thermodynamic calculations of three different HCC CHP configurations are presented, together with sensitivity analysis on the overall performance from varying BC parameters. The configurations studied include a fully-fired cycle, a parallel-powered cycle with parallel steam generation and a parallel-powered cycle with feedwater preheating, all topped by the new GTX100 gas turbine from ABB, now Alstom Power Sweden AB. The configurations are modelled with varying natural gas to biomass fuel input ratio, by varying the BC fuel input. An

attempt is made to evaluate the efficiency advantages of the modelled hybrid configurations compared to the average performance of two separate units based on the given fuel ratio. However, the power output of the simple GT cycle is used in the comparison (simple GT as independent user of the given amount of natural gas), instead of the power output from possible GTCC. [4.2]

Wyszkowski [4.8] in his recent MSc work at the Department of Heat & Power Technology in Chalmers University of Technology, Göteborg, 2001, has modelled and evaluated a fully-fired HCC in CHP mode for industrial applications. Various natural gas fired gas turbines are considered as topping engines, with the bottoming steam boiler being fired by wood-pellets. Calculations are performed with the various TC turbines at varying degree of supplementary firing in the biomass boiler to satisfy a fixed heat demand. The cycles are modelled only in fully-fired configuration without a steam turbine, all heat recovery is used as process steam. The main part of the work is a thorough economic analysis whose purpose is to evaluate the economical feasibility of biomass and natural gas hybrid CHP systems in such industrial applications. The hybrid configurations are compared economically to a simple biomass-fired steam CHP cycle, simple GT CHP cycle and simple biomass-fired steam boiler without a ST.
The results from the economic analysis have led to the conclusion that hybrid CHP cycles are not an attractive option for the industrial conditions investigated. Their costs are higher than those for simple gas turbine CHP units. [4.8]

A very recent study [4.3], performed jointly by senior researchers at the Royal Institute of Technology (Dr. Nabil Kassem) and Chalmers University of Technology (Dr. Simon Harvey) in Sweden, 2001, elaborates further on some interesting topics relevant to biomass-fired HCC development and promotion. A short literature revue on HCC with biomass-fired BC is included in the final report. The core of the study presents an investigation of two important issues: uncertainty (risk) analysis of hybrid CHP systems by Dr. Kassem and cost-effectiveness of CO_2 emissions reduction in industrial hybrid CHP plants by Dr. Harvey.
An industrial CHP model of a GT and biomass-fired bottoming boiler, without a ST in the bottoming cycle, has been used for the simulations/calculations. It is the same model as developed by Wyszkowski [4.8] and described shortly above, with natural gas fired GT topping cycle and wood pellets fired bottoming boiler.
A stochastic modelling tool for identifying, characterizing and handling uncertainties has been applied to the HCC calculation model. It proves to be useful for conducting performance and economic evaluations, risk analysis and feasibility studies of technology concepts like hybrid cycles. In terms of economy and cost-effectiveness of CO_2 reduction, it has been concluded that hybrid CHP systems in industrial application with only heat generation in the bottoming cycle are not cost competitive. A decrease in the investment costs for biofuel firing, decrease in the ratio of biofuel price to natural gas price and revised CO_2 taxation system would lead to increased economic competitiveness of hybrid cycles. It should be noted though, that only industrial CHP applications with bottoming cycle fired by pellets (without electricity generation in the bottoming cycle) have been considered and evaluated in the study. [4.3]

4.2. HCC with MSW as Bottoming Fuel

One particular feature of any waste-to-energy steam boiler is the need to avoid high steam conditions, due to the extremely aggressive nature of the flue gases, which can quickly corrode the tubing. Dangerous substances in the flue gases include hydrogen chloride, molten alkali salts and others. All of them are very harmful to the metal structures at high temperatures. Usually the highest feasible steam superheat temperature is around 400^{o}C, while pressures are generally also quite low.
Furthermore, the flue gas in the heat-recovery path must not be cooled down below 200^{o}C, to avoid condensation of aggressive compounds. To prevent environmental damage, flue gas scrubbing procedures are a necessity, which unfortunately has high penalty on electric efficiency. The result from these constraints is that MSW fired power cycles with steam boilers have very low net electrical efficiencies, not higher than 20-25% at the maximum.

A layout of a typical MSW steam boiler (grate incinerator) is presented on **Fig. 4.4**. Its basic thermal parameters are listed in **Table 4.1**.

Table 4.1: *Typical parameters for a MSW-fired steam power cycle. (Adapted from [4.14])*

HP steam	40 bar, 400^{o}C
LP steam	3.7 bar, saturated
Condenser pressure	0.1 bar
Deaerator pressure	3.5 bar
Minimum stack temperature	200^{o}C
Combustion efficiency	0.96
ST isentropic efficiency	0.85
Net Electric Efficiency (maximum)	24.9% (condensing mode)

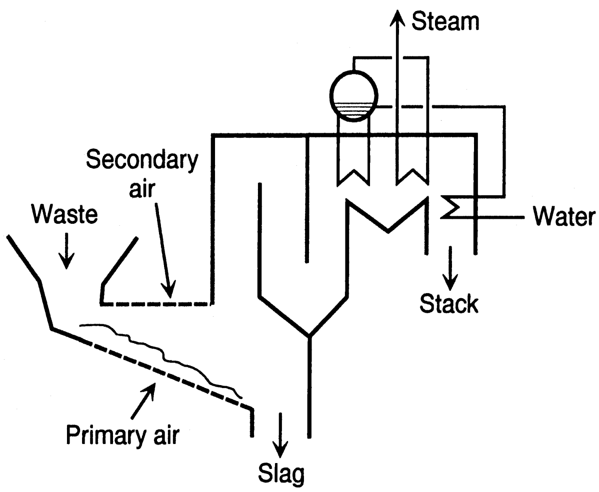


Fig. 4.4: *Simplified schematic of a MSW grate incinerator (steam boiler). Superheater tubes are placed downstream of the evaporator in the hot gas path, usually where gas temperatures are down to 600-650^{o}C, to prevent corrosion. Primary air is fed under the grate to cool it. A layer of unburnt waste also shields the grate. (from [4.14])*

If higher electrical efficiency is sought, the steam can be superheated to higher temperatures (higher energy contents) by external superheating with the help of cleaner fuels. This has been understood long ago and an exemplary scheme for natural gas burner superheating was suggested for example by Eber et al. [4.11] in 1989. They have evaluated thermodynamically and economically a combination of a gas fired superheating heat exchanger coupled after a MSW fired boiler. The energy input in the form of gas has been limited to 25% of the total energy input, as defined by regulations. Results from the analysis have been positive, proving that simple gas superheating of steam generated in a MSW boiler can increase the cycle efficiency enough to be technically and economically viable. [4.11]

In fact, higher electric efficiency has not usually been a priority in waste incineration. Since fuel cost is not normally an issue considered in MSW energy economics, there have been no incentives for increasing cycle efficiency. Instead, simple destruction of the waste has usually been the main purpose of the incinerators.
But this situation is changing. Realizing that MSW is a sustainable energy resource, available in reasonable quantities in every larger community, electric efficiency is becoming an important issue, together with the other merits of burning MSW.

The incorporation of the MSW boiler as a bottoming cycle into a HCC, where a TC can provide superheating and/or preheated combustion air, can improve the efficiency of MSW energy utilization substantially. This improvement is to a much bigger extent than when biomass or coal fired BC is considered. Thus, HCC is particularly attractive for MSW energy utilization.
The high temperature of a gas turbine exhaust makes it well suited to integration with a MSW incinerator. Exhaust heat can be recovered in a HRSG, where further superheating of steam coming from the incinerator takes place. Utilization of GT exhaust as preheated combustion air is another possible scheme, together with a HRSG or not. Combinations of a MSW incinerator with a topping GT into a hybrid combined cycle have been attracting increasing interest since some years. A number of publications have been made, in which various cycle configurations have been suggested, modelled and evaluated.

Chronologically the first suggestion for a MSW and natural gas fired HCC, presented as "combined cycle waste-to-energy", has been put forward by Lowry and Martin [4.15] in 1990. They evaluate a typical arrangement of a gas turbine TC, superheating the steam generated in a MSW incinerator BC. A patent has been granted to the authors for this cycle configuration by the US Patent and Trademark Office. Further in their paper, the authors provide results from their simplified economic calculations and conclude that such power cycle configurations can be economically viable, depending on natural gas prices and tipping fees for MSW disposal, as well as on utility prices of electricity. The share of natural gas fuel is less than 50%. The power cycle is supposed to produce mostly electricity as a saleable commodity, with a very small amount of process steam delivered to prospective consumers, which still qualifies the plant as a cogeneration facility to satisfy the regulatory standards. [4.15]

Korobitsyn et al. [4.14], 1999, have presented and evaluated HCC configurations, where a GT has been topped over a MSW steam boiler with the parameters from Table 4.1. Three different cases (three different HCC configurations of a MSW boiler and natural gas fired GT) have been carefully modelled and examined. The layouts of the various cycle configuration cases are presented in **Fig. 4.5**, **Fig. 4.6** and **Fig. 4.7**.

The results are compiled in **Table 4.2**.

Table 4.2: *Summary of results from MSW HCC modelling. (Adapted from [4.14])*

	Reference	**Case 1**	**Case 2**	**Case 3**
Fuel input, MW_{th}:				
- incineration boiler	92	92	92	92
- gas turbine	0	100.14	48.44	104.73
Total	92	192.14	140.44	196.73
MSW share, %	100	47.88	65.51	46.76
Natural Gas share, %	0	52.12	34.49	53.24
Power Output, MW_{el}:				
- steam turbine	22.92	46.46	36.89	47.79
- gas turbine	0	31.96	15.48	33.44
Total power output	22.92	78.42	52.36	81.23
Efficiency, %				
- based on total fuel input	24.91	40.81	37.28	41.29
- based on MSW	24.91	28.64	29.54	29.10
Specific heat exchange surface area, m^2/MW_{el}	294	710	340	518

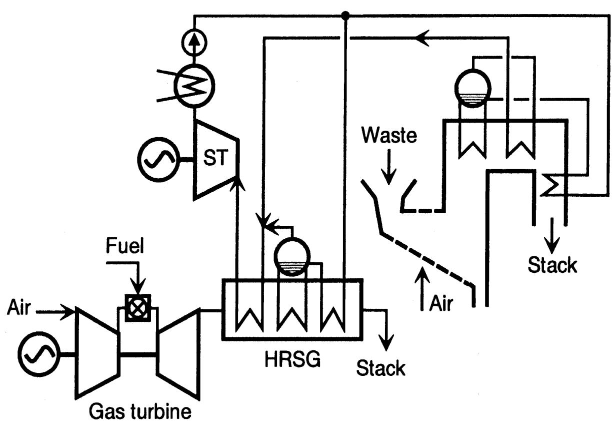


Fig. 4.5: *MSW and GT HCC in parallel configuration. Case 1 in Table 4.2. (from [4.14])*

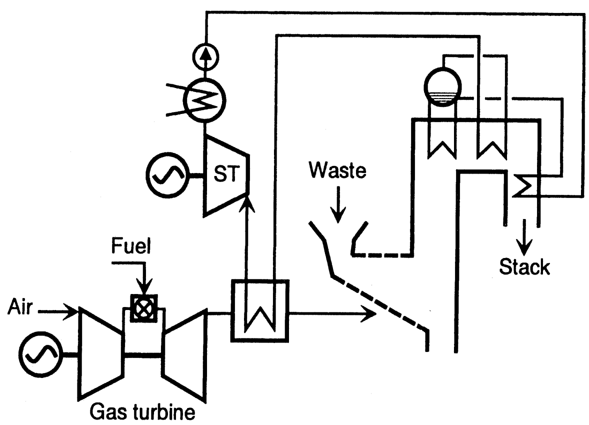


Fig. 4.6: *MSW and GT HCC in hot windbox configuration with superheating from the GT exhaust. Case 2 in Table 4.2. (from [4.14])*

Korobitsyn and co-workers have specifically investigated and pointed out the increase in efficiency of MSW boiler and GT HCC, compared with the average value between the MSW boiler reference case and a pure CC plant operating only on natural gas. Such a comparison is illustrated on **Fig. 4.8**. The results from Table 4.2 are compared in this respect in the figure.

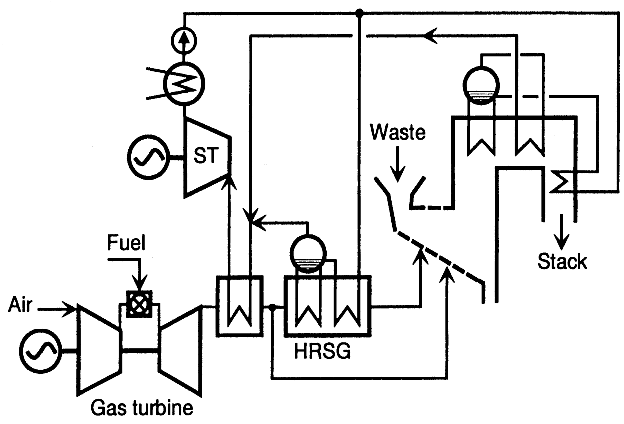


Fig. 4.7: *MSW and GT HCC in cold windbox configuration. GT exhaust is used for steam generating parallel to the MSW boiler and steam superheating. Case 3 in Table 4.2. (from [4.14])*

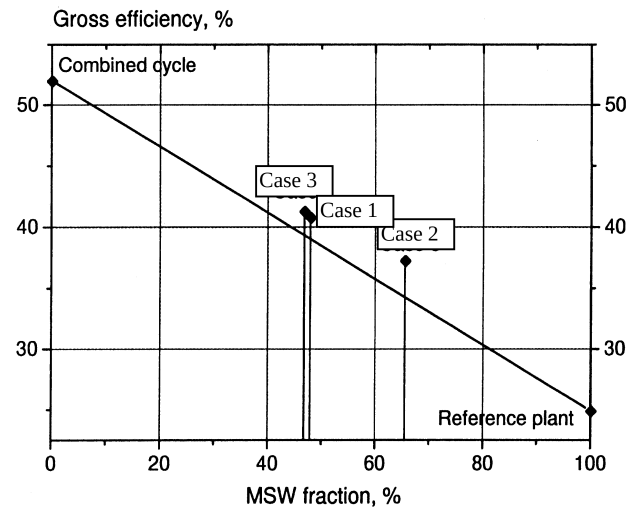


Fig. 4.8: *Efficiency gain for the MSW and GT HCC. Comparison of results shown in Table 4.2. (Adapted from [4.14])*

Case 1 and Case 3 (**Fig. 4.8**) show efficiency advantages of 2.2 and 2.3 %points respectively. Case 2 shows 3.6 %points efficiency advantage with even higher ratio of MSW-to-GT fuel input. It can be concluded that the cycle in windbox configuration (fully-fired) with superheating in the GT exhaust, combined with steam generation in the MSW incinerator, promises the highest increase in efficiency over the average of two separate power plants, despite its high share of MSW fuel input to natural gas fuel input (65%).
Another important result shown in Table 4.2 is the increase in efficiency of MSW energy utilization itself in the HCC, compared to the reference case (simple MSW-fired steam cycle). Case 2 features the highest increase. Namely, the increase in efficiency of bottoming fuel utilization is one of the biggest advantages of all hybrid combined cycles. The calculation of this efficiency increase is somewhat arbitrary and will be further discussed in Appendix A.3 "A short discussion on cycle efficiencies".
The authors also examine the increase in heat exchange surface area in each of the evaluated cycle configurations. Again Case 2 is most advantageous, showing only 15% larger heat exchange area than the reference. The configuration from Case 3 has 76% larger heat exchange surface area than the reference case. [4.14]

Consonni [4.10], 2000, also suggests MSW energy utilization cycle configurations with a GT as a TC (waste-to-energy CC systems, shown on **Fig. 4.9**) and evaluates their performance. He has modelled a parallel-powered combined cycle layout, mentioning the advantage of keeping the two flue gas flows apart. This allows the GT exhaust temperature to be lowered to its lowest permissible value, far lower than the MSW flue gas temperature. Compared to fully-fired arrangements, flue gas flow through the gas-treatment system is less (only from the incinerator), which allows savings of energy and investments. Fig. 4.9 shows the most complex arrangement modelled by the author, with two steam pressure levels plus reheat in the HRSG.

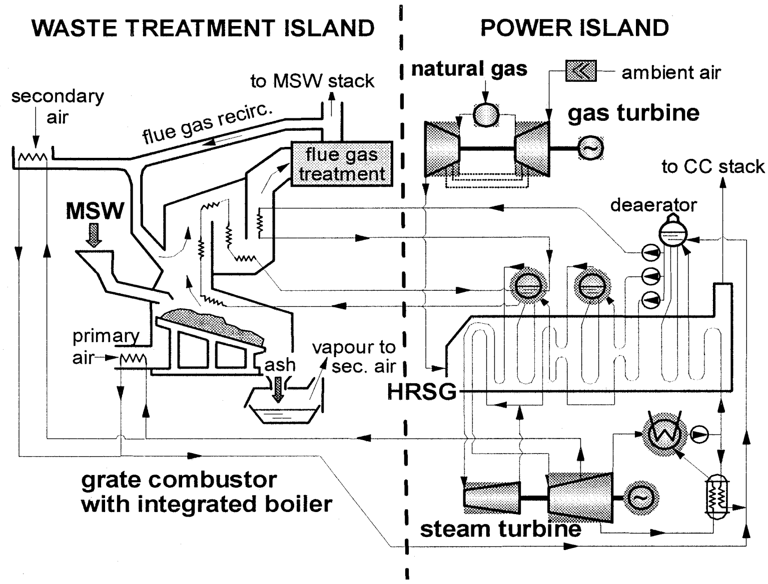


Fig. 4.9: *A MSW and natural gas waste-to-energy HCC schematic. Parallel-powered arrangement with combustion air heated by steam extraction from the ST. Two-pressure level steam cycle with reheat. (from [4.10])*

Consonni also stresses the importance of estimating the increase of MSW energy utilization itself, or in other words the efficiency attributable to MSW only, within the HCC. This is again discussed in Appendix A.3.
The results from the calculations show that such HCC can reach close to 36% net electric efficiency based on MSW, in the two pressure plus reheat configuration with 100 bar HP steam and burning more than 150'000 tons/year MSW. Compared to the simple MSW steam Rankine cycle, the increase in efficiency of electricity generation from MSW itself within the HCC is about 1.5 times, a remarkable value. Such a cycle configuration can provide good efficiencies based on MSW even at very small scales. If the live steam pressure is lower (around 65 bar for conventional MSW-fired boilers), the MSW-based efficiency is still over 30%, varying with scales. The MWS-based efficiencies of the less complicated HCC arrangements (one steam pressure level in the HRSG or two steam pressure levels without reheat) are above 30%, unless the scales are very small.
Consonni has also performed in his study a thorough cost calculation and economical analysis of MSW energy utilization plants of different capacities for European conditions. He points out that one Spanish engineering company claims to hold a patent on a cycle configuration, similar to the one modelled in his study, and that such a power unit is being built in Spain (2000). [4.10]

Otoma et al. [4.16], 1997, have performed a life-cycle analysis for the production of electric power based on MSW incinerators in Japan. They have evaluated the actual power output from the MSW plant and the actual reduction of CO_2 emissions, taking into account the in-house energy consumption, energy for plant construction and O&M, energy for construction of MSW collection vehicles and fuel consumption for collection of MSW from a given area. The results from their calculations show that the energy production from the waste-to-energy plant is 9.5 times higher than the energy

involved in construction, operation and maintenance over the whole life-cycle of the facilities. Similarly, the avoided CO_2 emissions were estimated to be 4.1 times higher than the emitted CO_2 from construction and collection vehicle operation.
Otoma and co-workers also evaluate two options for topping a MSW boiler with a gas turbine, having a base case MSW boiler with very low steam characteristics: pressure of 27.5 bar, temperature of 300^{o}C and electric efficiency around 15%. [4.16]

In fact, Japanese waste incinerators with steam turbines have traditionally been designed with very low steam parameters, temperatures of 300^{o}C at most, in order to diminish as much as possible the tube corrosion. Consequently, power generation efficiency in Japanese waste incineration plants is around 20% at best.
Corrosion from flue gases (hydrogen chloride mostly) starts to proceed quickly at 320^{o}C or more. However, steam parameters in waste-fired boilers in Europe reach 380^o–400^{o}C, as mentioned above (see also Table 4.1). This has probably been a result of the higher incentives in Europe for utilization of MSW energy with better efficiencies. Consequently, shorter lifetime for the superheater must be tolerated in MSW-fired units with higher efficiencies [4.19]. A stable trend towards elevated steam parameters is also present in Japan now, with new plants already working at higher steam temperatures and pressures and a development plan under way to achieve 30% power generation efficiency with simple Rankine MSW-fired cycles. [4.18]

Ito et al. [4.13], 1996, have performed a careful mathematical analysis of the economic and energy characteristics of a MSW boiler topped by a GT, in cogeneration mode (called by them a "super waste incineration plant"), for Japanese conditions. [4.13]

There have been three MSW incinerators topped by gas turbines in operation in Japan, as of 1997. [4.16]

Wiekmeijer [4.19] was among the first to propose and evaluate HCC configuration of MSW incinerator with topping GT in 1990. He stresses on the parallel-powered cycle arrangement with economiser and final superheater in the HRSG behind the GT. He also reviews some financial aspects of such applications and points out that the price for the additional power is very low, especially compared to the very high specific investment costs per unit generated power in simple MSW-fired steam cycle. [4.19]

Alternative options have also been suggested for topping MSW-fired steam boilers. For example, steam-injected gas turbines can be expected to fit well in MSW-fired power units, simplifying the cycle arrangement [4.17].
Coal- or oil-fired steam boilers in parallel configuration with a MSW incinerator are other possible arrangements. For example, steam generated in a MSW boiler can be fed to a coal-fired steam cycle, where superheating and expansion in the ST together with the coal-generated steam can easily take place [4.18].
Biomass-fired steam boiler can also serve as a superheater for MSW-generated steam in a parallel configuration.

Certain research and promotion of hybrid configurations for MSW incinerators has been taking place also in the form of student projects and MSc theses, for example in Italy [4.10].
One Swedish example is a MSc thesis, Holmgren R. [4.12], performed under Sydkraft Konsult AB in 1998. Thermodynamic calculations with sensitivity analysis of overall

performance on BC parameters and economic evaluations with sensitivity analysis of costs are presented in the thesis. Three different HCC configurations of a GT combined with a MSW-fired boiler are carefully modelled in CHP mode: a fully-fired cycle, a parallel-powered cycle with parallel steam generation and a parallel-powered cycle with parallel steam generation and final superheating of all steam in the GT exhaust. Different scales of MSW incineration are considered, relating to different amount of utilized waste per year from urban centers of different sizes. Investigation of increased superheating temperature and pressure for all different configurations has been done with the aim to find an optimum performance, despite the fact that only one of the configurations use final superheating in the GT exhaust, while higher superheat temperatures in the MSW incinerator are not economically justified. An attempt is made to evaluate the advantages of the modelled hybrid configurations by comparison with existing MSW-fired CHP plants and GTCC plants. [4.12]

A recent report by Bartlett and Holmgren K. [4.9], 2001, describes the Gärstad waste incineration hybrid CHP plant in Linköping, Sweden, and investigates the feasibility of converting the gas turbine cycle into an evaporative GT cycle while raising the steam pressure, thus improving the performance. The statements made in the report are supported by extensive simulations and calculations of the cycle modifications. An insight into the effect on CO_2 emissions and district heating load is also provided [4.9]. The Gärstad CHP plant was constructed in 1994 and is a perfect example of a MSW incinerator and gas turbine hybrid combined cycle, where the GT exhaust is used for superheating of all steam. A comprehensive description of its configuration is included in Chapter 6.

5. SPECIAL ATTENTION TO INTERNAL COMBUSTION ENGINE COMBINED CYCLES (ICECC)

The issue of combined cycle applications for ICE has always sparked uneasy debate. Being widely used prime movers with good thermal efficiency, this very features of the ICE that make them unrivalled in most industrial, transport and CHP applications, actually are the reasons for their unpromising performance in combined cycle mode.

Internal combustion engines are traditionally employed in numerous applications, with power outputs from less than 1 kW up to 40-50 MW per unit. They are reliable and well-developed prime movers, featuring the highest thermal (mechanical) efficiency of all simple-cycle heat engines. As compared to the gas turbines, piston engines rule in the low-power range, up to several MW, where their efficiency is much higher than that of a GT with similar output. The ICE does not have the scaling possibilities of the GT, it has much lower power-to-mass and power-to-volume ratio, grows very quickly with growing power output and its maximum size is limited. The largest ICE units available today have power output of around 50 MW.

If a simple comparison between gas turbines and internal combustion engines must be made, it can be concluded that:

In terms of thermodynamic performance, ICE has better thermal efficiency, which does not vary so largely with size as is the case with GT. ICE has heat losses with cooling media in several low-temperature streams, with relatively small amount of heat in the exhaust gas. GT has all its heat exhausted with the exhaust gases. ICE show less (close to none) decrease of power output with rising ambient temperature than GT, but are more prone to output power deterioration with increasing backpressure (pressure losses in heat exchangers in the exhaust gas path). ICE has the best part-load performance of all heat engines as well. Oxygen content in the exhaust gases of ICE is lower than in GT exhaust. Typical figures lie around 10-13 %v O_2 in ICE exhaust, compared to 15-18 %v O_2 in GT exhaust.

In terms of economy, ICE has still lower price per installed kW than GT, but slightly higher operation and maintenance costs. ICE lifetime between major overhauls is slightly longer than that of a GT. Construction complexity of ICE is higher than that of GT, but ICE has less expensive materials incorporated in its structure and generally less energy is involved in the production of ICE than in GT, measured per kW rated power output of the engine. This is also due to the established mass production of internal combustion engines.

In terms of pollutant emissions, internal combustion engines produce considerably more NOx than modern gas turbines. Contemporary techniques for lowering NOx (lean-burn and special tuning) achieve lower emissions, yet still higher than GT combustors. In many cases, engines must be equipped with NOx reduction units in order to meet stringent emission regulations.

In the small-scale power range, up to several MW, internal combustion engines show much better performance and economy than gas turbines and even than GTCC.

Internal combustion engines are particularly well suited for CHP applications. Various configurations are possible, depending on the type of heat recovery and the end user of the heat recovered. Most of the stationary engines for electricity production, industrial mechanical drives and ship propulsion, work in cogeneration mode with heat recovery as hot water and/or low pressure steam. Many engines in the cold climate regions supply heat to district heating networks. An example is shown on **Fig. 5.1**.

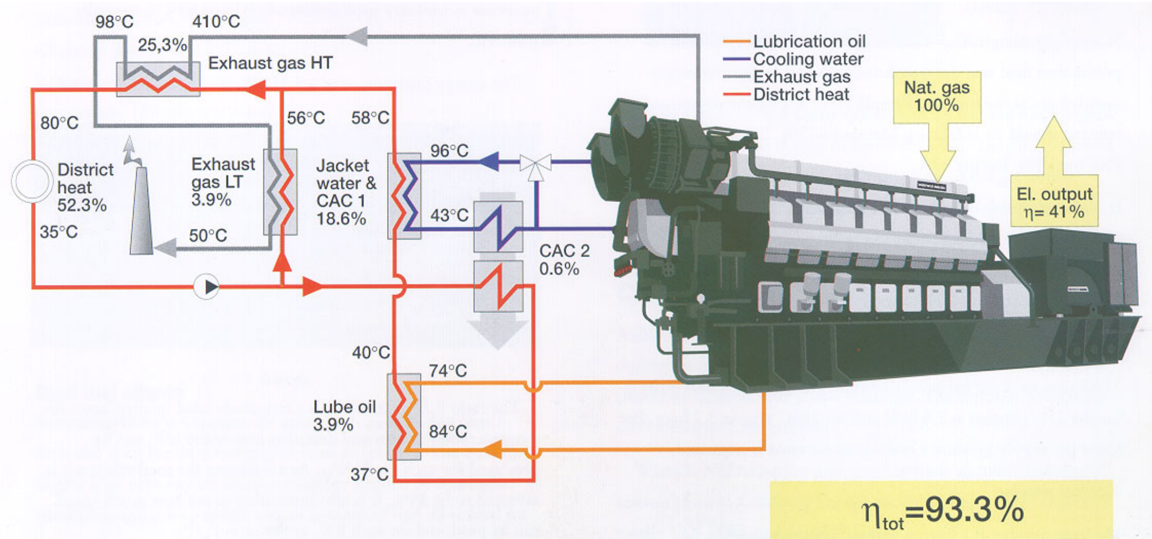


Fig. 5.1: *A typical example of a CHP cycle for ICE. (from [5.30])*

Steam can also be generated by exhaust heat recovery from ICE. Of course, part or all of it can be for example expanded in a steam turbine to produce additional power. The ST together with the HRSG and other auxiliary equipment would comprise a typical bottoming cycle.

The high thermal efficiency of the ICE leads to comparatively low exhaust gas temperatures, which do not allow superheating in a bottoming steam cycle to a level enough for high efficiency performance. The other typical feature of the ICE, namely the existence of several low-temperature heat-rejection streams with cooling media (jacket water cooler, lube oil cooler and charge-air intercooler), decrease significantly the amount of heat carried by the exhaust gases and do not allow their energy to be used in a power cycle with reasonable efficiency, due to the low temperatures.

As an example of the diversity of heat rejection flows from a typical stationary internal combustion engine, the real case with a Wärtsilä 18V28SG engine similar to that on Fig. 5.1 could be considered.

The Wärtsilä 18V28SG is a gas-fired spark-ignition workhorse for stationary power generation, with 4.6 MW rated electric output. Turbocharging of the combustion air with two-stage intercooling is applied.
There are four important heat exchangers, providing necessary cooling at low temperatures for the engine. These are:

- jacket water cooler,

- lubrication oil cooler,
- first stage charge air intercooler and
- second stage charge air intercooler.

The jacket water cooler and lubrication oil cooler are crucial for the engine, without them the engine would not be able to operate at all. The two charge air coolers are not so critical for the engine’s operation, but are important for increasing the mechanical efficiency. The first stage charge air cooler (CAC 1 on Fig. 5.1) has the purpose to cool the charge air from 152^{o}C after the turbocompressor down to 75^{o}C. This temperature is very close to the temperature of the jacket water fed back to the engine after the water cooler (where it should be cooled from around 96^{o}C down to 72^{o}C), so the two heat exchangers are often interconnected and unified under the name “high temperature coolers”. The lubrication oil cooler rejects heat at 84^{o}C down to 74^{o}C level. The second stage charge air cooler (CAC 2 on Fig. 5.1) rejects heat at a very low temperature level (35^{o}C - 40^{o}C), the rejected heat is usually unusable for other purposes and must be dumped to the environment. Sometimes the CAC2 and lube oil cooler are unified under the name “low-temperature coolers”.
The exhaust gas itself has quite a low temperature (around 400^{o}C for spark-ignition engines and less than that for diesels).

In general, the heat rejection streams and energy losses out of the engine carry away amount of heat as follows (expressed in percentage of the total fuel energy input):
- Exhaust gases – 28,5%
- Jacket water – 10%
- Lubrication oil – 6%
- Charge air cooler first stage – 5.8%
- Charge air cooler second stage – 2.3%
- Unburned fuel – 2.9%
- Radiation to engine room – 1.6%
- Fuel input – 100%
- Shaft mechanical power – 43%
- Generator losses – 1.5%
- Electric efficiency – 41.5%

In contrast to gas turbines, where nearly all rejected heat is contained in the exhaust gas stream at temperatures usually beyond 500^{o}C, internal combustion engines do not have the possibility to feed energy into an efficient bottoming cycle. However, almost all heat rejection can be utilized for district heating purposes for example, without any difficulties.
This, together with the comparatively small sizes of the ICE, makes them unfeasible for unfired combined cycles applications. Both the performance of the bottoming cycle and the expenses for constructing it in the sizes dictated by the topping engine size, lead to an undisputedly low profitability, which cannot justify the investments.

However, the sharp increase of oil prices several decades ago changed this situation. Suddenly, all industrialized countries became interested in waste heat utilization from piston engines and numerous research projects were started. The largest amount of research and development work was indeed directed to increasing the efficiency of the engines themselves, but various bottoming cycle options for ICECC configurations were considered as promising alternatives and were carefully investigated.

The "big boom" of this research took place right after the first oil crisis in the 1970-ies when the first serious technical and economical feasibility evaluations of ICECC bottoming cycles were carried out, both for stationary and transportation engines. The interest in this topic and the work on it hasn't stopped ever since.
Apart from research organisations and energy-planning bodies, most engine manufacturers have also entered this field themselves and have largely started to develop bottoming cycles. Today, all large stationary engine manufacturers can deliver turnkey projects in CC mode.

Development of the ICECC concept started in general with careful consideration and evaluation on the viability of different bottoming cycles. The primary objective has been evaluation of their thermodynamic performance in connection with their economical feasibility, i.e. whether they justify the investment costs by providing fuel savings and how long the payback period would be. This was of course based on the fact that price per kW installed power in the BC is far higher than the price per kW of the main engine, while the BC contributes only a miserable percentage of the total CC output. BC arrangements are viewed by the user as incremental investments, which are attractive only if they have a shorter payback period than the base plant. Less than three years payback period and return on investment above 15% are usually regarded as limiting values.
Such studies, in one form or another, were performed in Europe, USA, Japan and many other countries. Various bottoming concepts were developed and evaluated. Examples are [5.17], [5.6], [5.29], [5.35].

In the USA, applicability of several different BC configurations to truck engines has been examined. Long-haul truck engines have been regarded as the most promising acceptors for wide-spread use of BC systems, most amenable to waste heat recovery due to constant driving modes with long periods of quasi-stationary power loads and a possibility for economy-of-scale due to potentially large market. [5.17], [5.6]

Turbocompounding, steam Rankine, organic Rankine and air Brayton cycles have been investigated as bottoming cycles in pure unfired ICECC configurations. More complicated BC arrangements (multiple pressure levels in Rankine cycles, intercooling or subatmospheric expansion in Brayton cycles), gave higher energy recovery, but of course higher system complexity and higher investments were necessary. In general, the studies led to the conclusion that if higher prices are stable for long time on the fuel market, the BC could be economically justified. Raising the temperature of ICE exhaust and diminishing as much as possible the heat loss in low-temperature cooling was believed to be the future trend in ICE development, which would allow higher efficiencies and better payback for the BC investment.

Turbochargers became standard components of large stationary, marine and truck engines. With the improvement of the turbocharging equipment, available power in the engine exhaust became more than the necessary power for intake air compression. It became possible to utilize a certain part of the turbocharger mechanical power as additional mechanical output for the main engine shaft. Gradually, the turbocompound concept was developed and used in large stationary and marine diesels.
Turbocompounding has also been considered as bottoming cycle, although it is actually not a real waste heat recovery. Separate power turbine can be installed after the turbocharger turbine, expanding further the exhaust from the turbocharger turbine

or expanding completely its own portion of the main engine exhaust, which in the absence of a power turbine would be vented through the waste gate. Otherwise, a larger turbocharger can be used, which supplies mechanical power in excess of the power needed for compression of the charge air, without a separate power turbine. **Fig. 5.2** shows the evolution of turbocharging and turbocompounding with an exemplifying comparison of the efficiency advantages that they bring.

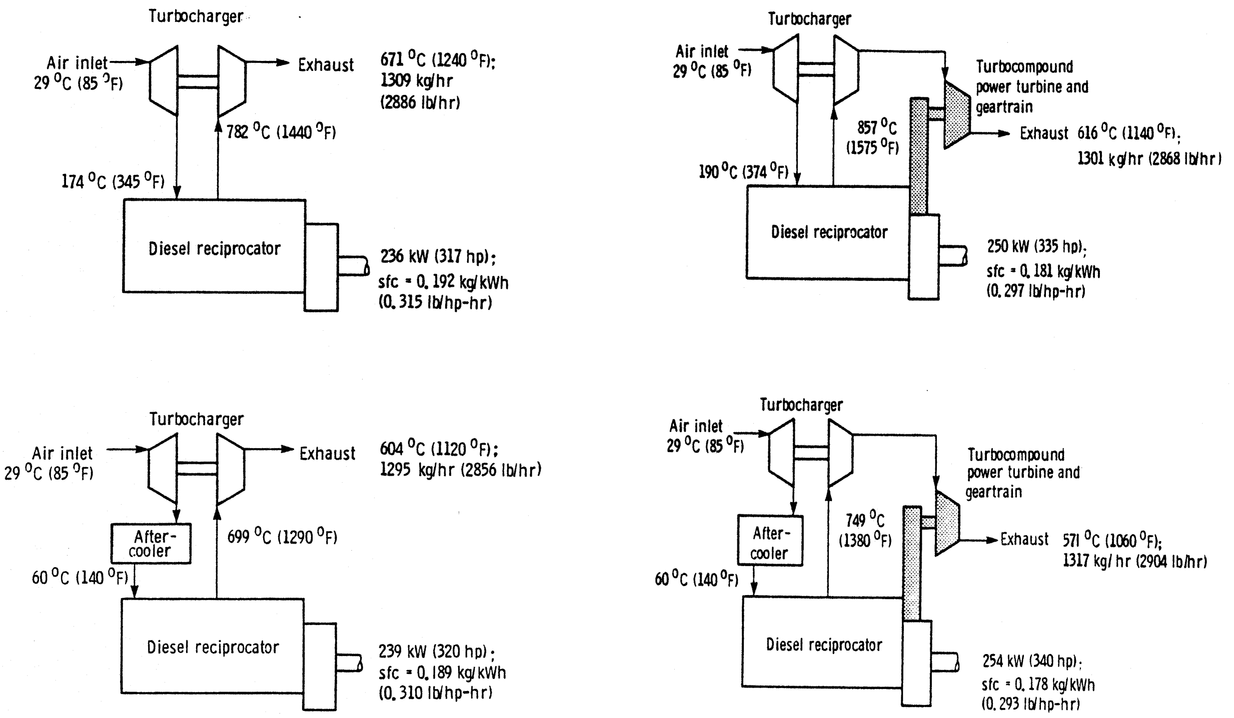


Fig. 5.2: *ICE with turbocharger (left) and turbocompound system (right). Comparison of performance and efficiency improvement attainable. (from [5.6])*

Turbocompounding alone cannot bring big savings in fuel consumption, i.e. its potential for efficiency increase is limited. It is, however, the simplest and the cheapest method for direct utilization of some of the energy in the pressurized exhaust gas of the main engine. A separate power turbine uses a relatively small (10-15%) part of the exhaust gases, resulting in small size and moderate performance. Additional output from the power turbine in general does not exceed 2.5-2.8% of the main engine output, with correspondingly fairly small savings in fuel consumption [5.28].
Coupling the power turbine to the main engine shaft can only be accomplished via reduction gearbox. This applies also to any BC turbine. Due to the small sizes, the rotational speed of the bottoming turbines is far higher than the main engine shaft. Gearboxes add cost and complexity, while decreasing efficiency and reliability of the system. In most actual ICECC configurations, the bottoming turbines are connected to their own electric generators, again via gearboxes.

Organic fluid Rankine bottoming cycles have generally been pointed out as most promising. Steam Rankine cycles have also been considered very attractive. Practical ICECC applications have been commissioned with both organic and steam BC. Rankine cycles do not interfere with the operation of the main engine (apart from the

small pressure loss in the HRSG) nor deny the application of turbocharging. Turbocompounding supplemented by Rankine cycle would probably give the higher BC power output among all alternatives.
Organic fluids are more expensive, some of them toxic, require larger heat exchange surfaces and higher flow rates for same power output as compared to steam [5.9]. Their properties may change with time and they are generally unstable at temperatures higher than 380-400^{o}C. Higher flow rate of organic fluids however ensures better performance (smaller losses) in the expansion turbine, compared to steam. In the small sizes under consideration, turbine losses are a major portion of the losses in the BC. In addition, organic fluid turbines have smaller number of stages than steam turbines, which leads to smaller and cheaper expansion equipment [5.31], [5.32]. Organic fluids' potential for better heat recovery and higher additional power still receives attention.

A Japanese team [5.23] has suggested, evaluated and experimentally tested an organic Rankine BC for gasoline car engines. Their main objective has been to design an extremely compact heat recovery system, suitable for application in private cars. The considerations for compactness have led to the idea of recovering the heat only from the engine cooling water. This decision is also based on the fact that small gasoline engines usually reject more heat in the cooling water than in the exhaust gas, especially in part-load. Direct evaporative cooling of the engine with the organic fluid can be employed, using the place of the standard water cooler for a BC condenser. Steam is generated directly in the engine, expanded in a compact scroll expander, condensed and pumped back to the engine, **Fig. 5.3**. The scroll expander is the only additional component. The authors have calculated and tested the BC performance for different BC pressure ratios and different regimes of the main engine. The results show that approximately 3% increase in engine output can be achieved with such a BC concept. Of course, BC output can be further improved by superheating from engine exhaust gases, but this requires a bulky heat exchanger (superheater with recuperator or larger condenser if recuperator is not used) and will violate the idea for maximum system compactness. [5.23]

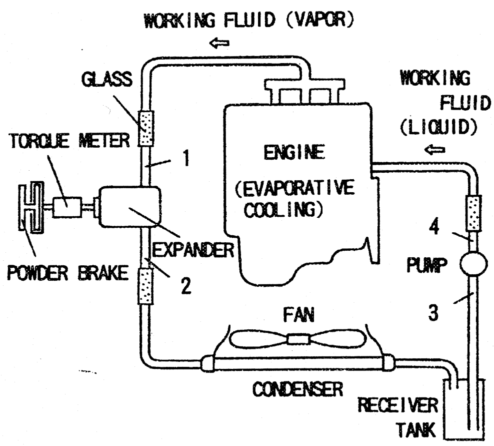


Fig. 5.3: *Proposed evaporative engine cooling system with organic Rankine BC. (from [5.23])*

Organic Rankine cycle optimized for maximum heat recovery can provide much higher additional power output than the cycle on Fig. 5.3. Evaporation is performed by the engine cooling water in a vapour generator, after which the organic vapour is superheated by the engine exhaust gases. A recuperator is very important for increasing the efficiency of the BC. Two recuperating heat exchangers can be included, before and after the evaporation. Such an arrangement, bottoming a bus diesel engine, can supplement the base engine power with 16% of its output. [5.4]

Calculation and experimental results for a steam BC optimized for maximum heat extraction from all heat rejection streams of the main engine has been presented in [5.9]. The cycle layout is shown on **Fig. 5.4**. The energy distribution before and after installing the BC is shown on **Fig. 5.5**. The dual-pressure steam cycle has provided 14% increase in power of the main engine (4.5 %points). Reduction in fuel consumption has been 8% to 13% over a wide load range. Further work on standardisation, scaling-up possibilities and skid-mounting of the BC equipment has been undertaken. [5.9]

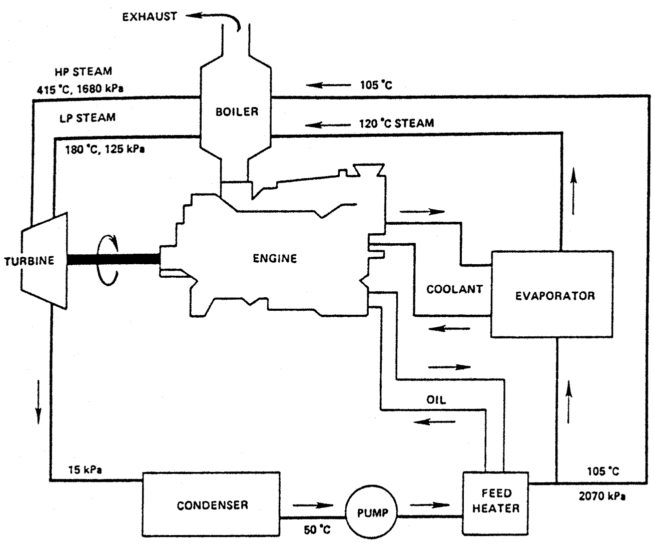


__Fig. 5.4:__ Cycle schematic of an optimized steam Rankine BC system for stationary ICE. Double pressure steam generation. (from [5.9])

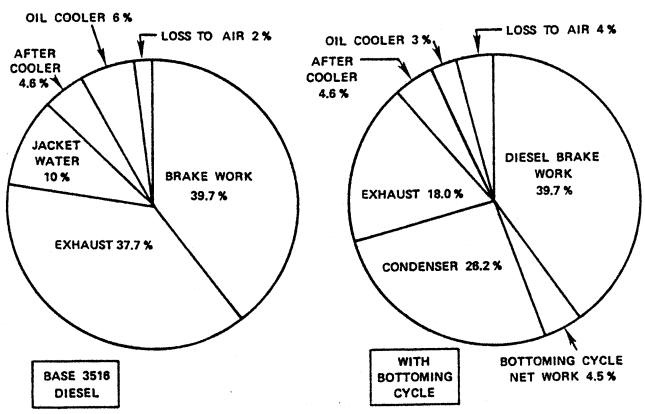


*__Fig. 5.5:__ Energy distribution for the engine from Fig. 5.4 before and after installing the bottoming cycle.
(from [5.9])*

Organic bottoming cycle with flash steam generator in the low pressure section has also been suggested and practically implemented [5.29]. The system has lower cost, due to substitution of part of the heat exchange surfaces with flash-steam producer, **Fig. 5.6**. An arrangement of this kind adds around 10% to the output of the parent engine. A certain part of the heat is available for heating purposes. [5.29]

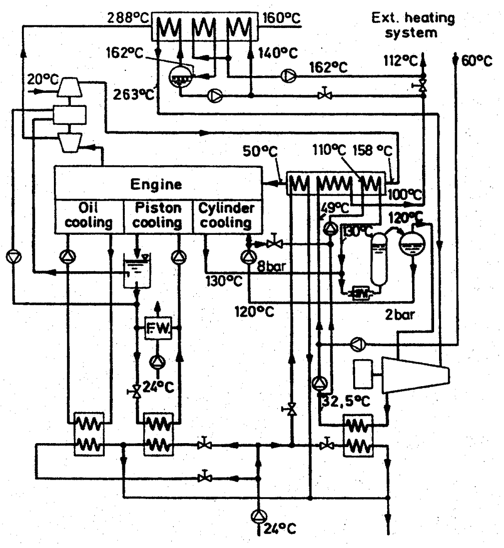


Fig. 5.6: *BC system with optimized exhaust heat recovery and flash steam generator for marine engines. The potential for enough heat supply in CHP mode must not be lost for such marine engines. (from [5.29])*

Alternative configuration of a steam Rankine BC, where heat of ICE exhaust is utilized for superheating before the gas expands in the turbocharger, is presented in [5.33]. The decrease in fuel consumption achieved with such configuration is around 20%.

Ammonia-water mixtures as fluids for ICE Rankine BC have also been suggested. A relevant study [5.14], highlighting the improved heat recovery and larger BC power generation potential of ammonia-water mixtures compared to steam, has been performed at the Division of Energy Processes, Royal Institute of Technology, Stockholm. The study extends previous work by various authors on development of bottoming cycles with ammonia-water mixtures for gas turbines, into an investigation of the potential for waste heat recovery from all heat rejection streams of a gas-diesel ICE. It has been concluded that the exergy efficiency of the best ammonia-water cycle configuration is 43-48% higher than the efficiency of a single-pressure steam cycle and 20-25% higher than that of a dual-pressure steam cycle. [5.14]

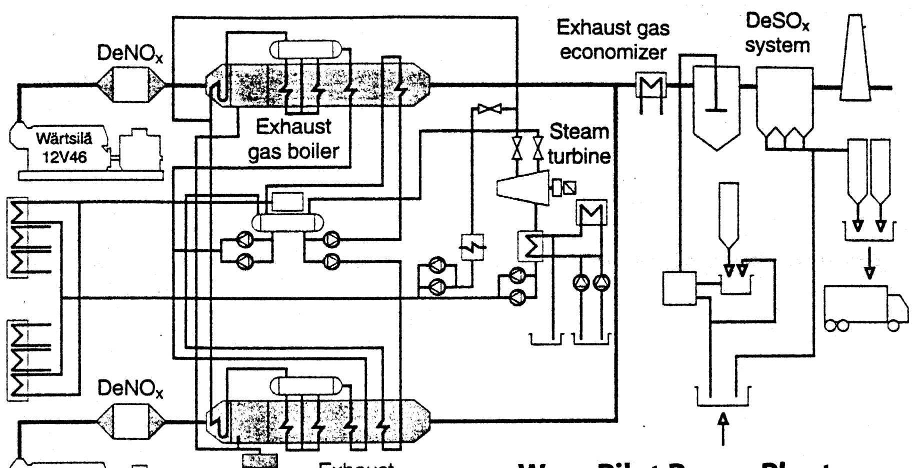
Many case studies on practical ICECC configurations commissioned have been published, apart from theoretical evaluations and experiments or together with them. Examples are [5.31], [5.32], [5.29], [5.19]. Interesting operational experience has been reported also.

Soot fouling of the heat recovery steam exchangers in the exhaust gas can be a problem. Spontaneous fire in the soot layer is possible, so cleaning methods must be considered or improved parent engine combustion must be implemented. One advantage of the comparatively low temperature of the engine exhaust is the fact that heat exchange surfaces can be left dry (uncooled) in the exhaust flow without the risk of overheating.

Meanwhile, development of new ICE models with greatly improved efficiencies is continuing. Contemporary engines have higher simple-cycle efficiency than older engines with bottoming cycles. The number of installed large stationary engines for power generation or marine propulsion is constantly on the rise. Together with growing sizes of new engines, the potential for waste heat recovery increases.

Nowadays, almost all ICE manufacturers and turnkey power plant suppliers are considering the ICECC arrangement as a promising, economically profitable alternative. A number of such power plants exist in the world, where a bottoming steam turbine utilizes all or part of the heat rejected from the topping ICE for additional power generation, with or without a turbocompound system, achieving up to 7% additional power output by the BC. Usually, several engines with a HRSG each, are coupled to one ST. [5.22]

A typical example is the Wasa diesel ICECC power plant, build by Wärtsilä in Vaasa, Finland. The diesel engines installed there are specially adapted for CC applications, after years of research for raising the temperature of the exhaust gases. Research and experimental work in Wärtsilä for designing new generation diesels with high efficiency, low heat losses, high temperatures of the exhaust gas and consequently high combined cycle efficiency, has been fruitful [5.27], [5.12], [5.2]. The ICECC plant has 38 MW_{el} power output and achieves 53% electric efficiency. Its layout is shown on **Fig. 5.7**. Wärtsilä has also built ICECC units in other countries [5.22], [5.12].



Wasa Pilot Power Plant simplified process flow diagram

Fig. 5.7. *(from [5.2])*

Another manufacturer who is seriously taking the ICECC idea into consideration is MAN B&W. Their huge two-stroke diesel engine units with power ratings of up to 50 MW_{el} and simple-cycle thermal efficiency of up to 51% are suitable for ICECC arrangements. Such large units operate in ICECC mode with turbocompound and steam bottoming heat recovery. A good example is the Coloane 222 MW_{el} power station in Macao with six MAN B&W heavy fuel oil fired, low speed diesel units. Additional power produced by the turbocompound and steam turbine bottoming systems is 5.5% of the output of the main engines. Considerable amount of heat is also needed for conditioning the heavy fuel oil for the engines. [5.22]
ICECC idea have been put forward in technical articles, for example [5.15], [5.16].

The performance of ICECC can be easily upgraded by supplementary firing for raising the parameters of the bottoming cycle. The remaining oxygen in the ICE exhaust is enough for small amount of supplementary fuel fired in it, or otherwise additional air can be easily added to the exhaust flow. This "sacrifice" of fuel for supplementary firing (addition of fuel on a lower temperature level, bypassing the engine) can well justify itself, because it leads to substantial increase in the output of the BC [5.22]. A simplified example of such supplementary fired cycle is shown on **Fig. 5.8**.

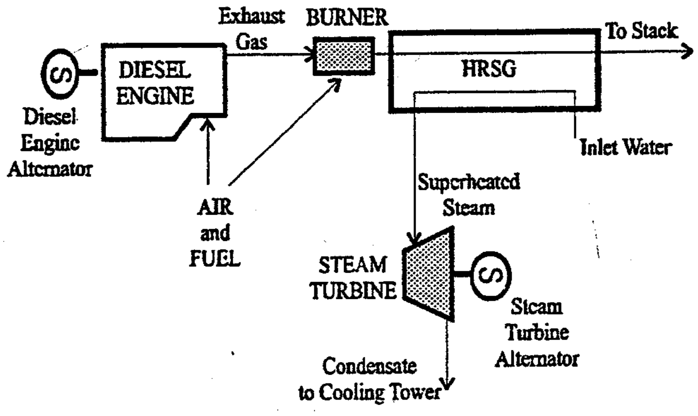


Fig. 5.8: *Simplified schematic of an ICE with supplementary firing in the HRSG to maximize BC power output. (from [5.10])*

In this respect, enhanced waste heat recovery in combination with turbocompounding is possible and has been suggested [5.28]. Exhaust from the main engine is supplementary fired with a small portion of the same fuel in a combustion chamber upstream of the turbocharger and power turbine. The additionally heated exhaust gases expand in the turbocharger turbine and in the separate power turbine. The turbocharger delivers more compressed air than is needed for the main engine aspiration. Excess air is fed directly to the supplementary fired combustion chamber to ensure stable combustion. The power turbine is larger and produces much more mechanical power than a standard turbocompound arrangement and consequently has better performance. Remaining heat in the exhaust gases after the turbines (with a high temperature thanks to the supplementary firing) is transferred to a steam Rankine bottoming cycle. With such configuration, supplementary firing of 13.5% of the main engine fuel consumption provides an increase in plant output of 16%. Incorporation of steam injected power turbine instead of a bottoming ST in the cycle provides a further increase of 8.7% in total plant output. [5.28]

On the other hand, many engines in industrial settings are connected to boilers, in which part or all of the engine exhaust is used as combustion air, together with fresh air. The boiler's role is not to support a bottoming cycle, but simply to produce large quantities of steam or hot water for industrial process needs. According to proponents, the temperature and O_2 content of diesel-engine exhaust are compatible with traditional boilers. Fresh air can be added just to optimize combustion and achieve stable flames [5.20], [5.1]. Engines with steam producing HRSG can be also parallel-connected to additional steam generators, achieving highest flexibility and reliability for process steam supply [5.18].
Sometimes the steam consumption could be much higher than the need for mechanical power from the engine and the main fuel input would go to the steam generator. Such arrangements with fully-fired engine exhaust and a large quantity of fuel and fresh air added to the boiler combustion chamber can show huge heat-to-power ratios in the order of 15:1. This is often the case in various industrial processes [5.3], [5.21]. A simplified example of such layout is presented on **Fig. 5.9**.

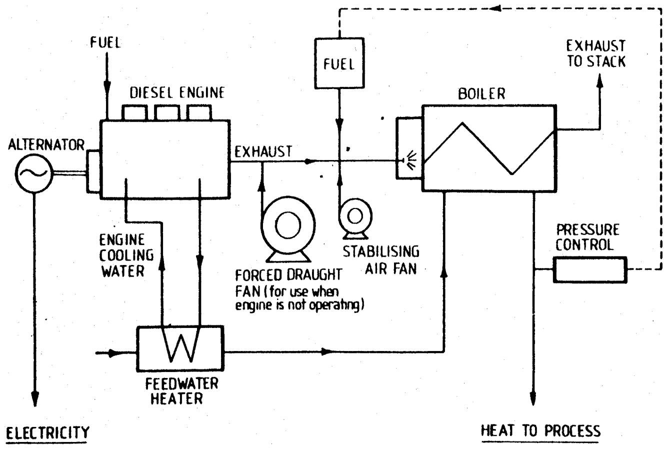
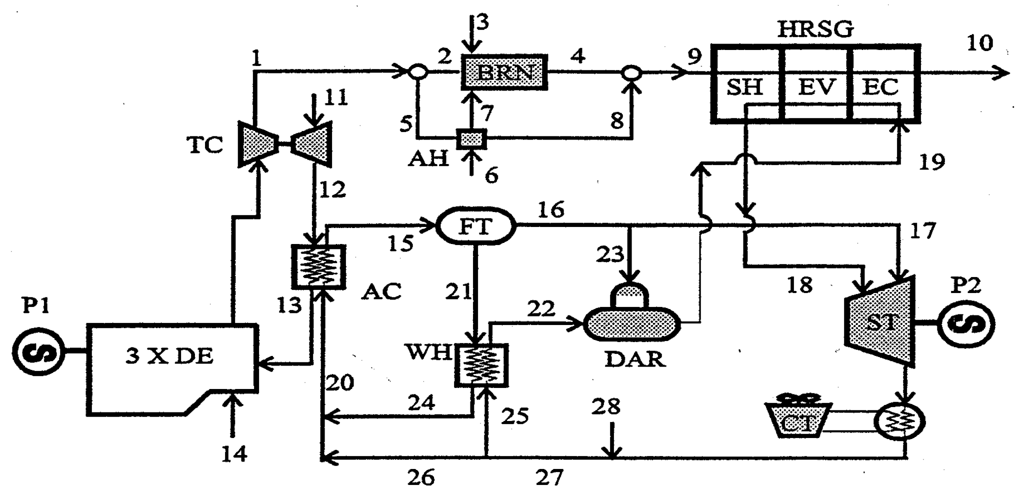


Fig. 5.9: *Simple diagram of an ICE with supplementary firing (boost-burning) in the HRSG to increase the heat production in the BC. (from [5.3])*

The great advantage of supplementary firing or fully-firing of ICE exhaust with any fuel or type of boiler for decreasing the amount of NOx by reburning is stressed in several works. Generally, 50%-70% NOx reduction in the ICE exhaust after reburning in the boiler combustion chamber of the BC is assumed possible [5.11], [5.15], [5.25], [5.10]. Special combustor designs have been developed for reburning ICE exhaust in a fully-fired CC scheme as that shown in **Fig.5.8** [5.10]. Such improved combustors can enhance NOx reduction through controlled reburning. A layout of the CC unit, incorporating these combustors and a flash steam producer is shown on **Fig. 5.10**. This system, called by the authors "Diesel Combined Technology Combined Cycle", does not achieve higher efficiencies than the unfired CC, marginally higher (<5%) than the simple cycle engine. Its true benefit is the decrease in NOx emission levels. [5.10]

A relevant study for NOx destruction in ICE exhaust, supported with experiments, has been performed by Wingård et al. at Chalmers Tekniska Högskola, Sweden [5.34]. They assess the possibility for NOx reduction by reburning when ICE exhaust is used as combustion air in a CFB boiler fired by mixed coal and biomass. The results have been extremely satisfactory. Chemical reduction of NOx to molecular nitrogen has been 75-90% (some catalytic effects may have taken place), the higher conversion rate appearing at higher NOx content in engine exhaust. The authors point out also the applicability of ICE for CHP and/or HCC configurations and their advantages to similar systems with gas turbines in the small-scale power range. [5.34]

Accordingly, hybrid ICECC using lower-quality solid fuels for supplementary firing (other than the TC fuel) can prove to be efficient and technically and economically feasible. They combine the high efficiency of the ICE with the possibility for utilization of a different (cheaper) fuel for the BC, while achieving good overall performance and still not losing the potential for CHP applications. [5.22]
A perfect example for cogeneration application of a diesel engine combined with a coal/RDF fired grate boiler (producing steam and hot water for district heating and industrial purposes), constructed in England (Fort Dunlop), has been presented in a case study from 1987 [5.24]. The layout is shown on **Fig. 5.11**. Part of the engine exhaust gas is used as combustion air in the boiler (together with fresh air), while the rest of the exhaust heats an economiser, parallel to the economiser of the boiler. Early experience from operation of this CHP station is reported. Performance in all operating modes has been satisfactory. The further option for using this HCC concept with a steam turbine is suggested. [5.24]



AH	air preheater	**1**	DEG from 3 engines	**16**	117°C steam
AC	engine intake air cooler	**2**	DEG ducted to burner	**17**	Excess flash steam
BRN	burner	**3**	HFO delivered to burner	**18**	Superheated steam
CT	cooling tower	**4**	combustion products	**19**	117°C feedwater
DAR	deaerator (179.3 kPa)	**5**	bypassed DEG	**20**	Water into air cooler (*)
DE	WD 18V46 engine	**6**	ambient burner air	**21**	117°C water
EC	economizer	**7**	232°C air to burner	**22**	85°C water
EV	evaporator	**8**	bypassed DEG after AH	**23**	117°C steam
FT	flash tank (179.3 kPa)	**9**	bulk gas to HRSG	**24**	Cooled flash tank water (*)
P1	DE alternators	**10**	stack gases	**25**	43°C sat. water to water heater
P2	ST generator	**11**	ambient intake air for 3 DE's	**26**	43°C water to air cooler
SH	superheater	**12**	heated intake air	**27**	43°C sat. total return
ST	steam turbine	**13**	cooled intake air (*)	**28**	43°C sat. makeup water
TC	DE turbocharger(s)	**14**	HFO delivered to 3 DE's	**(*)**	Enthalpy to be determined
WH	water heater	**15**	177°C water		

Fig. 5.10: *Diesel Combined Technology CC with NOx destruction through reburning in a specially designed burner for supplementary firing in the engine exhaust. (from [5.10])*

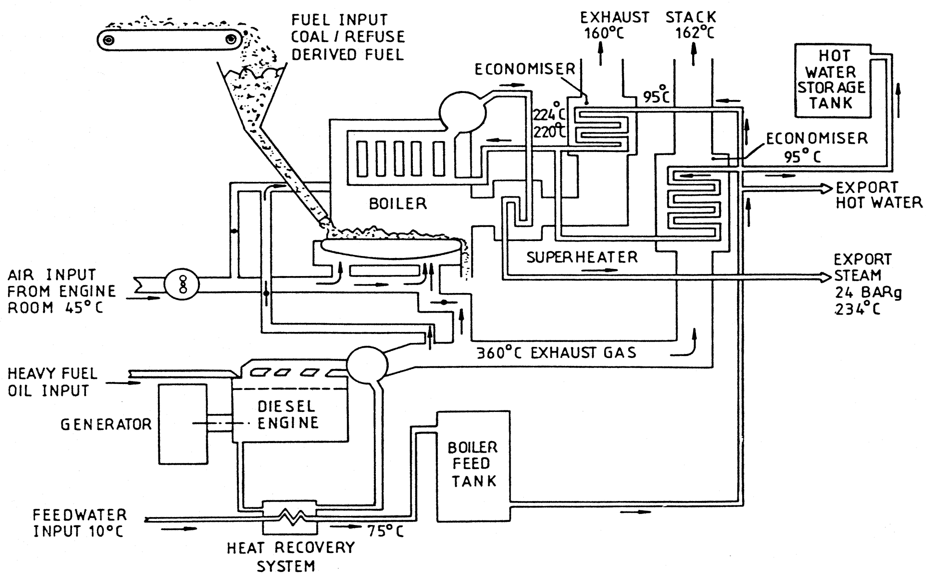


Fig. 5.11: *Flow schematic of a Hybrid CHP cycle comprising a diesel engine connected to steam generator fired by coal and RDF. No power production equipment in the bottoming cycle, only hot water and steam for district heating and industrial processes. (from [5.24])*

Just as with GT topping cycles, combination of ICE and solid-fuel-fired steam boiler in large or small scales can prove to be a viable option for utilization of the engine exhaust heat in a very efficient way. In addition, repowering options with internal combustion engines (large diesels or gas/diesel engines) instead of gas turbines are possible and have been proposed.
This possibility has been well recognised and put forward by some authors in their journal articles or conference papers. For example, [5.20] presents a general discussion on all options for repowering of old steam plants including also repowering with diesel engines. Potential advantages of ICE relative to repowering with gas turbines have been pointed out as follows: greater flexibility matching prime mover to existing steam cycle, less impact on performance from ambient temperature and air conditions (especially for warm climates), greater fuel flexibility, less arduous operation and maintenance and better capability for meeting radically changing thermal and electric loads. [5.20]

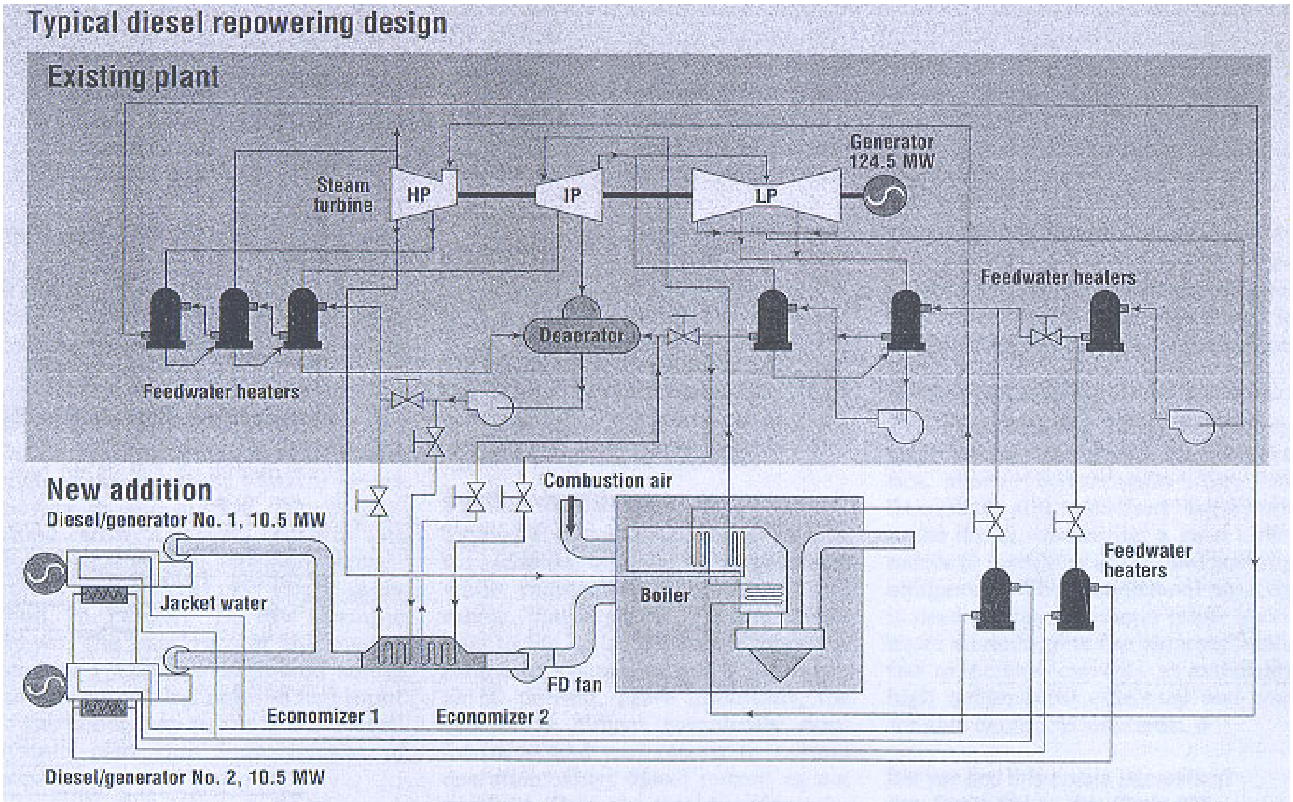


Fig. 5.12: *An old steam power plant repowered with 2 diesel engines. (from [5.26])*

Another perfect example is [5.11], where fully-fired combined cycles of diesel engines with gas or oil fired boilers and hybrid combined cycles of diesel engines with coal-fired boilers are presented. Efficiencies vary based on the ICE performance and the size and complexity of the Rankine cycle. In general, efficiencies are competitive with those of gas turbine hybrid combined cycles in less than 100 MW power range (40-45% LHV net electric efficiency). Cycle configurations with a total power output of 46.4 MW_{el} and 66.5 MW_{el} and fuel energy input ratio (diesel engines to boiler) of 0.7 and 1.17 respectively, have been calculated. The author underlines the good fuel flexibility of both engines and boilers, together with the unrivalled load flexibility and economic advantages of the diesel-hybrid system. Experimental test results from a small-scale diesel-coal hybrid cycle (built and operated as a test in 1994) are described. Reduction of NOx emissions from diesel exhaust after passing through the boiler’s burners (the boiler being fired with pulverized coal) have been successfully demonstrated. It has been concluded that the diesel-coal system definitely has a promising future. [5.11]

In a series of technical articles, Mack Shelor (from the North American representation of Wärtsilä NSD) promotes the idea for feedwater heating repowering of steam power units with gas/diesel engines [5.25], [5.26]. The cycle layout is shown on **Fig. 5.12**. The concept is viable in all terms. Investment costs are estimated to be far less than those for GT repowering.
Jacket water and exhaust gas of the engine are used to heat the steam cycle feedwater, replacing all LP and some of the HP preheaters. The engine exhaust is then admitted to the boiler as combustion air. Its small quantity and low temperature would not require refurbishment of boiler air ducts and would make the addition of engines totally non-intrusive to the boiler. Total system efficiency in this HCC mode may be improved by as much as 10%. The increase in ST power output due to closed steam extractions offsets the costs for interconnection of the engines to the steam system. If very low NOx emission levels are desired, selective catalytic reduction unit can be included in the system. [5.25], [5.26]

Wärtsilä has installed several power plants (also with gas engines) operating successfully with supplementary-fired exhaust gas boilers [5.30], [5.2].

Several MSc theses and student projects have been performed under Vattenfall AB in Sweden, evaluating different possibilities for the repowering of existing steam power stations with gas turbines or internal combustion engines.
Two of them, [5.5] and [5.8], are devoted to the repowering of Marviken power plant by topping with different types of Diesel engines. Marviken power station is an oil-fired steam Rankine type with very low ST inlet parameters and consequently very low thermal efficiency (29.4%). The station has been designed as a nuclear one, but the nuclear reactor has never been taken into operation, due to change of plans. Instead, a steam boiler fired by fuel oil has been installed, in order to find application for the already commissioned ST and auxiliary equipment. The plant is used only for peak-load shaving duties.
Careful thermodynamic calculations on the expected cycle performance and general economic assessments have been performed in these thesis works. The evaluated cycle arrangements are of the mixed fully-fired and feedwater heating parallel-powered type, as described above (see again Fig. 5.12). Heat from the engines (from both jacket water and exhaust gas) is first utilized for BC feedwater preheating, then engine exhaust is fed to the steam generator as combustion air, together with fresh air. The general conclusions have been that repowering with large Diesel engines is a viable alternative, yet providing less efficiency gain and higher economic burden than the GT repowering alternatives. [5.5], [5.8]

Another thesis work [5.7] has been devoted to similar evaluation of the option to connect into a combined cycle two large diesel engines with a steam power plant. The engines and the steam plant have already been installed, situated close to each-other and operating separately for some time. Interconnections between them for creating a CC could have been designed and built comparatively easy. The evaluated CC layout is again similar to the ones above. Engine jacket water and exhaust gas are used for boiler feedwater preheating in three preheaters. Exhaust gas is then fed to the boiler. Different alternatives on this base cycle layout have been evaluated, with varying ratio of fresh air to exhaust gas in the boiler and variation of engine exhaust bypassing the boiler, due to boiler restrictions. The results from the calculations have been quite positive, indicating that the CC would provide a 3.3 %-point increase in efficiency compared to the observed average performance of the separate engines and steam cycle, from 34.3% to 37.6%. Total electric power output from the CC, compared to the sum of the separate units' power could have been raised with 2.2%, from 50.4 MW_{el} to 51.5 MW_{el}. NOx emissions would decrease. [5.7]

One very interesting MSc thesis work has been performed under Sydkraft Konsult AB in Sweden in 1998 [5.13]. The main topic has been a careful modelling, calculation and investigation of a HCC comprising ICE and biomass-fired boiler. Three different hybrid configurations are modelled in CHP mode: A parallel-powered cycle where part of the steam is generated by the ICE exhaust while the final superheating is in the boiler, a parallel-powered cycle where the ICE exhaust is used for feedwater preheating and a fully-fired cycle. Three different engines with different sizes, efficiencies and exhaust gas temperatures are used in all configurations. In all simulations, the low-temperature heat rejection streams from the engine are used for district heating, together with heat from the backpressure condenser of the ST. Every configuration is extensively simulated at varying natural gas to biomass fuel input ratios. Careful economic calculations with sensitivity analysis of costs are also presented. An attempt is made to evaluate the thermodynamic advantages of the modelled hybrid cycles by comparison to the average performance of two separate simple cycle units utilizing the two fuels at the given ratio. The performance of the ICE in simple cycle mode is used for this comparison, while the pure ICECC arrangement is also presented but recognized as inappropriate as basis for comparison. The effect of scales (larger or smaller sizes) on the performance and steam parameters of the steam turbine is also recognized and inserted in the procedures for evaluation of thermodynamic advantages. [5.13]

As a conclusion, it must be pointed out that the ICE is a suitable topping cycle engine for combinations with bottoming cycles. Such power cycle arrangements will be modelled, calculated, analysed and presented in the major research project following this Literature Study.

6. STATE-OF-ART OF HYBRID COMBINED CYCLES WITH BIOFUEL-FIRED BOTTOMING CYCLE IN SWEDEN AND ITS NEIGHBOURING COUNTRIES

6.1. Finland

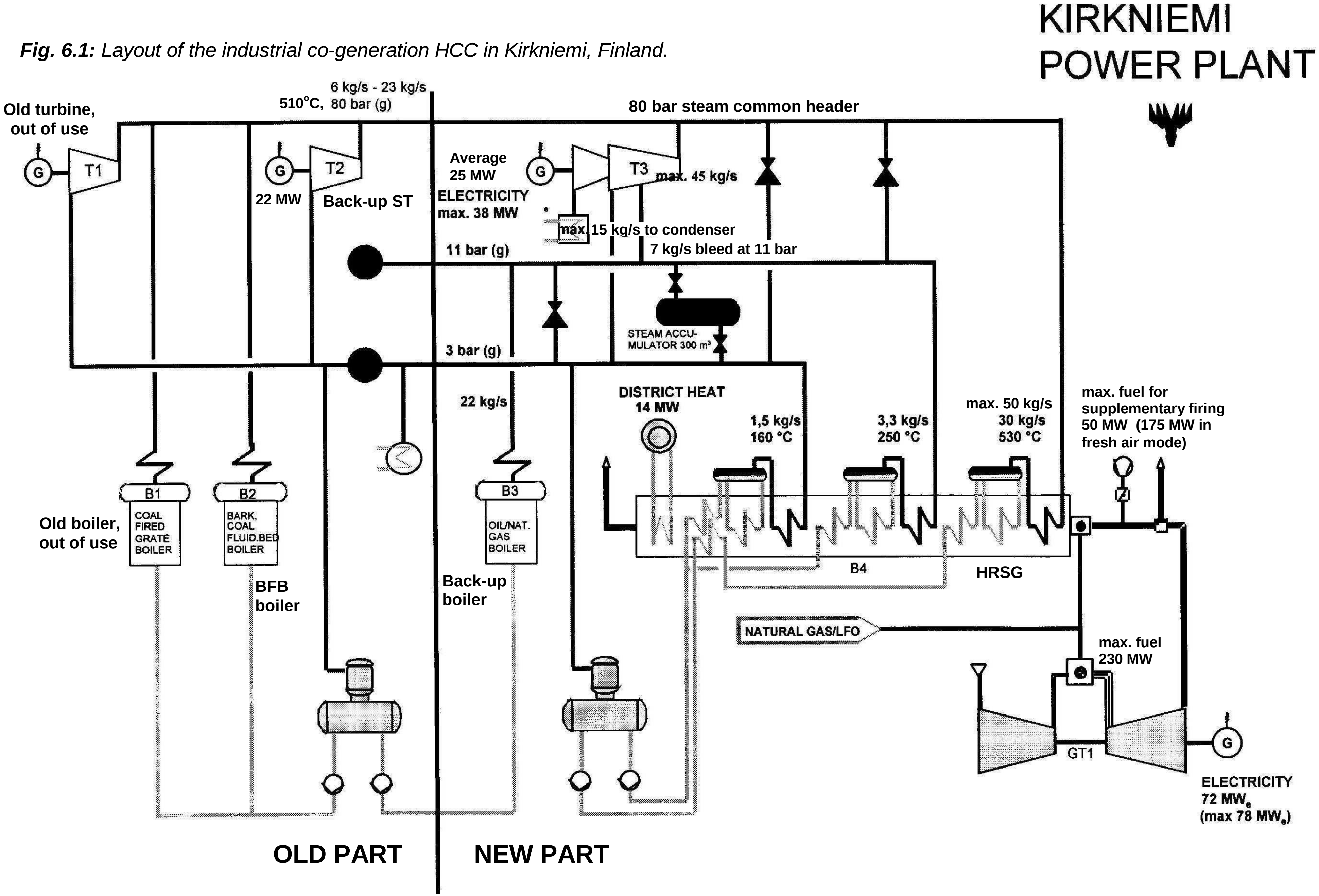
There are two outstanding power units of relevant configuration in Finland. Both are industrial co-generation installations for the pulp & paper industry, and both feature a parallel-powered configuration of a NG-fired gas turbine and a biofuel-fired steam generator, working independently and supplying steam to a common steam turbine and process users. The connection between topping and bottoming cycle is only at the high-pressure steam header towards the steam turbine.

6.1.1. Kirkniemi Industrial Cogeneration Power Plant

The Kirkniemi Power Plant is located near the town of Lohja, south-central Finland, about 65 km west of Helsinki. The plant serves the local paper mill owned by Metsä-Serla Corporation, recently renamed M-Real. The power plant itself is owned by Imatran Voima Oy, recently renamed Fortum, and is operated by the paper mill's personnel. Decision for building the combined cycle power plant was taken in 1995, in connection with Metsä-Serla's plans to invest in a new machine for high-quality paper at the Kirkniemi mill. The enlargement of production capacity required additional supply of process steam and electricity, which led to considerations for finding a reliable and economically competitive option for investing also in a new heat and power unit. The design and construction of the new power plant was outsourced to Imatran Voima Oy and a range of subcontractors.
Being an industrial co-generation unit, the major purpose of the combined cycle plant is to provide steam according to the process needs. Its design and operation are governed by the process-steam demand.

Fig. 6.1 shows the cycle layout with its main parameters. On the left side of the figure are the old components, on the right side are the new ones.
The combined cycle comprises one General Electric Frame 6FA gas turbine with a three pressure-level HRSG (B4 on Fig.6.1.) and a circulating fluidized bed steam boiler (B2). Natural gas is the fuel for the gas turbine, with LFO as back-up fuel. The GT has dry low-NOx combustor and operates with more than 34% electric efficiency in simple cycle mode, with 70MW power output at ISO conditions. In pure unfired combined cycle configuration, net efficiency of 53% can be achieved. Burners for supplementary firing are installed in the flue gas duct before the HRSG. Steam generation with fresh air supply in the HRSG (when the GT is out of operation) is also possible, two fresh air fans are installed.
Steam is generated in the HRSG at 80 bar, 11 bar and 3 bar pressure. The HRSG is horizontal, with natural circulation. A district heating economiser utilizes the rest of the heat in the GT exhaust (cooling it down below 100^{o}C) and delivers 14 to 15 MW heat. The main steam production is at 80 bar, 530^{o}C, which is mixed with steam of similar parameters from boiler B2 and goes directly to steam turbine T3. At full load without supplementary firing in the HRSG, 30 kg/s steam can be generated at 80 bar.

Fig. 6.1: *Layout of the industrial co-generation HCC in Kirkniemi, Finland.*



With supplementary firing, steam production can be increased to 50 kg/s. With fresh air operation of the HRSG, steam production is 45 kg/s. The steam expands in steam turbine T3 to 11 and 3 bar, where it is mixed with generated steam at the respective pressures in the HRSG and fed to the industrial process users. A small amount of steam is condensed after the turbine. The cooling water after the condenser is used directly in the industrial process. Direct pressure reduction valves are available for supplying steam to the process users if T3 is out of operation.
Commercial operation of the new part of the power plant (GT, HRSG and T3) has started in November 1997.
Boiler B3 is a simple generator of slightly superheated steam at 11 bar pressure, fired with natural gas or oil. It has been installed to supply steam to the process users while the combined cycle has been still in construction. Nowadays B3 is kept as a back-up boiler and is constantly in warm stand-by.
A 300 m^3 steam accumulator has been installed in order to help equalizing the much variable steam demand for the industrial process.

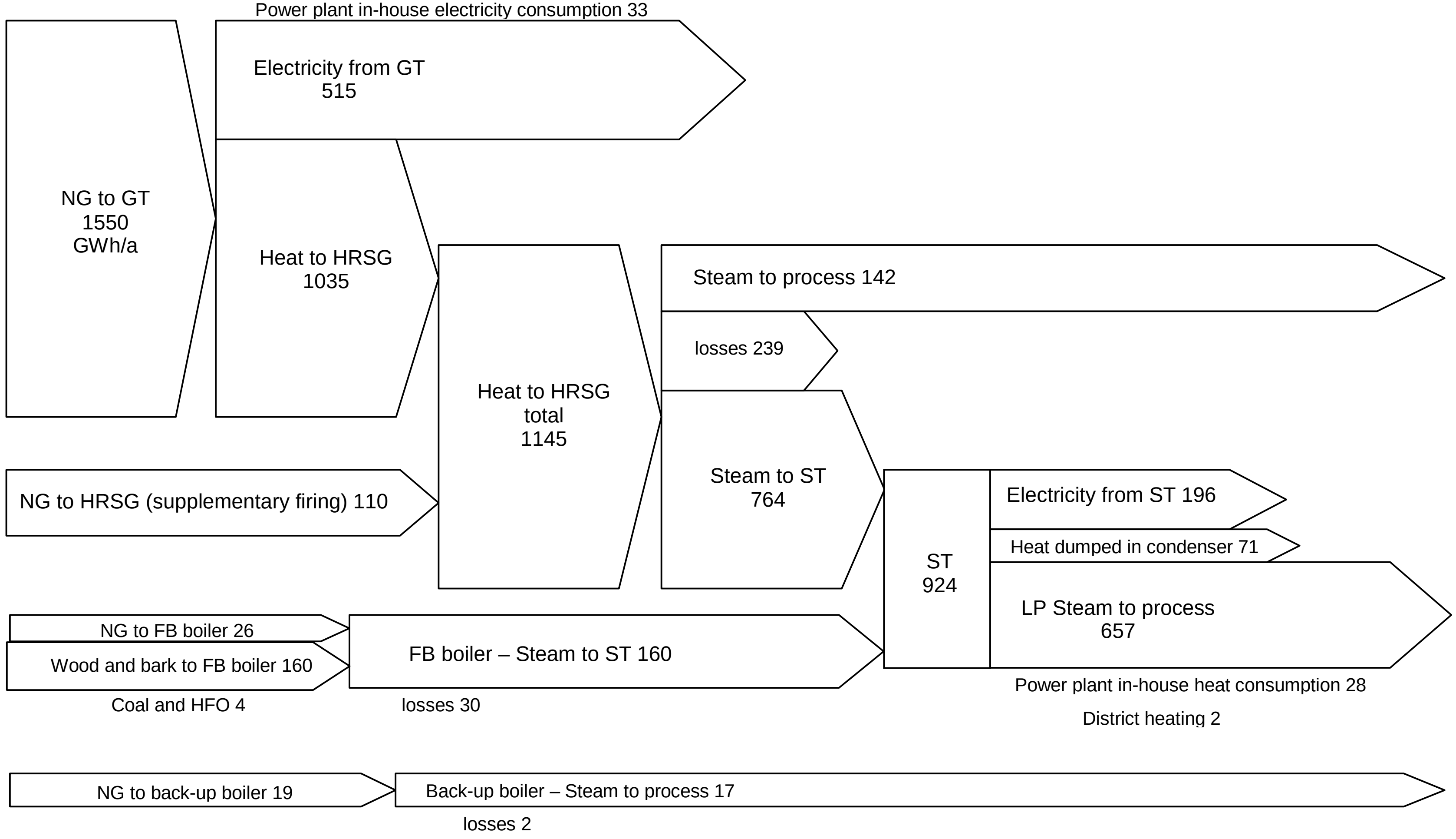
The fluidized bed boiler B2 has been installed in 1971 as an oil-fired steam generator, together with the steam turbine T2. The boiler has been converted to bubbling fluidized bed in 1985, burning wood waste and bark from the mill (residues from mechanical pulp production), sludge (dried to 45% moisture content) and some other bio-wastes from the area. Biomass fuel is usually supplied to B2 at a rate of 3-4 kg/s. Oil and coal serve as support fuel. In 1997 natural gas has become available at the site, so the support burners of B2 have been converted to natural gas. Since then, use of oil and coal is close to zero, but HFO is still a back-up fuel. B2 can be considered as the bottoming boiler of the overall combined cycle. It is operated always in base load, burning as much fuel as is available at the moment. It supplies steam to the common 80 bar network, with temperature of 510^{o}C. Steam production varies according to fuel availability, usually around 7 kg/s on biofuel only. Maximum 24 kg/s steam can be generated with the help of the support burners at full load.
Steam turbine T2 is out of operation, but is kept as a back-up. Boiler B1 and steam turbine T1 comprise the oldest power unit installed at the site, in 1966. They are both out of operation now.
The power plant supplies all process-steam for the paper machines and covers about 80% of the mill’s electricity requirements. The total efficiency is more than 80%. Electricity-to-heat ratio (α-value) at full load is 1.05 (0.91 at the most often used loading). An year-average Sankey-diagram is shown on **Fig. 6.2**.

The combined cycle in Kirkniemi has provoked much attention, because it is the largest industrial heat & power unit of its kind in Finland and employs the first GE Frame 6FA gas turbine installed in Europe.

Information about Kirkniemi power plant was acquired from:
1. Anonymous (1998), “Frame 6FA Cogeneration Commissioned at Kirkniemi”, Modern Power Systems, Vol.18, No.2, February 1998, pp 43-51.

And through personal communication with:
1. Kalevi Merinen, general power-plant manager (M-Real Kirkniemi, FIN-08800 Kirkniemi, Finland)
2. Marko Heiskanen, operator.
Their contribution is gratefully acknowledged.

Fig. 6.2: *Energy Sankey-diagram for M-Real pulp & paper mills' power plant in Kirkniemi, Finland. Annual energy flows in GWh/a.*



6.1.2. The power plant for the pulp & paper mill in Kotka

The city of Kotka is located in the most south-eastern corner of Finland. One of the largest industrial establishments there is the StoraEnso pulp & paper mill, part of which is the combined cycle industrial cogeneration power plant described below.
The power plant in Kotka is quite similar to the one in Kirkniemi. A gas turbine with a HRSG produces steam, which is simply mixed with steam from a chemical recovery boiler and is fed to a common steam turbine. The cycle provides steam for the industrial process in the mill and feeds electricity to the local and national grids. The combined cycle power plant was put into operation in 1993.

The general layout of the combined cycle is shown on **Fig. 6.3**. The left side of the figure shows the new components of the combined cycle, the right side shows the old ones. The gas turbine is a heavy-duty Frame 6 machine, supplied by European Gas Turbines. It is equipped with dry low-NOx combustion chamber and is fired with natural gas. The power output in simple cycle mode at $0^{o}C$ is 41.8 MW. The HRSG is a horizontal one with natural circulation. Steam is generated at two pressure levels – 80 bar and 6 bar. Supplementary firing burners are installed in the GT exhaust duct before the HRSG. Operation with fresh air is also possible, a fresh air fan is installed.
The 80 bar steam is produced at a rate of 17.8 kg/s at full GT load without supplementary firing, 33.4 kg/s with maximum supplementary firing and 30.8 kg/s at fresh air operation. The steam is superheated to $500^{o}C$ and is fed into the common 80 bar steam header. The 6 bar steam is superheated to $195^{o}C$ and is produced at a rate of 4.2 kg/s without supplementary firing, 2.6 kg/s with maximum supplementary firing and 2.9 kg/s at fresh air operation. The 6 bar steam is fed directly to the industrial users. The last heat-exchange surface in the HRSG is a district heating economiser, providing around 10 MW of heat.

The bottoming cycle comprises a chemical recovery boiler, burning black liquor from the chemical pulping process and generating steam at 80 bar, $480^{o}C$, 30 kg/s. Steam from the boiler is mixed in the common 80 bar header with the steam from the HRSG and is fed to the steam turbine. The steam turbine is a reaction type, with maximum power output of 30.3 MW. Steam is expanded to the two major pressure levels of the industrial users – extraction at 12 bar (serving the pulp digesters) and backpressure at 6 bar (mixed with the LP steam from the HRSG and feeding the paper machines). Pressure reduction valves from 80 bar to 12 and 6 bar are available as a back-up. Additional pressure reduction valve from 6 to 3 bar provides steam for other users and heating purposes.
An year-average Sankey-diagram for the overall combined cycle is shown on **Fig. 6.4**.

Information about the industrial power plant in Kotka was acquired through personal communication with:
Eero Ristola, general power-plant manager (StoraEnso Laminating Papers, P.O.Box 62-63, FIN-48101 Kotka, Finland).
His contribution is gratefully acknowledged.

Special gratitude is due to Antto Kulla from Helsinki University of Technology, Espoo, Finland, for perfectly organising and leading the visits to the power plants in Kirkniemi and Kotka. The financial help from Svenskt Gastekniskt Center AB, Malmö, Sweden, is also acknowledged.

Fig. 6.3: *Layout of the industrial co-generation HCC in Kotka, Finland.*

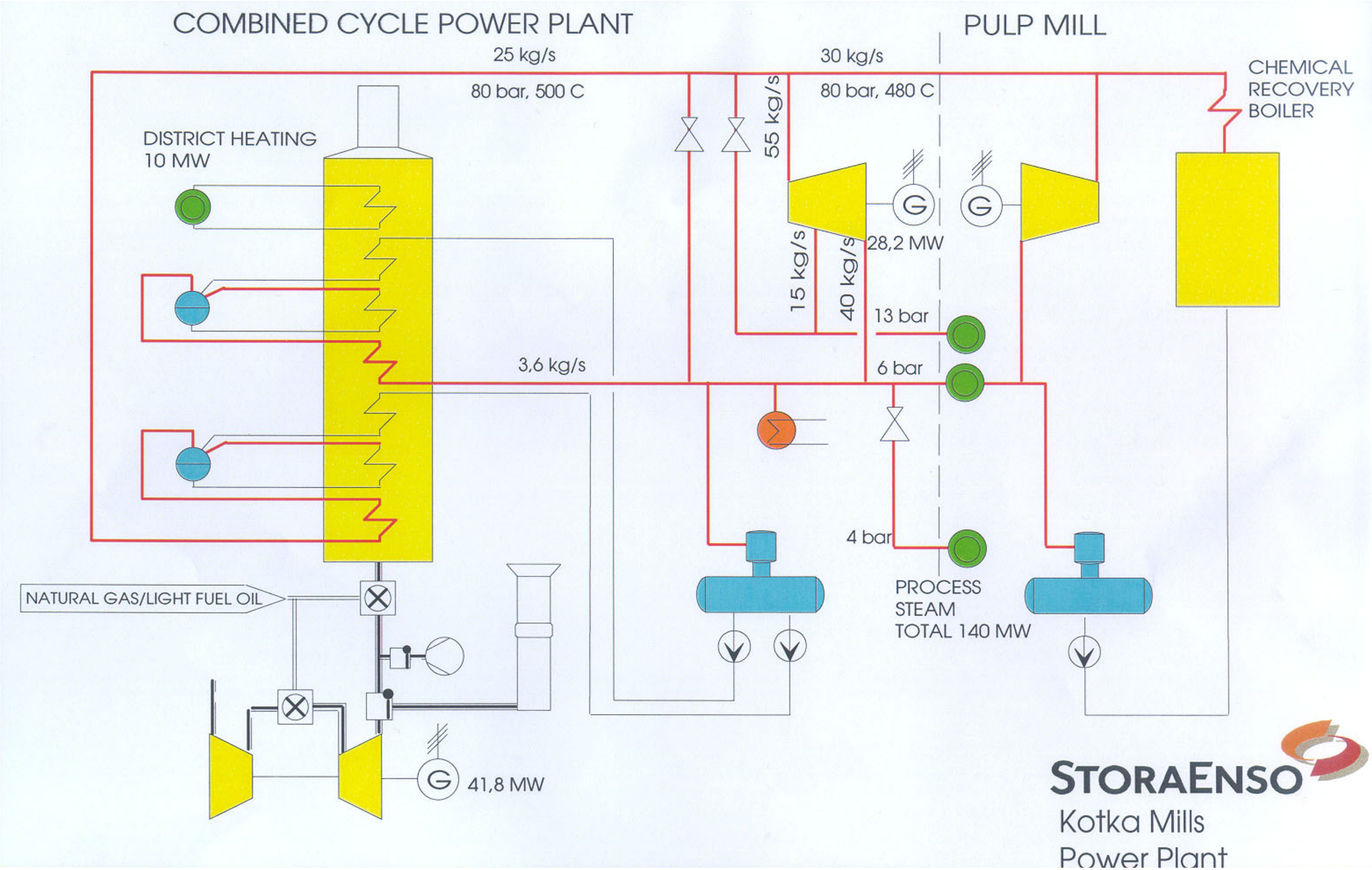
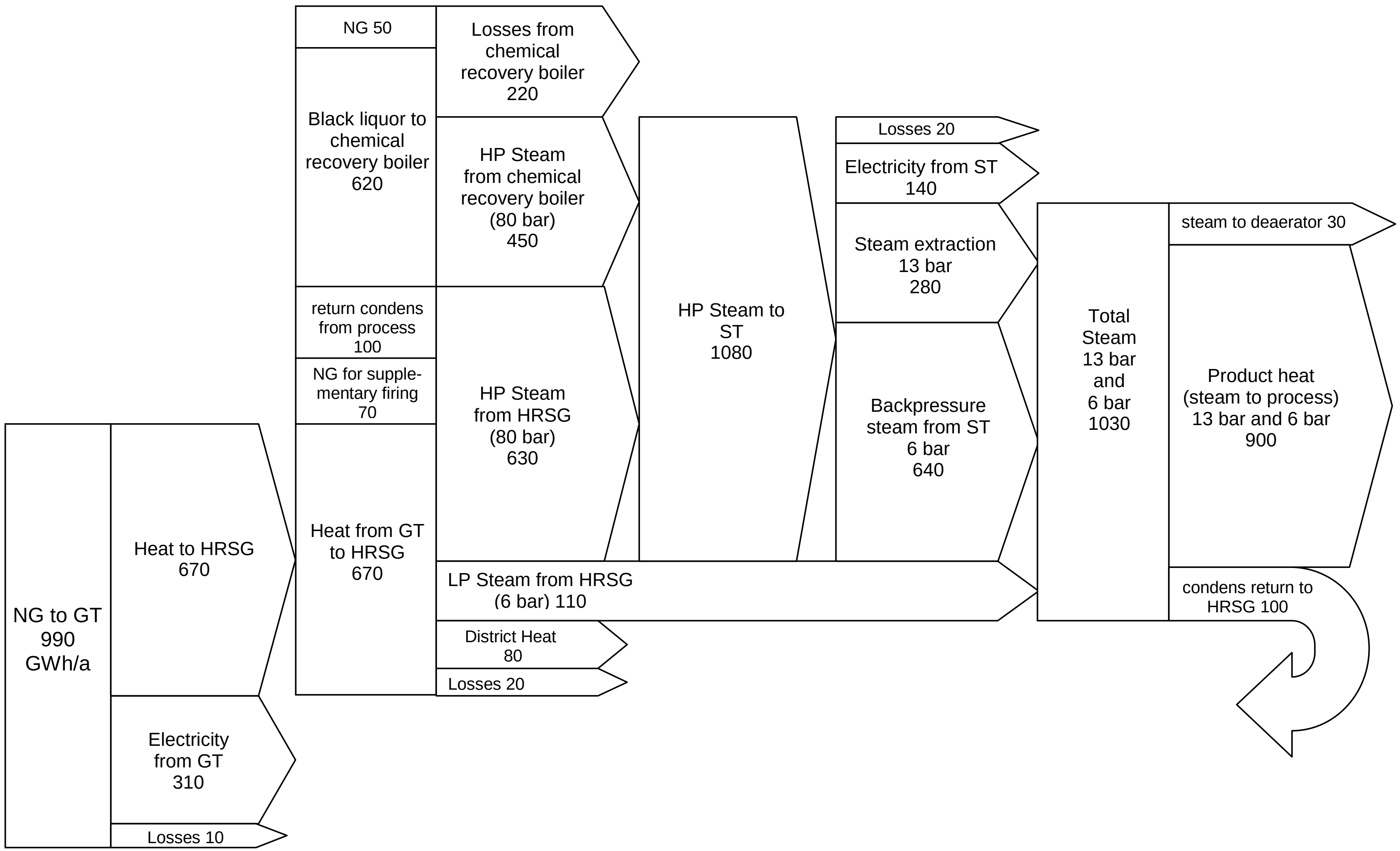


Fig. 6.4: *Energy Sankey-diagram for StoraEnso pulp & paper mills' power plant in Kotka, Finland. Annual energy flows at the most often used load regime, in GWh/a.*



6.2. Sweden

6.2.1. The CHP plant in Sandviken

In 1987 the management of the district heating plant in Sandviken (owned and operated by Sandviken Energi AB) started to consider possibilities for upgrading the plant into a CHP unit in order to produce electricity to cover most of the district heating system's own needs. In 1988, during the investigations for the project, it has been concluded that conversion of the existing peat-fired boilers into generators of superheated steam was not feasible. Instead, an idea was conceived for installing a gas turbine (fired with liquefied petroleum gas), whose exhaust would be used for superheating of saturated steam generated in the existing boiler. The steam would be expanded in a back-pressure steam turbine, where the condenser is part of the district heating system. The pre-study for the project was finalised with an exact proposal for the system configuration and economic calculations. During the following years, possibilities were sought to find financial aid for the project.

In 1991, the project was successfully completed and the resulting hybrid combined cycle has been taken into commercial operation. A layout of its specific parallel-powered configuration is presented on **Fig. 6.5**.
The gas turbine is a Centaur H model and runs on liquefied petroleum gas, with LFO as a back-up fuel. A mixture of peat and biomass (wood residues and others) has been introduced as fuel in the two boilers after the reconstruction. Steam parameters are quite low (governed by the lowest-cost option), pressure of 20 bar in the boiler and superheat to 470^{o}C by the GT exhaust before entering the ST.

Total electricity output is 9.4 MW, with annual production of maximum 40'000 MWh$_{el}$. The electric efficiency attributed to the GT fuel only, reaches 60%. The electric efficiency calculated on total fuel input in the hybrid cycle is 18%, while the main output is in the form of hot water for district heating. Specific investments per kW installed electric power have been slightly less than 5000 SEK/kW$_{el}$ (1991).

During the recent years, after the deregulation of the electricity market in Sweden in 1997, operation of the GT proved to be too expensive. The GT has gradually been taken out of service (kept only as a back-up power), while the constantly increasing heat load required installation of new hot-water boilers and a larger reconstruction of the two steam generators from the hybrid cycle. Until the middle of 2002, the two steam generators will be refurbished with superheating tubes and their power will be increased to 2 x 20 MW$_{th}$. The steam will be expanded in the steam turbine, producing power just enough to cover the in-house load of the plant and district heating system.

Information about the power plant in Sandviken was acquired from:

1. Winlöf, Tord, "Kort historik om projekt Elproduktion i HVC Björksätra", a brochure issued by Sandviken Energi AB, Sandviken, Sweden, August 1989.
2. "Ny elproduktionsteknik. Ombyggnad av hetvatten pannor för produktion av ånga och el", a brochure issued by Sandviken Energi AB, 1991.

The brochures were provided by Göran Panth, heat & power production manager (Sandviken Energi AB, 811 40 Sandviken, Sweden).
His contribution is gratefully acknowledged.

The parameters of the hybrid CHP unit in Sandviken are described and discussed by Westermark in 1992 [4.7].
A thorough description of the plant can be found in Kassem and Harvey's report [4.3], as well as in Egard et al. [4.2].
Information about the present situation and future plans of the plant can also be found on the company's web page: www.sandvikenenergi.se.

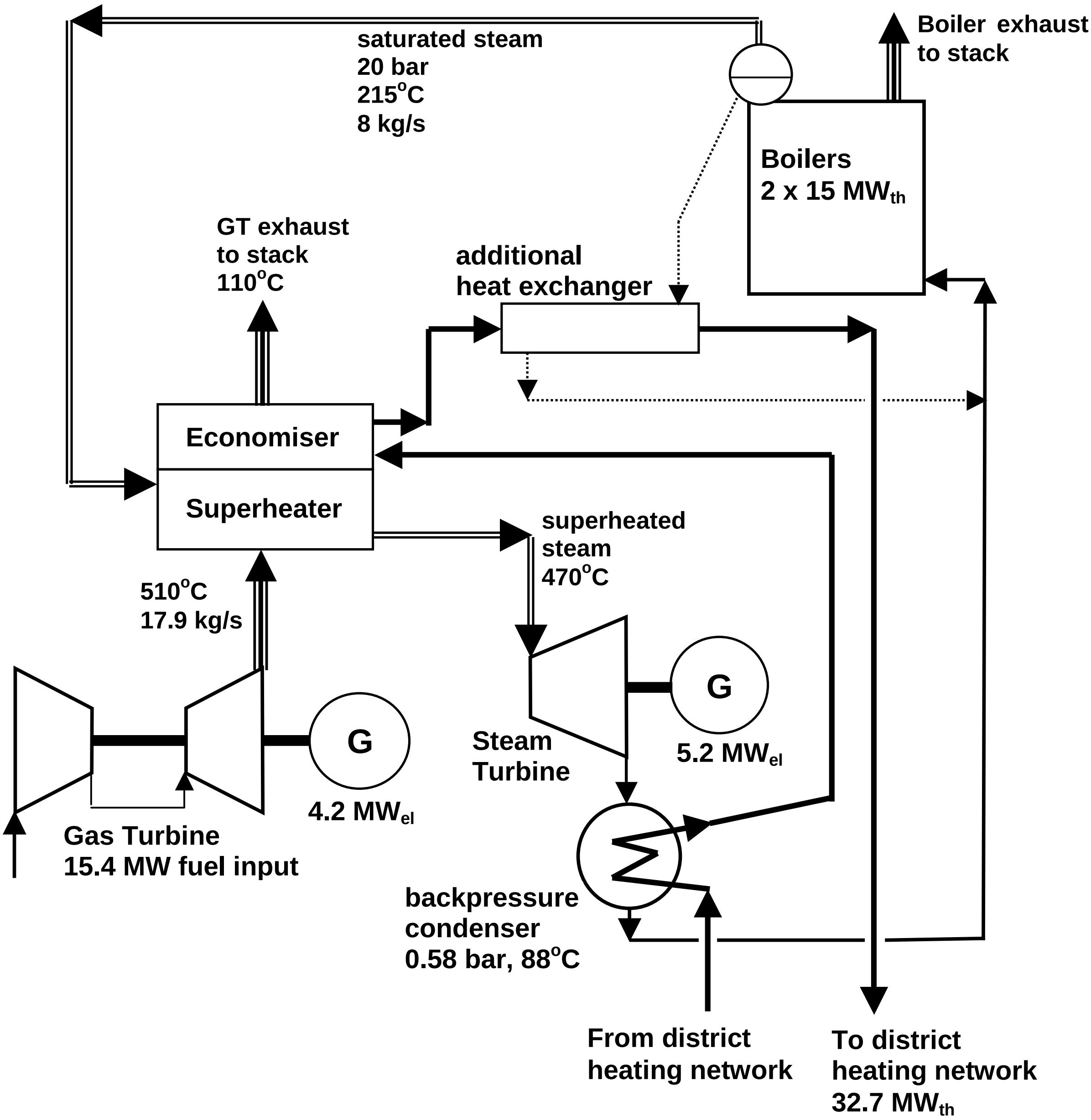


Fig. 6.5: *Configuration schematic of the HCC CHP unit in Sandviken, Sweden.*

6.2.2. The old CHP unit in Eskilstuna

This is the only example of a cycle resembling a fully-fired hybrid combined cycle ever put into operation in Scandinavian countries. The old CHP unit in Eskilstuna entered commercial service at the end of 1991. The cycle employs supplementary firing of part of the exhaust gases from a gas turbine with wood residues in a CFB boiler. The plant configuration and operational experience acquired from its operation are unique.

In the end of the 1980's, the owner and operator of the district heating network in Eskilstuna, Tekniska Verken (presently Eskilstuna Energi & Miljö AB), has decided to improve the security of district heat supplies through self-sufficiency with electricity. Until then, the system has had no electricity generation units. It has been totally dependent on power from the common electricity network in order for the boilers and circulation pumps to operate. The available emergency units (diesel generators) had not been able to cover the power demand in times of possible electricity shortages. One of the options for solving this task and improving the system's reliability had been the acquisition of new larger diesel generators as emergency power supply units. Another option had apparently been the installation of stationary electricity generation equipment and converting the system into CHP.
Among the several possibilities for converting one of the boilers into a co-generation unit, the most rewarding and cost-effective one was found to be the purchasing of a small gas turbine and thermally connecting it to the district heating network and the boiler. The option for transforming the boiler into a steam generator and installing a steam turbine has been abandoned.

The selected gas turbine is a Solar Centaur H model, supplied by ABB Stal AB (presently Alstom Power Sweden AB) through sub-contractors. The GT is of a single-axle design, packaged into a container as a complete unit with generator and air filter. The electricity output at 0^oC ambient air temperature is 4.3 MW, electric efficiency 27%, pressure ratio 8 and exhaust temperature 510^oC. Liquefied petroleum gas has been selected as the main fuel for the GT (with diesel oil as a back-up and start-up fuel). Natural gas was believed to become available at the site after several years, so at the time of project preparation natural gas has been viewed as the prospective fuel for the GT in the long run.

The boiler to which the GT had to be thermally connected has been the largest and most modern boiler in Eskilstuna at that time. It is a 50 MW_{th} CFB unit, generating saturated low-pressure steam and hot water entirely for the district heating network. It has been supplied by Kvaerner Generator AB and has started operation in 1986. Forest residues and sawdust are the fuels for the CFB boiler. Flue gas condensation unit has been installed behind it, recovering 7 MW heat for the district heating network. Another 8 MW low-temperature heat from the flue gas condensation is upgraded by heat-pumps and delivered to the district heating network in very cold days.

The first plans for coupling the GT to the CFB boiler have featured a fully-fired cycle configuration, where all exhaust from the GT is directly fed to the boiler and used as combustion air. This however has proved to be unfeasible, the boiler could not accept all exhaust gases from the GT. A costly and complicated restructuring of the boiler had been necessary, which would also lead to high dependence of the boiler on the GT with consequent sacrifice of reliability. Furthermore, the GT itself could not handle the high backpressure caused by the fluidized bed boiler (high axial forces occur, which would cause malfunctions). The fully-fired configuration needed to be revised.

After further considerations, the arrangement of the components has been altered. Total independence of the boiler and GT, minimum risk and minimum cost have been sought. Only half of the exhaust from the GT (52% of the total flow) would be supplied as combustion air to the CFB boiler, after cooling down to 100-110^{o}C in a heat recovery unit (district heating heat exchanger). The heat recovery unit has been supplied by Kvaerner Generator AB. The combustion air FD fan of the CFB boiler has been kept in operation, supplying a mixture of fresh air and half of the exhaust from the GT (the temperature of the mixture being 70^{o}C - the maximum the fan can handle in terms of temperature and volume). GT exhaust would not be used as combustion air in the boiler when the GT is fired with diesel fuel, to alleviate any risks of sulphur corrosion in the boiler and FD fan. The flue gas recirculation system of the boiler has been adapted to make possible further amount of GT exhaust to be fed to the boiler, instead of recirculating flue gases from before the stack. The rest of the GT exhaust flow would be released to the stack without passing through the flue gas condenser (higher O_2 content would lower the flue gas condensation temperatures and the recovered energy would be in general quite low).
The final cycle configuration is shown on **Fig. 6.6**.

The hybrid cycle in Eskilstuna has started commercial operation in the end of 1991. Specific investments have amounted to around 7250 SEK (1991) per kW_{el} installed power. Total efficiency of energy utilization of the GT fuel was calculated to be 84% at independent GT operation with only heat recovery for district heating, 94% in hybrid combined cycle configuration (half exhaust flow from GT to boiler) and 98% with flue gas condensation.
Measurements of the NOx emissions from the combined unit have been taken in April-May 1992. The results have proved the possibility to destroy NOx from GT exhaust in the CFB boiler, which is one important advantage of the fully-fired arrangement.

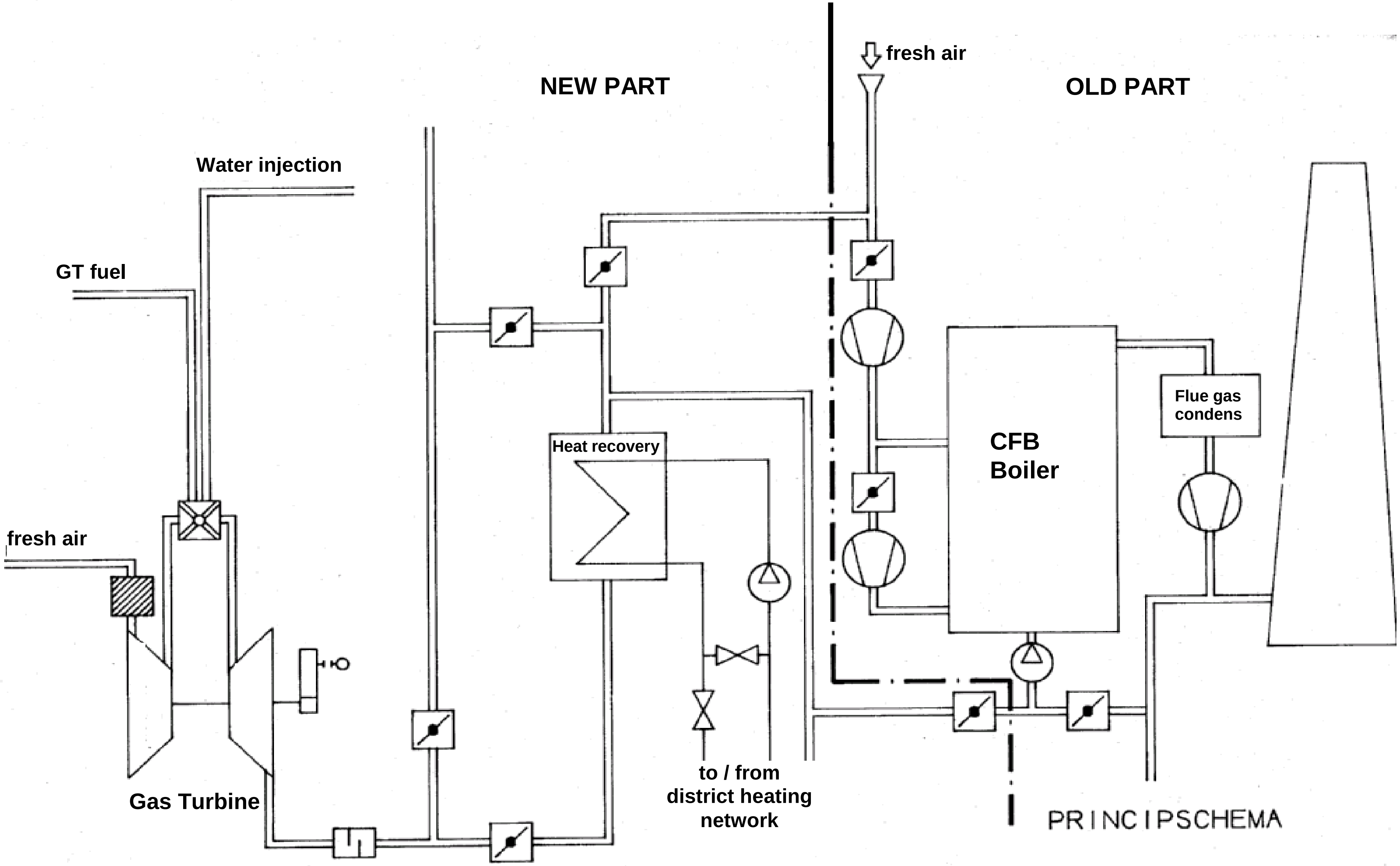
Low emissions of NOx have been one of the main goals during the development of the whole project. The GT has been equipped with water injection in the combustion chamber (940 l/h water consumption), which decreases NOx formation from 12.5 kg/h to 4 kg/h (at power output slightly less than full power).

NOx emissions from the CFB boiler alone have been 15 kg/h at full power. The total emissions from the GT and boiler, working independently, equal 19 kg/h.

NOx emissions from the CFB boiler, when half of the GT exhaust is supplied as combustion air, have remained at the level of 15 kg/h. This leads to the conclusion that all NOx content of the GT exhaust fed to the boiler has been destroyed in the boiler. Total emissions of NOx from the hybrid combined cycle configuration (only around half of the GT exhaust is fed to the boiler) are therefore 17 kg/h. The emissions from the GT itself have been decreased two-fold.

The decrease in total NOx formation achieved by the hybrid configuration, compared to separate operation of the GT and CFB boiler, is close to 10% (or 50% for the GT exhaust itself).

Fig. 6.6: *Schematic layout of the old CHP HCC unit in Eskilstuna, Sweden.*



The gas turbine has been operating for several thousand hours annually at full load, during several years after its installation. Natural gas is still not available at the site and there is no perspective for a NG pipeline to reach the site in the near term. After the deregulation of the electricity market in Sweden in 1997, electricity prices have fallen sharply and operation of the GT on any fuel other than NG has proved to be too expensive. Nowadays, the GT is not operated (the old CFB boiler is still in operation as a supplement to a very new and modern wood-chips-fired steam cycle CHP unit at the same site). The gas turbine is kept for emergency power supply.

Information about the old CHP plant in Eskilstuna was acquired from:

1. Björklund, Anders; Bohman, Sam, “CFB-panna med biobränsle som avgaspanna och NOx-reduktion för gasturbin”, Final Report, Tekniska Verken (Värme), Eskilstuna, Sweden, August 1992.

And through personal communication with:
Anders Björklund, heat & power plant manager (Eskilstuna Energi & Miljö AB, Värme, 631 86 Eskilstuna, Sweden).
His contribution is gratefully acknowledged.

The hybrid unit in Eskilstuna is described also by Egard et al. [4.2].

6.2.3. The Gärstad MSW-incineration CHP plant in Linköping

A perfect example of a MSW-incinerating hybrid combined cycle co-generation plant is the Gärstad Avfallsanläggning in Linköping, Sweden.

The MSW incinerators in Linköping have been first installed in 1981-83 in order to provide increase in district heating supplies while replacing expensive fossil fuels and alleviating the landfill problems for municipal wastes. A total of three incinerators have been installed, with a total heat output of 73 MW_{th} initially in the form of hot water. In the beginning of the 1990's, new plans started to emerge for upgrading the district heating plant into a CHP unit, which meant converting the incinerators into steam generators. The optimal solution was sought, and eventually a hybrid combined cycle configuration of MSW-based steam generation with superheating by gas turbine exhaust gases was selected as the most promising.

The conversion of the plant was done in 1994 and in early 1995 the new hybrid co-generation combined cycle entered commercial operation. The overall cycle layout with its basic parameters is shown on **Fig. 6.7**.

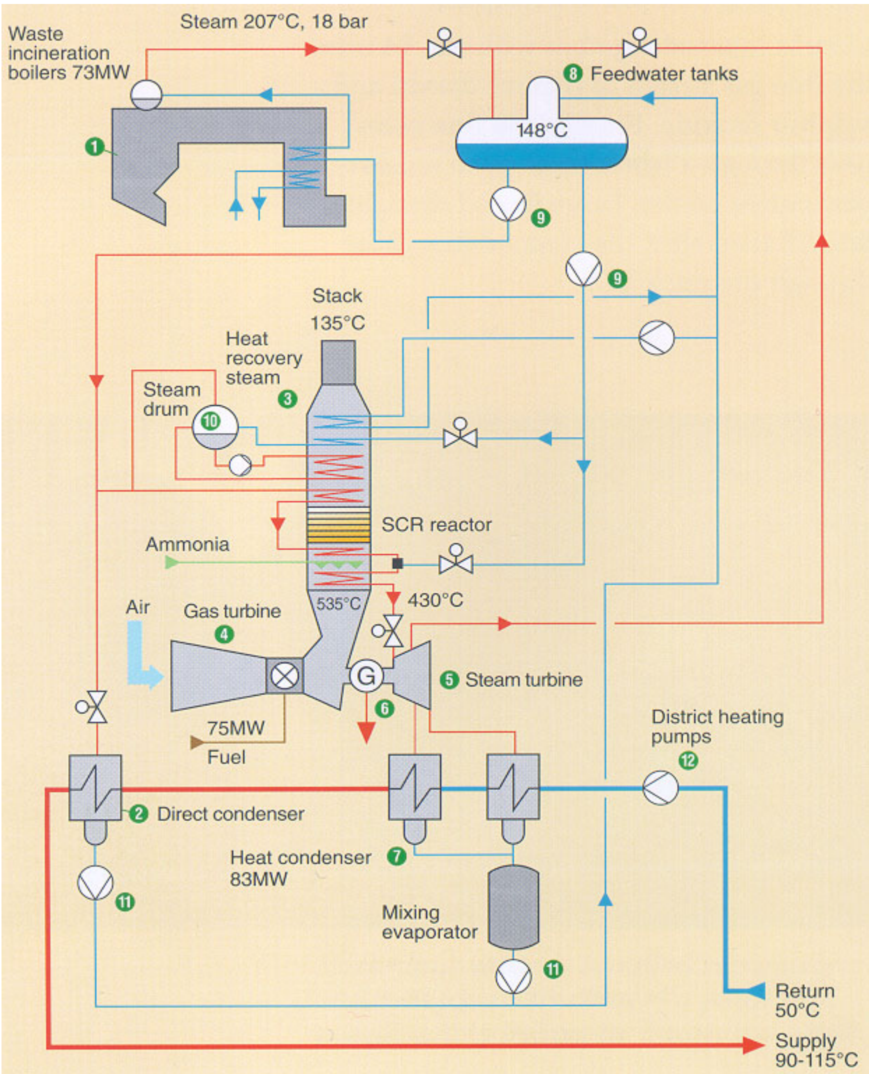


***Fig. 6.7:** Gärstad HCC CHP plant in Linköping, Sweden, with MSW-fired bottoming cycle. Cycle layout and basic parameters.*

The three incinerators generate saturated steam, which is superheated in the HRSG after the gas turbine and fed to a steam turbine. All steam is entirely superheated by exhaust gases from the single gas turbine. Backpressure condenser and another steam extraction further up-flow from the ST provide district heating. The GT, vertical HRSG, ST, electric generator and their auxiliary equipment have been the new additions to the MSW incinerators during the plant conversion into a CHP unit. The GT, ST and electric generator lie on one single shaft.

The steam parameters of the MSW-incinerators are quite modest – 207^{o}C at 18 bar (33.6 kg/s), ruled by the "lowest-cost" solution for conversion of the old hot water boilers into steam generators. Additional steam is produced in the HRSG at a rate of 4,4 kg/s and directly mixed with the steam from the MSW incinerators before the superheater. Steam is superheated to 430^{o}C (with total mass flow of 38 kg/s) and expanded in the ST to 0.86 bar (high-temperature condenser for district heating) and 0.36 bar (backpressure condenser for district heating). The district heat supply at full load reaches 85 MW_{th}. Direct steam condenser with pressure reduction valve is installed for providing district heating (78 MW_{th}) at times when the GT (and consequently the ST) are out of operation.
A Selective Catalytic Reduction de-NOx reactor is installed in the HRSG, between the first and final stages of superheating.

The gas turbine and steam turbine have equal outputs of 25 MW_{el}, leading to total electrical output of 50 MW at full load. ABB Stal AB (presently Alstom Power Sweden AB) has supplied both turbines. The GT is a twin-shaft "GT10" model, fired by LFO. Its efficiency at ISO conditions is 33%. Water is injected in the combustion chamber for NOx reduction. Further NOx reduction is complemented by the Selective Catalytic Reduction unit, which consumes around 100 l/h ammonia-water solution (25% NH_3 in water). The ST is a low-pressure ABB VAX LTDH type.

Fig. 6.8 presents a schematic of the MSW incinerators with their major auxiliary equipment for exhaust gas heat recovery and treatment. Sorted municipal waste is received and stored for a short while in the waste pit / bunker (1). The operator-controlled crane (2) feeds in batches the fuel hoppers of the MSW incinerators. Each incinerator has an inclined travelling grate, to which MSW is continuously fed from the fuel hopper, without any pre-processing. The main combustion zone in the furnace is above the middle part of the grate (3). Fans supply combustion air below the grate (4) and at two other levels up in the furnace. Ash from the furnace falls from the travelling grate into a slag tank, containing water (5). The ash is cooled and conveyed to a slag bunker for subsequent sorting and deposition. Generated steam in the wall-tubes is collected and refined in the steam drum above the furnace (6). Recirculated flue gas is fed back to the furnace from before the stack (7). Urea is injected directly into the furnace (8) for control on NOx emissions. Flue gases after the furnace pass through the second economiser stage and then through an electrostatic precipitator (9), which is used mostly at boiler start-up regimes and in special occasions. Flue gases are further cooled down by the first stage of the economiser and additional district heating surfaces, before they are fed to the flue gas scrubber (10), at a temperature slightly more than 200^{o}C. In the scrubber, flue gases are treated for removing hydrochlorides, ammonia and volatile metals by saturation with water. The toxic pollutants are converted into soluble compounds, mostly acids. After the scrubber, the flue gases with temperature of 150^{o}C enter another district heating heat exchanger (11), which

recovers heat by condensing part of the moisture in the flue gases. Then the flue gases enter a rotating air preheater (12), where moisture is further condensed. Condensate from the scrubber, heat exchanger 11 and the air preheater with very high level of acidity is collected and sent to the water treatment plant (13), where it is neutralised with lime and subjected to complicated treatment. After the air preheater on the flue gas path, lime can be injected (14) for control of dioxin and sulphur emissions. Finally, the flue gases pass through a large bag-house / fabric filter (15), which consists of 700 textile tubes and is the main dust-control component. Before the fabric filter, certain amount of heat is transferred back to the flue gases by water from the district heating heat exchanger 11, in order to assure no condensation in the filter. An ID fan (16) injects part of the flue gases back into the furnace and the rest through the exhaust stack (17) at a temperature of 70^{o}-90^{o}C.
The amount of heat recovered for district heating by condensation of the moisture in the flue gases is around 15 MW$_{th}$, which adds to the heat from the ST condensers, but part of it is transferred back to the exhaust gas to avoid condensation in the fabric filter and stack. The total efficiency of energy utilization from MSW is as high as 95%.

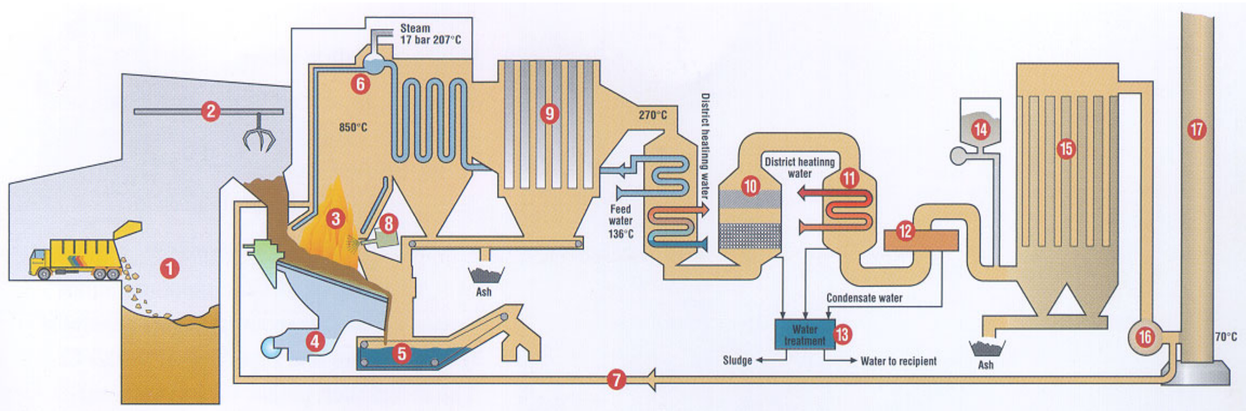


Fig. 6.8: *MSW incinerator schematic in Gärstad CHP plant. Flue gas path.*

The Gärstad power plant configuration is quite unique. Upgrade of the old MSW-fired incinerators into a CHP unit has been achieved with minimum investments, of the order of 6000 SEK per kW installed electric capacity (1994). The specific integration of the combined cycle and the total dependence of electricity generation on gas turbine operation (full superheat by GT exhaust) leads to low flexibility of the power plant, whose main purpose actually is to incinerate MSW. The MSW-fired boilers and gas and steam turbines have very small range of independent operation, yet at small deviations from full power the performance is quite satisfactory. **Table 6.1** presents an example of some load combinations.

The combined cycle CHP unit has been operated without any major problems for several years, supplying 700'000 MWh of district heat on average per year. The usual operating mode for the GT has been around 5000-6000 hours per year, which together with the ST generate 235'000 MWh electricity. The plant burns wastes not only from the local municipality, but also from many other communities within a radius of more than 100 km, with a total of around 600'000 inhabitants. Approximately 220'000 tons of MSW are delivered and combusted every year. Ashes in amount of

50'000 tons are processed for recovery of iron (4'000 t/y) and deposited in a special landfill close to the power plant. The plant has permission and capacity to burn 250'000 tons MSW per year, the three incinerators together can handle 31 t/h (8.6 kg/s).

Table 6.1: *Data for some load combinations of Gärstad MSW-fired HCC.*

MSW incinerators / GT load ratio	%	100/100	80/100	100/80	100/50
MSW incinerators' load	MW	73	58	73	73
GT fuel input	MW	74	74	61	43
Electricity output	MW	49	45	42	32
District Heat output	MW	83	72	79	73
Ratio of electricity output to GT fuel input (el. efficiency attributed to GT fuel only)		0,66	0,61	0,69	0,74

At present, the gas turbine is not operated, due to low electricity prices after the deregulation of the electricity market in Sweden in 1997. The GT fuel prices rule out any economic operation. Only district heating is delivered, by using direct expansion of the saturated steam from the MSW incinerators in the pressure reduction valve and condensation in the direct condenser. The two turbines (GT and ST) are kept ready to start electricity generation whenever necessity may arise.

Information about Gärstad MSW-fired CHP plant in Linköping was acquired from:
Kreij, Sven-Erik, "Cogeneration and Refuse Incineration in Linköping", a brochure issued by Tekniska Verken AB, Linköping, Sweden, April 1994.
Kreij, Sven-Erik, "Konvertering till Kraftvärmeverk", VVS-FORUM, Nr.1, January 1994.
"The Gärstad CHP Plant", a brochure issued by Tekniska Verken AB, Linköping.

And through personal communication with:
Inge Lindahl, project research & development (Tekniska Verken AB, Box 1500, 581 15 Linköping, Sweden).
His contribution is gratefully acknowledged.

The Gärstad waste incineration hybrid CHP plant in Linköping is also described in the recent report by Bartlett and Holmgren [4.9], 2001. The authors investigate the feasibility of converting the gas turbine cycle into an evaporative GT cycle and raising the steam pressure, thus improving the performance. The statements made in the report are supported by extensive simulations and calculations of the cycle modifications. An insight into the effect on CO_2 emissions and district heating load is also provided. [4.9]

The plant is also described by Egard et al. [4.2].

6.2.4. The CHP plant in Karlskoga

Another example of a hybrid power unit from the early 1990's in Sweden is the CHP plant in Karlskoga.

The district heating plant in Karlskoga (owned and operated by Karlskoga Energi & Miljö AB) was initially put into operation in 1985, comprising five boilers, delivering district heat and also hot water and process steam for the nearby industries. In 1991 it was converted into a CHP plant by reconstruction of the boilers and addition of a gas turbine with a heat recovery steam generator and a steam turbine. The total heat output of the plant was also increased.

The gas turbine is a twin-shaft "GT10" model from ABB Stal (presently Alstom Power Sweden AB), fired with liquefied petroleum gas. The design power output of the gas turbine is 25 MW. A one pressure level heat recovery steam generator is situated behind the gas turbine, supplying steam to the steam turbine at 45 bar, 460^{o}C.
The steam turbine is a backpressure type and has a nominal power output of 12 MW. It is installed on a common train with the gas turbine and the electric generator (both turbines have gearboxes). The total net electricity output is 36 MW. Steam from the common header connecting the old boilers is delivered to the steam turbine at 25 bar, 300^{o}C. The backpressure condensers supply district heat.

The five steam boilers utilize a variety of fuels. The smallest one (12 MW_{th}) is a MSW incinerator of the grate type, firing around 35000 tons MSW annually. The next two boilers are oil and gas/oil fired. The last two boilers (CFB boilers of 40 MW_{th} each) were initially fired with coal and peat. Later on, increasing amounts of woodchips were used, together with peat and small amount of coal. Woodchips of various origin can also be mixed with the MSW and fired in the waste incinerator. The boilers deliver steam to their common steam header at 25 bar, 300^{o}C, from which it is distributed to the steam turbine and to industrial processes in the vicinity. The total heat output of the plant is 160 MW_{th}. The total fuel energy input in design mode is 210 MW.

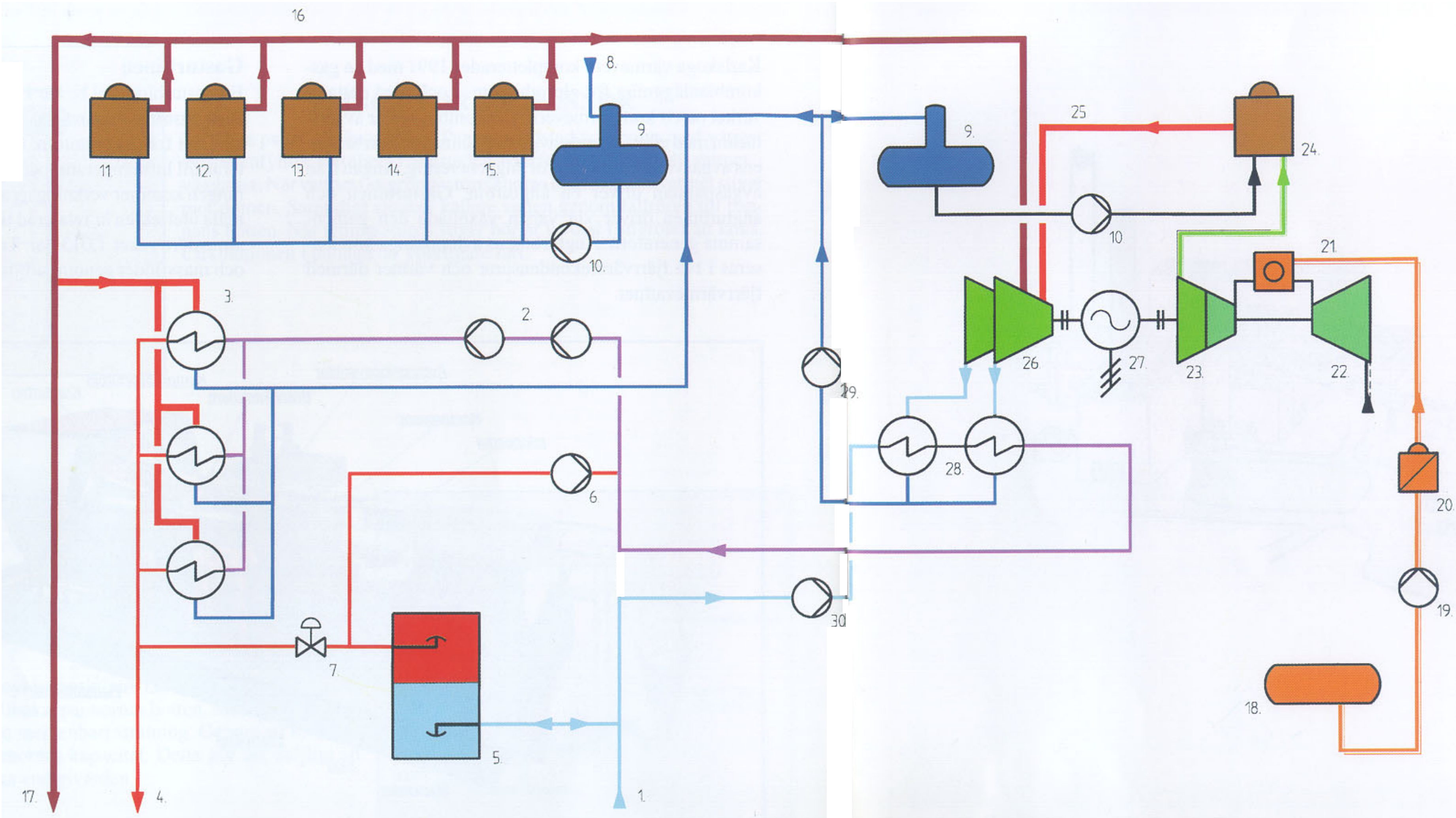
The gas turbine with its heat recovery steam generator have been taken out of operation in the beginning of 1997, after the deregulation of the Swedish electricity market and the sharp decrease in electricity prices. The situation is similar to the other Swedish plants described above, which cannot generate electricity with a profit unless cheaper gas turbine fuel is available. The gas turbine is still kept for back-up power.
The schematic layout of the hybrid cycle in Karlskoga is shown in **Fig. 6.9**.

Information about the hybrid CHP unit in Karlskoga was acquired from:
1. "Karlskoga kraftvärmeverk", a brochure issued by Karlskoga Kraftvärmeverk AB, (presently Karlskoga Energi & Miljö AB), Box 42, 69121 Karlskoga, Sweden.

And through personal communication with:
Bengt Rönnqvist, operation manager, and
Per Lidell, heat & power plant manager, (Karlskoga Energi & Miljö AB, Box 42, 69121 Karlskoga, Sweden).
Their contribution is gratefully acknowledged.

The plant is also described by Egard et al. [4.2].

Fig. 6.9: *Schematic layout of the CHP unit in Karlskoga, Sweden. Main components are numbered in the figure as:* ***1****- district heating return,* ***3****- final heat exchangers for district heating,* ***4****- district heating supply,* ***5****- heat accumulator,* ***9****- feedwater tanks,* ***11****- MSW incinerator 12 MW_{th},* ***12****- oil-fired boiler 25 MW_{th},* ***13****- gas/oil-fired boiler 25 MW_{th},* ***14****- biomass/peat/coal-fired boiler 40 MW_{th},* ***15****- biomass/peat/coal-fired boiler 40 MW_{th},* ***16****- main steam header at 25 bar, 300^{o}C,* ***17****- steam to industrial users,* ***18****- storage for GT fuel (liquefied petroleum gas),* ***20****- evaporator for the liquefied petroleum gas,* ***21-22-23****- gas turbine,* ***24****- heat recovery steam generator,* ***25****- steam at 45 bar, 460^{o}C,* ***26****- steam turbine,* ***27****- el.generator 36 MW_{el},* ***28****- backpressure steam condensers for district heating.*



6.2.5. The CHP plant in Helsingborg

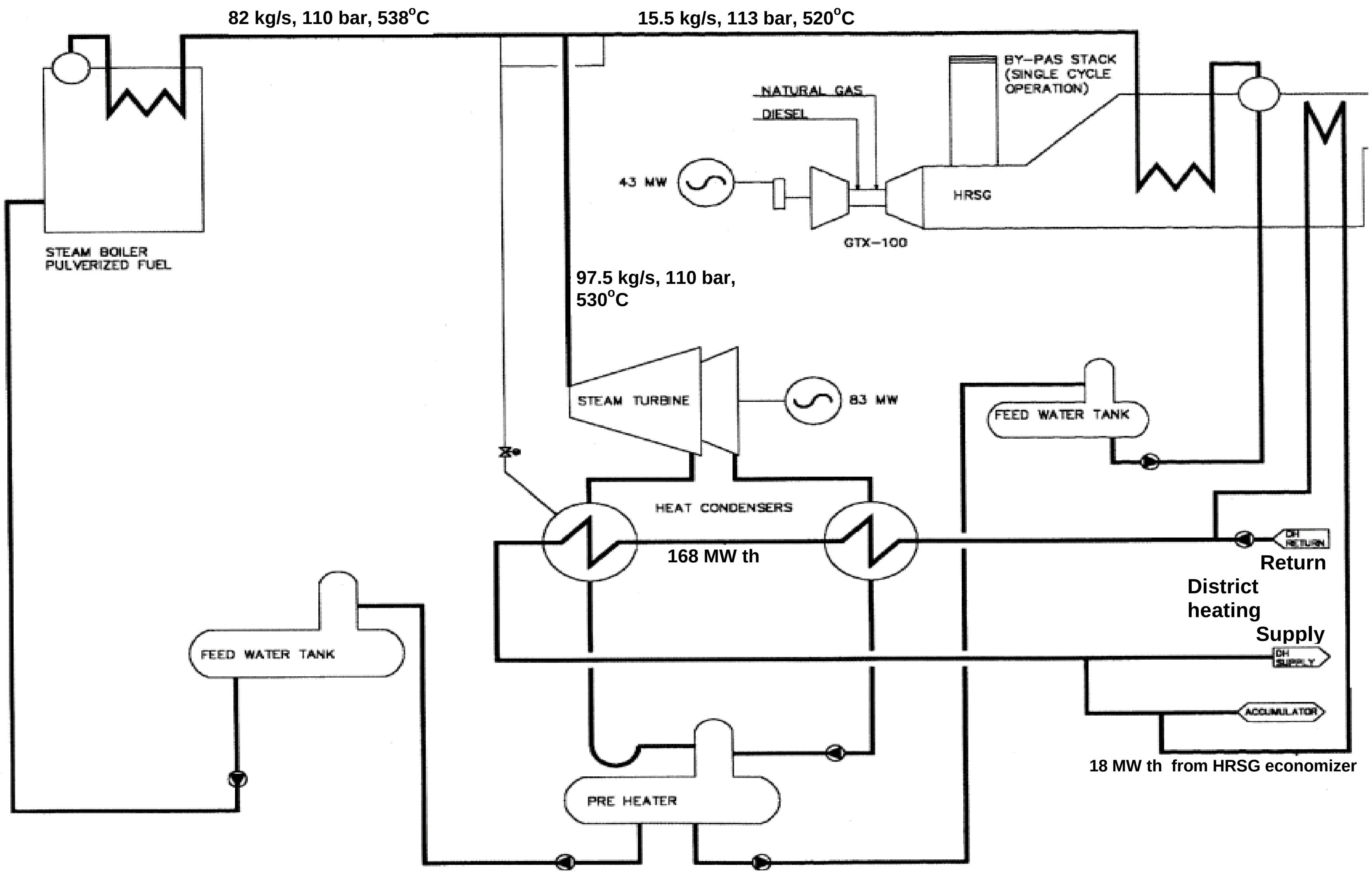
The major CHP unit in Helsingborg, south Sweden, has been recently reconstructed and upgraded into a hybrid combined cycle. This is the newest installation in Sweden and the commercial première for the "GTX100" gas turbine model developed by ABB Stal AB (presently Alstom Power Sweden AB).

The major CHP unit in the town of Helsingborg, a steam cycle fired mostly by coal, owned and operated by Helsingborg Energi AB (presently Öresundskraft AB), had a capacity of 64 MW of electricity and 132 MW of district heat. Necessity has arisen for an increase of the unit's capacity in order to cover the growing local heat demand and the sharply growing electricity demand throughout Southern Sweden, together with increasing fuel flexibility and reliability.
In accordance with the municipality's strive for lower environmental impact, firing of wood pellets has been introduced in the boiler since 1997, replacing most of the coal. Furthermore, natural gas is available in the region, so any future expansion should have obviously been based entirely on natural gas as fuel.

In the end of 1997, decision was taken to install new capacity at the plant's site. Two options were considered – extension of the steam plant based on a condensing steam turbine or integration of a gas turbine into the old steam cycle. The second option was viewed as much more attractive for the specific conditions in the town's energy system (specific correlation of electricity and heat loads, good power-to-heat ratio and decrease of coal consumption). With the support of the Swedish National Energy Administration and the Delegation for Energy Supplies for Southern Sweden, ABB Stal AB of Sweden has been awarded a contract for the installation of their first unit of the very new model "GTX100", officially introduced in mid-1997. The final cycle configuration resembles those of the Finnish industrial CHP plants described in the first section of this chapter. The GT with a HRSG works independently from the main boiler and produces additional steam in parallel to the boiler, which is mixed with the steam from the boiler in a HP steam header and is fed to a common steam turbine. The total output of the hybrid combined cycle plant would be 126 MW of electricity and 186 MW of heat for district heating, leading to a power-to-heat ratio of 0.68, at total efficiency of 90.6%.
The combined cycle layout is shown on **Fig. 6.10**.

The gas turbine "GTX100" is developed as an industrial machine with efficiency rivalling that of the aeroderivatives, while featuring simple construction and low costs, low maintenance requirements and high availability and reliability. The GT has an electric output of 43 MW at ISO conditions (with NG as fuel), pressure ratio of 20:1 and thermal efficiency of 37%, maintained down to 70% load. Exhaust gases leave at 546^{o}C with a mass flow of 122 kg/s.
The GT is of a single shaft arrangement, with 15 compressor stages and 3 turbine stages. The compressor has been the object of some special design procedures, resulting in controlled diffusion airfoils and variable geometry for the first three stages. Tip leakage is minimised by applications of novel materials.
The combustor is of the annular type, having 30 burners of the AEV design for dry ultra-low-emission combustion. NOx and CO emissions are below 15 ppmv on natural gas and below 25 ppmv on liquid fuel, over a load range down to 50% (at 15% O_2).

Fig. 6.10: *Schematic layout of the newly reconstructed CHP unit in Helsingborg.*



Dual-fuel capability of the gas turbine is a built-in feature and switchover between fuels (gas or oil) at full load is possible.
The expander is built as one module and is bolted to the compressor shaft. Its three stages are highly loaded. The first stage vanes and blades are made of a single-crystal material and generous cooling is provided by extraction of air from the compressor. The GT runs at 6600 rpm.
In a pure combined cycle configuration, the GT can reach 62 MW net output at 54% efficiency with dual-pressure steam bottoming cycle without reheat.

The main boiler of the old steam cycle is a pulverized coal steam generator, using also HFO as support and start-up fuel. Its full-load capacity is 220 MW_{th}, which generate 82 kg/s steam at 110 bar, 538^{o}C. Firing of wood pellets has been introduced through the same fuel preparation system. However, full load cannot be reached only on wood pellets, so a 50/50 mixture with coal is usually used.
A HRSG with a bypass stack is installed behind the GT. Steam at a single pressure of 113 bar, superheated to 520^{o}C with a flow of 15.5 kg/s, is generated by the GT exhaust and mixed with the steam from the main boiler before being fed to the ST. A district heat economiser surface utilizes the low-temperature heat of the GT exhaust and adds 18 MW_{th} to the 168 MW_{th} from the ST condensers.

The GT, its electric generator (driven through a gearbox) and a HRSG are the new components of the combined system. Interesting approach towards the technical problems has been taken. With careful planning, the GT is installed on a minimum space, taking advantage of its small footprint, while still leaving the possibility for building a new (larger) steam boiler in the future. The steam turbine of the old steam cycle (type ATM, supplied by ABB Stal AB) has been refurbished in order to be able to handle the increased steam flow, raising its rated power output from 64 MW to 83 MW. The electric generator of the ST is not replaced, it is just provided with additional cooling and its power is increased to the desirable point (up to 83-85 MVA, originally rated at 75 MVA) by running at a power factor close to unity (1.0). In compensation for the reactive energy, the GT generator is chosen over-sized and will be run at a very low power factor, around 0.65 (its normal power factor being 0.85).

The hybrid configuration of the CHP unit in Helsingborg will have a wide operating range. Both the main boiler and the GT can be run independently. The ST can be operated with steam only from the HRSG, thus constituting a pure combined cycle with the GT, which would produce 56 MW_{el} and 53 MW_{th} (the main boiler shut down). The GT plus the ST working with steam from the HRSG will cover the heat demand up to the point where the main boiler has its minimum load. The boiler is actually the least flexible component, as its stop/start operating sequences are limited and energy consuming. The GT cycle will cover the large variations in load, both in periods of low heat demand (when the main boiler is not operating) and in periods of intermediate heat demand (when the main boiler operates at part load mostly on wood pellets, without large changes of load).

The total costs for the project have been around 220 million SEK (1999) or 28 million USD. The hybrid combined cycle plant was recently commissioned and is expected to produce 420 GWh of electricity annually. Operation time per year will be about 6000 hours, of which about 3500 hours together with the main boiler. The decrease of operating hours for the main boiler (down from 5500 hours before the addition of the

GT) further contributes to the decrease in coal consumption and provides lower NOx and CO emissions levels. Total reduction of NOx is estimated to be around 40%. A decrease in CO_2 emissions directly follows from the substitution of coal with wood pellets and the lower operating time for the main boiler, as well as from the fact that the electrical production form the GT substitutes coal-based electricity imports from Denmark.

Information about the CHP unit in Helsingborg was collected from:

1. Olsson, Christer; Nilsson, Bertil "A first at Västhamn", Power Engineering International, November 1998.
2. "The Combined Cycle Power Plant in Helsingborg", a brochure issued by ABB Alstom Power (now Alstom Sweden AB), Energimyndigheten (Swedish National Energy Administration) and Helsingborg Energi (now ÖresundsKraft AB), 1999.
3. "Produktion", a brochure issued by ÖresundsKraft AB, 2001.
4. Engvall, Jessica, "Bättre Miljö i Helsingborg med Naturgas", Kraft Ordet, Nr.4, 1999.
5. "Production of Electric Power and District Heating", a brochure issued by Helsingborg Energi AB, 1997.

Most of the information materials have been provided by Christer Olsson and Lena Phalén (ÖresundsKraft Produktion AB, Box 642, 25106 Helsingborg, Sweden).
Their contribution is gratefully acknowledged.

6.3. Denmark

6.3.1. The MSW incinerating CHP plant in Horsens

A good example of a hybrid CHP unit where a gas turbine is combined with MSW incinerators can also be found in the town of Horsens, Denmark. Its specific feature is the parallel steam production from the gas turbine (with its heat recovery steam generator) and the two waste-fired steam boilers, at the same pressure and temperature of 47 bar, 425 ^{o}C. The GT exhaust is not used for superheating of the MSW-generated steam, rather than generating the main amount of steam delivered to the steam turbine through the common steam header.

The gas turbine unit has a nominal power output of 22 MW and is fired with natural gas. The steam production from its one pressure level heat recovery steam generator corresponds to 7 MW power output from the steam turbine.

The steam turbine with its own electric generator have electric output of 13 MW. Total electric output from the overall cycle is 35 MW. District heat is delivered from the backpressure steam condensers behind the steam turbine and an economiser surface in the GT exhaust. The total heat output from the plant is 43 MW_{th}, out of which 27 MW_{th} can be attributed to the natural gas and 16 MW_{th} to the MSW incineration in nominal operation. On an annual average, MSW delivers 57% of the total heat supplied to the district heating network and 29% of the electricity produced.

The two MSW incinerators utilize up to 71000 tons of waste annually, which corresponds to around 52% of the total fuel energy input into the cycle. Natural gas supplies the other 48% of the total energy input on annual average.

The Horsens' CHP plant has been in trouble-free operation since its commissioning in January 1992. Presently the plant is owned and operated by the Danish power utility ELSAM. The annual net production of electricity and district heat is around 140 GWh_{el} and 230 GWh_{th} respectively. These annual figures can reach a maximum of 188 GWh_{el} and 243 GWh_{th}. The electric efficiency is estimated at 36%, usually lying around 29% on net year average. The total efficiency is 80% on year average.
The cycle schematic is shown in **Fig. 6.11**.

Information about the hybrid CHP unit in Horsens was acquired from:
1. "Horsens Combined Heat and Power Plant", a brochure issued by I/S Skærbæk Power Company (a subsidiary of Elsam A/S), 1991.
2. "Horsens Kraftvarmeværk A/S. Miljøredegørelse 2000", a brochure issued by Horsens Kraftvarmeværk A/S, Endelavevej 7, 8700 Horsens, Denmark.

And through personal communication with:
Leif Sørensen, heat & power plant manager (Elsam Affald og Energi, Horsens KraftVarmeværk A/S, Endelavevej 7, 8700 Horsens, Denmark).

The hybrid unit in Horsens is also described by Egard et al. [4.2].

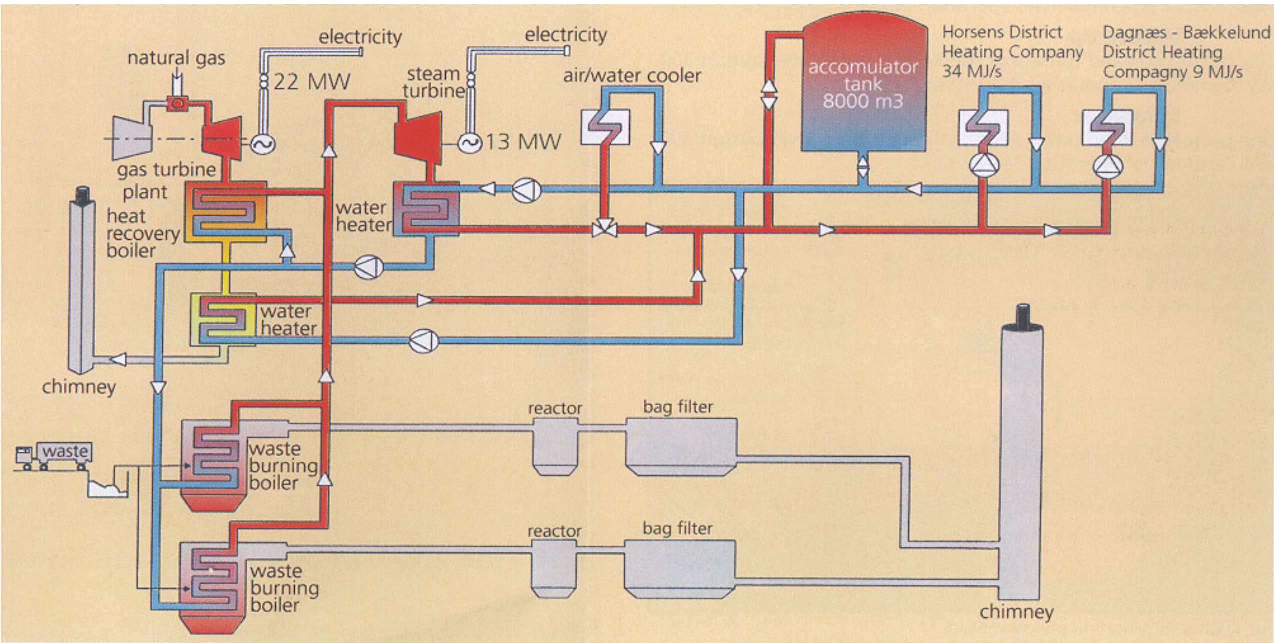


Fig. 6.11: *Schematic of the hybrid combined cycle in Horsens, Denmark. Natural gas fired gas turbine with heat recovery steam generator, parallel to MSW incinerators, delivering steam to a common steam turbine at 47 bar, 425^{o}C.*

6.3.2. Avedøre 2

One very new CHP unit in Denmark deserves attention. This is the second unit (just recently commissioned) of the Avedøre CHP plant, built by a joint venture of Denmark's SK Power and Sweden's Vattenfall, situated about 10 km south of Copenhagen city centre.

The unit comprises one supercritical steam cycle with a large steam generator fired with natural gas, two aeroderivative gas turbines fired with natural gas whose exhaust will be used for feedwater preheating for the steam cycle, and one steam generator (again with supercritical parameters) fired with biomass – straw and woodchips, feeding steam to the common steam turbine.

Avedøre 2 features generous deployment of advanced materials and novel design. The steam parameters are ultra-supercritical – pressure of 305 bar and temperature of 582^oC (reheat to 600^oC). The main boiler is a once-through boiler of the straight-up design, all convective water/steam heat exchangers are situated above the furnace in a vertical upward duct. Only the air preheater is situated separately in the flue gas path beside the boiler, with a downward flow of the flue gases, after the Selective Catalytic Reduction (de-NOx) unit. The total height of the boiler reaches 80 m.

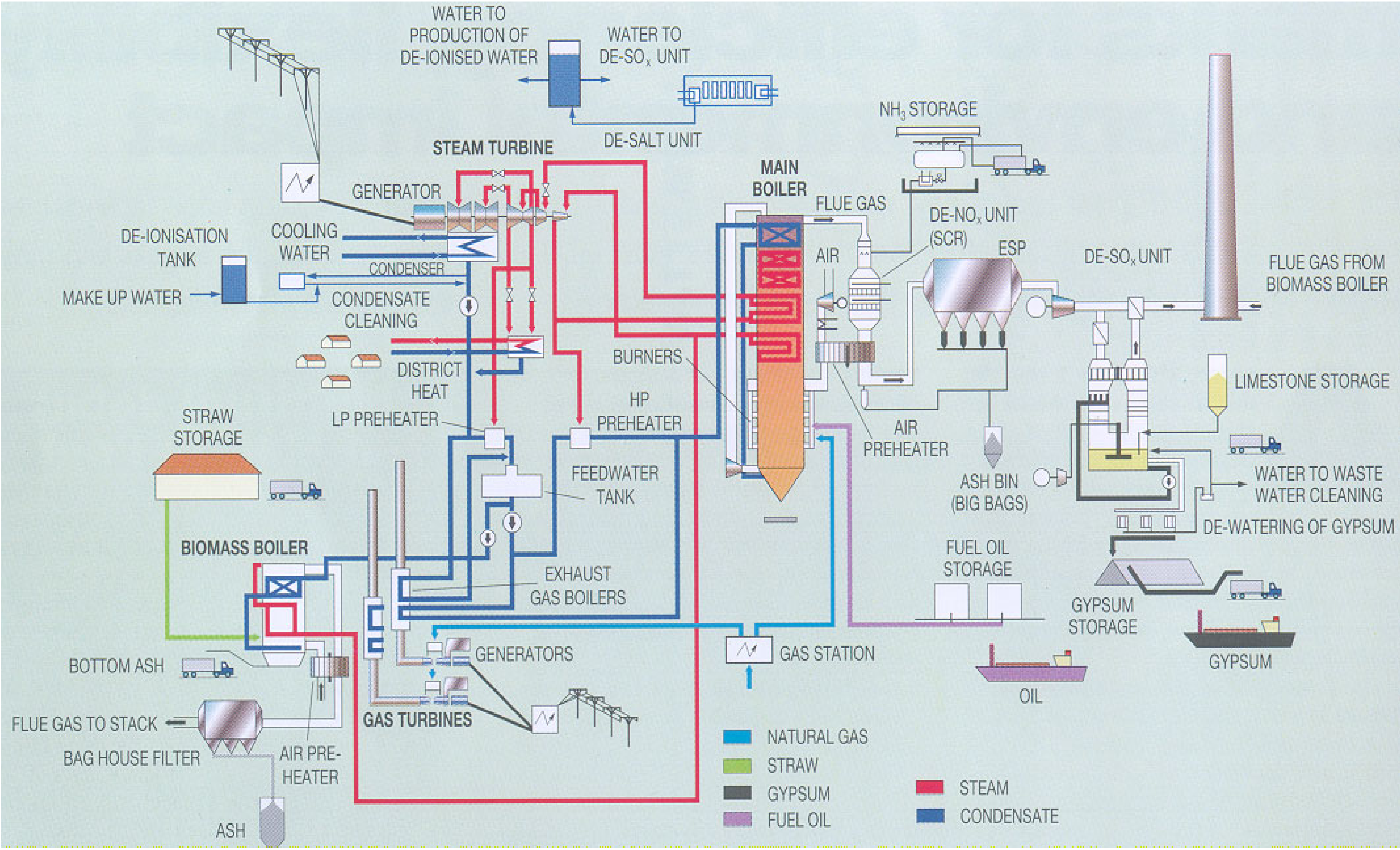
The main boiler has been designed and supplied by FLS Miljo/BWE. The fuel input is 800 MW, which corresponds to 16.57 kg/s (20.6 Nm^3/s) natural gas. It has been initially conceived as a coal-fired steam generator, but the growing pressure against use of imported coal in Denmark, the intentions to reduce CO_2 emissions and the availability of NG at reasonable prices led to its transformation into a NG fired boiler. HFO with sulphur content of up to 3% will be used as back-up fuel (de-SOx unit is installed for desulphurisation of the flue gases when HFO is used). The high-temperature tubing (supplied by various sub-contractors) incorporates the most extensive large-scale commercial application of advanced materials in Europe.

The steam turbine itself is a modern high-performance machine with high isentropic efficiency and HP section employing advanced materials. It has been designed and manufactured by Ansaldo Energia at their Genoa Campi production facility. The steam turbine consists of five bodies on a single shaft – one HP, two MP and two LP sections. The nominal live steam parameters at the turbine inlet are 300 bar, 580^oC, 336 kg/s. Steam is reheated to 600^oC at 74 bar and flow of 284 kg/s. The steam cycle is a co-generation one, with cold condenser and steam extractions for district heating. Low-temperature seawater is used for condenser cooling, providing condenser pressure of 0.023 bar. Steam for district heating purposes is extracted at two different pressure levels after the second MP section. At part-loads the turbine is operated with full throttling admission and sliding pressure. The maximum gross output from the steam turbine for the overall combined cycle is 535 MW_{el}. Maximum district heating output can reach 620 MW_{th}.

The steam cycle configuration employs 11 (eleven) feedwater preheaters – six LP ones, a deaerator, and four HP ones. Feedwater is preheated to 320^oC.

The cycle layout is shown on **Fig. 6.12**.

Fig. 6.12: *Layout of the "Avedøre 2" hybrid combined cycle power plant in Denmark.*



The two gas turbines are Rolls-Royce Industrial Trent aeroderivative models. In simple cycle mode at ISO conditions, they provide 51 MW electrical output each, at 41.5% efficiency. The GT pressure ratio is 35, exhaust temperature 426^{o}C. The compressor-turbine arrangement is derived from the Rolls-Royce Trent three-spool high-bypass turbofan engine designed for large airliners. The aero fan is replaced by two LP compressor stages. The final two stages of the LP turbine have also been re-designed specifically for industrial service. Dry low-NOx combustion chambers assure emission levels not higher than 50 ppm NOx and 31 ppm CO. The GT exhaust gas heat is used to preheat the steam cycle feedwater in specially designed heat exchangers supplied by Aalborg Industries. They are situated in parallel to the standard feedwater preheaters of the steam cycle, as in a typical parallel-powered hybrid combined cycle configuration with feedwater preheating. The gas turbines will be operated as peak-shaving capacity, not in continuous base load.

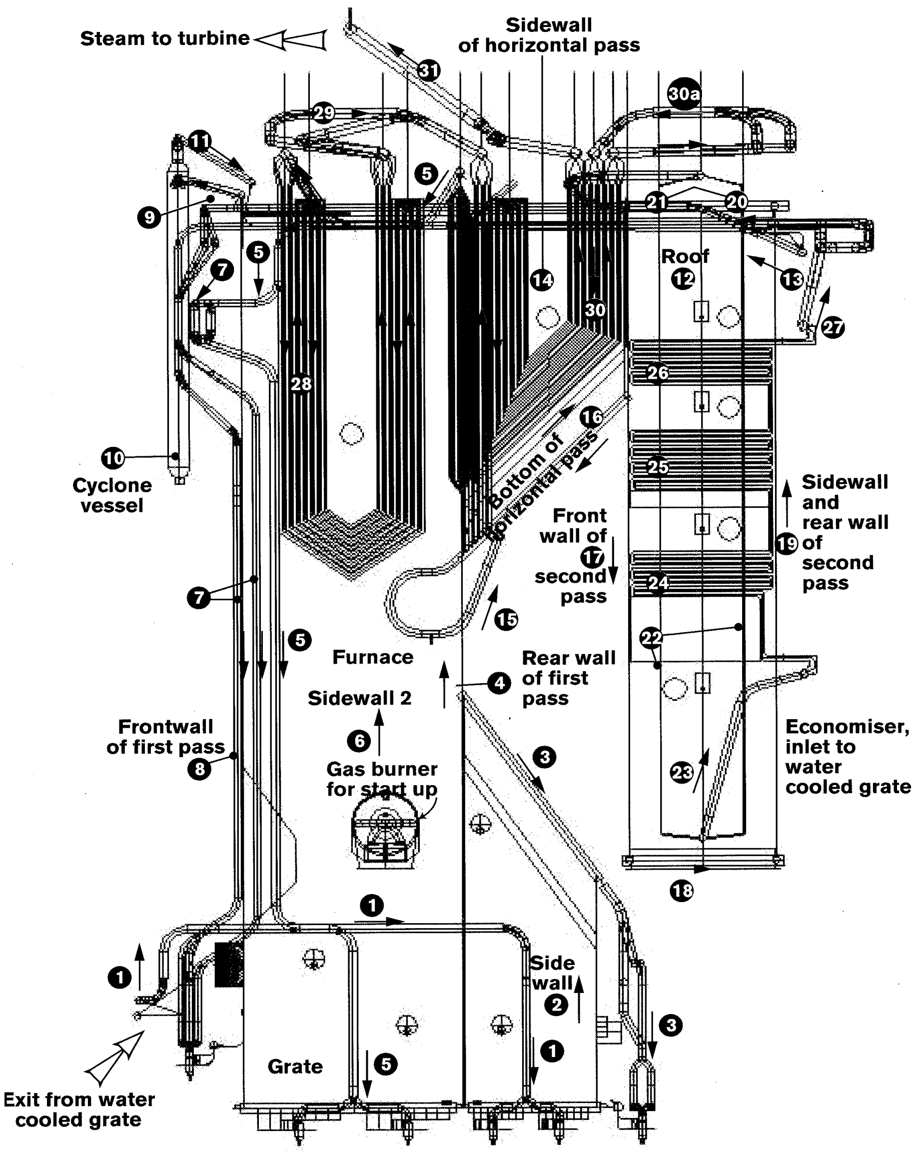


Fig. 6.13: *Straw-fired boiler with supercritical steam parameters for the "Avedøre 2" hybrid combined cycle. Numbers and arrows show water/steam flow in the boiler.*

The biomass boiler is the third important element in the power cycle. It generates superheated steam, which is directly mixed with the steam from the main boiler and fed to the ST. There is no reheat in the biomass boiler, all reheat is performed in the main boiler. The biomass boiler schematic is shown in **Fig. 6.13**.

The biomass boiler has been designed and constructed by a consortium of Ansaldo Vølund and Babcock Borsig Power – AE Energietechnik. It is the world's largest and most effective straw-fired boiler with a heat input of 105 MW, corresponding to 26 t/h straw. Steam parameters are ultra-supercritical, similar to the ones for the main boiler – 310 bar, 583^{o}C. The boiler is of one-through design, with configuration comprising a radiant furnace, convection part in a horizontal flue gas duct, vertical downward flue gas duct and another vertical upward duct. Advanced materials are employed. The tubing in the combustion chamber is vertical, in several passes each with upward flow. There are downcomers between the upward passes and in some points special mixing devices to distribute water and steam evenly. The horizontal and downward ducts also have membrane wall-tubes, in which the steam is already slightly superheated. First stage superheater is situated in the downward duct, last stage superheater is in the horizontal duct. Water flow is illustrated by arrows and number-points on Fig. 6.11. First stage economiser and air-preheater are not shown on the figure, they are situated in the subsequent upward flue gas duct.

The furnace features a water-cooled vibrating grate and water-cooled channel with screw feeders and regulating table for feeding of disintegrated straw (also woodchips in the future). The boiler is optimised for efficient and problem-free utilization of difficult biomass fuels like straw. Emphasis is put on oxygen distribution in the furnace, control of local temperatures in the different zones and control of ash behaviour. The fuel has an ash content of 5.2% (with various dangerous clinker-forming elements, typical for herbaceous biomass), and LHV of 14.3 MJ/kg. Generally, assuring low temperature and high oxygen content in certain critical zones can prevent formation of clinker by the ash in the furnace. Combustion above the feeding table is prevented by geometry and secondary air jets. The main gas phase combustion zone (after initial pyrolysis of the fuel) is localised in the centre of the furnace, giving even thermal loading on all walls and high radiation flux to the vibration grate, assuring high char burnout. Secondary air jets create a vortex in the furnace above the grate. Tertiary air jets are located high in the throat, before the radiative superheaters, to assure high burnout of the volatiles and low CO emissions. The temperature in the centre of the furnace just above the grate reaches 1600^{o}C, dropping down to 792^{o}C at the entrance of the horizontal duct before the convective superheater. The exhaust is cooled down to 115^{o}C after the air preheater and is passed through a bag-house filter. Fly-ash from the filter is dumped, while bottom ash from the furnace will be used as fertiliser.
The emissions from the biomass boiler are 625 ppm CO and 240 ppm NOx.

The biomass boiler is scheduled to run at full power whenever there is enough fuel available. It generates 40 kg/s superheated steam, which is about 13.5% of the steam generation in the main natural gas fired boiler. Feedwater for the biomass boiler is extracted right after the common feedwater tank, it has a temperature of 180 to 230^{o}C, depending on the load. The ratio of biomass fuel in the biomass boiler to natural gas in the main boiler is very low, yet the presence of the biomass boiler converts the overall combined cycle into a hybrid one and presents an unique example of the growing importance of high-efficiency biomass utilization for power generation.

The performance of Avedøre 2 power unit is breath taking. In condensing mode the steam cycle without the gas turbines is able to deliver 430 MW_{el} net power at a net efficiency of 49.2% (LHV). This high efficiency of the steam Rankine cycle is due to the supercritical boiler parameters, sophisticated arrangement, deep vacuum in the condenser and very low losses in the steam turbine. With the gas turbines in full power, the overall net electric power output reaches 570 MW at 51% efficiency. The power unit will be actually operated in co-generation mode with average outputs of 360 MW_{el} / 480 MW_{th} without gas turbines or 485 MW_{el} / 545 MW_{th} with gas turbines.
The total efficiency in co-generation mode will be 94%. The overall hybrid combined cycle power unit is extremely flexible. The biomass boiler can be kept at full load when the main boiler is operated between 20-100% of its load range, one GT can be kept at full load while the steam cycle works within 35-100% of its load range, both gas turbines can be operated at full load when the steam cycle operates within 50-100% of its load range.

Information about the new power unit in Avedøre was acquired from:

1. Anonymous, “Avedøre 2 sets new bench-marks for efficiency, flexibility and environmental impact”, Modern Power Systems, Vol.20, No.1, January 2000, pp 25-36.

6.4. A few words about other relevant examples in Northern Europe

6.4.1. The MSW incinerator in Alkmaar, The Netherlands, topped by internal combustion engines

A unique example of a hybrid combined cycle comprising a gas-fired ICE and a MSW incinerator is the CHP plant in Alkmaar, The Netherlands.

In 1992, the consortium VOF NRS decided to rebuild the MSW-fired power unit in Alkmaar. Steam parameters of the MSW-fired boiler have been increased and gas engines have been installed as a back-up power in times of malfunctions of the MSW cycle equipment, which could be detrimental to the boiler.
As long as natural gas supplies in the area are abundant and at attractive price, it has been clear that the gas engines could be run continuously and the power plant (including the steam turbine of the MSW boiler) could generate more power than the whole local community itself consumes. Moreover, it has been obvious that efficiency increase of the MSW-based steam cycle can be attained, if the engines' rejected heat is used to preheat the feedwater for the steam cycle. Such a conversion has been done with minimal expenses in 1995, while the positive impact on MSW-based cycle efficiency has been remarkable (having in mind that the original steam cycle has had actually no feedwater preheaters except the deaerator).

The engines installed in Alkmaar are three Wärtsilä 16V25SG, with 2.8 MW electric power output each, electric efficiency of 38.5% and total power output of 8.4 MW_{el}. The jacket water, first stage charge air cooler and oil cooler provide preheating for the condensate stream before the feedwater tank, while the engines' exhaust gas further transfers heat to the feedwater after the feedwater tank. The steam turbine has power capacity of 40 MW_{el}.
Specific investments for such a hybrid power plant (including price of engines) has been estimated to be around 1800 €/$kW_{el,}$ which is just 60% of the investments needed for a new advanced MSW-fired steam cycle (around 3000 €/kW_{el}).

Information about Alkmaar hybrid co-generation unit has been acquired through personal communication with:
Thomas Stenhede (Wärtsilä Sweden AB, Box 920, SE-461 29 Trollhättan, Sweden).
His contribution is gratefully acknowledged.

6.4.2. MSW incinerator and gas turbine HCC in Germany

"Entsorgungsgesellschaft Mainz" has signed a contract for construction of a new MSW incinerator/steam generator, with a capacity of 700 t/day in Mainz, Germany. It will have the standard steam parameters for MSW-fired steam generators of 400^{o}C at 40 bar. An integrated topping gas turbine will provide superheating of the MSW-generated steam up to 555^{o}C, before feeding it to the steam turbine. The GT will be fired with natural gas.
The HCC unit is expected to enter commercial operation in the end of 2003 and is the first such installation in Germany.

Information about this new MSW HCC installation in Germany was acquired from:
Corfitz Norén (Svenskt Gastekniskt Center AB, 205 09 Malmö, Sweden).
His contribution is gratefully acknowledged.

6.4.3. Industrial HCC co-generation for the pulp & paper industry in Germany

Two parallel-powered hybrid combined cycles comprising a gas turbine and a bio-residues fired FB boiler have been installed at two pulp & paper factories in Germany in 1993/1994, in Baienfurt by Stora Billerud GmbH and in Eilenburg near Leipzig by Sachsen Papier Eilenburg GmbH. Both cycles closely resemble the two such ones in Finland, described in the beginning of this chapter. Indeed, the parameters are almost the same, as long as all these units work in industrial CHP mode, where the main energy supply is in the form of steam for pulp digesters and paper machines, together with producing electrical power for covering the mill's own needs.
The history and structure of both these installations is similar. Old heat and power supply for the mill based on oil-fired boilers have been replaced with new FB boilers fired with biomass residues from the mill, integrated with a gas turbine. The GT is fired with NG and has supplementary firing in the exhaust stream before the exhaust gases enter the HRSG (with possibilities for fresh air operation without the GT), where steam is generated in parallel to the biomass-fired boiler.

The GT in Baienfurt is a FT8-30 industrial version of the Pratt & Whitney's JT8D-219 aircraft engine. It has a three-spool arrangement and delivers 25.5 MW shaft power at ISO conditions at the site of installation, with a thermal efficiency of 36.4%. The exhaust flow is 85.5 kg/s at 439^{o}C. Water is injected in the combustion chamber for NOx reduction down to 100 mg/m^3 at 15% O_2 dry.
The biofuel boiler in Baienfurt has a thermal output of 15 MW.

The unit in Eilenburg features a Frame 5 GT supplied by European Gas Turbines GmbH in Essen, Germany. The GT has power output of 27 MW_{el} at 28.3% efficiency, exhaust flow of 124.8 kg/s at 488^{o}C. The HRSG is horizontal with a single steam pressure level. Steam is produced at 84 bar, 490^{o}C, 25 kg/s. Economiser surface extracts further 21 MW heat from the GT exhaust in the form of hot water. Both the HRSG and the biofuel boiler have been supplied by Tampella Power Inc. The whole CHP unit has been constructed by IVO International Ltd. (now renamed Fortum).
The biofuel boiler is a BFB type, burning mostly sludge residues in amount of 196 t/day. Steam is generated in amount of 11 kg/s, 490^{o}C and is supplied through the common steam header at 84 bar to the steam turbine, together with the steam from the HRSG. The ST has an electric output of 17 MW_{el} and is supplied by ABB Turbinen Nürnberg GmbH. Steam extractions at 25 bar and 3.8 bar feed the heat consumers with 129 t/h steam in total.

7. DISCUSSION AND CONCLUSIONS

There is already a stable trend for constant increase of biomass use for energy purposes in Sweden. Having in mind that further expansion of hydropower is very limited and no new nuclear units are to be built, enhanced biomass utilization for electricity production with higher electric efficiencies and lower costs is essential.

Domestic Swedish biomass resources are vast and renewable, but not infinite. They must be utilized as efficiently as possible, in order to make sure that they meet the conditions for sustainability in the future.

When talking about the CO_2 neutrality of biofuels, one must always bear in mind that reduction of CO_2 emissions resulting from the use of biofuels is less than 100%. A certain amount of fossil fuels is always involved in the production, transportation and handling of biofuels, which decreases the positive effect of their utilization. Fossil fuel involved in the production and construction of power generating facilities must also be included in the accounting. Many studies have been carried out in an attempt to estimate the real (actual) emissions reduction of all pollutants from the use of biofuels, following the so called "life cycle" approach, taking in mind all fossil energy involved in the life cycle of biofuel utilization, facilities construction and facilities demolition. The exact figures for energy intensity of certain products and processes are somewhat bleak and approximated, but in general it has been proved that biofuel energy utilization leads to a considerable net reduction of CO_2 and other pollutant emissions.

In Sweden, the unique situation of very low electricity prices (governed by hydropower and nuclear power in the deregulated Nordic power exchange market) makes it difficult for biomass power units to compete in price of electricity. Furthermore, the high need for heat generation for district heating purposes (which cannot be provided by hydro or nuclear power) raises the price of heat energy and does not provoke higher electric efficiencies in thermal power stations.

This situation (i.e. low electricity prices and no motivation for higher electric efficiencies) is not expected to be permanent. The development towards sustainable energy systems will promote:

- Higher electric efficiency while co-producing a certain amount of heat with high total efficiency,
- Rational, wise and efficient use of biomass resources.

There are many technologies for biomass energy conversion, aiming at either direct utilization of biomass thermal energy in combustion processes, or at upgrading biomass to more valuable gaseous and liquid fuels, applicable in high-efficiency power cycles or as expensive fuels for transportation. Some of these advanced technologies are well-developed and commercial, others are still on the test-bench or are not economically feasible yet, although promising.

When stationary power generation is in the spotlight, traditional simple combustion in boilers is still the most readily applicable method, featuring good overall energy conversion efficiency and highest simplicity and reliability among all technologies for biomass energy conversion.

There is plenty of room for electric efficiency enhancement of traditional biomass power units, either by raising steam parameters and applying modern steam turbines, or by alternative options, such as externally fired gas turbines for example (with or without a combination with a steam turbine).

One easy way to utilize biofuels with higher efficiencies, while keeping the process simple and reliable and using only standard equipment, is the incorporation of a biofuel-fired power cycle as bottoming cycle in a hybrid combined cycle with a topping high-grade fuel fired heat engine. A small amount of fossil fuel in the topping cycle can provide higher efficiency of biofuel utilization in the bottoming cycle. To reach full sustainability and higher reduction of CO_2 emissions, the topping cycle can be powered by biomass-derived high-grade fuels.

The hybrid combined cycle concept with coal fired bottoming cycle is a viable option, technically and economically. Such cycles can effectively combine the increasing share of natural gas as fuel for power production with the traditionally strong presence of coal on the power front. Such installations exist since many years and new power units in HCC mode are being constructed, although not in large numbers.
The thermodynamic advantages of HCC power units have been demonstrated. Increased efficiencies of HCC combination of a TC and a BC, compared to the average efficiency of two separate simple cycles with the same total power output, have been registered, according to the theoretical expectations.
The cost advantages of HCC power units have also been proven. Low risks, short construction times, standard equipment and limited need for development are other advantages. The concept is readily applicable to biomass and MSW as fuels for the bottoming cycle.

Natural gas and biofuel powered HCC units are a simple and affordable way to increase the electric efficiency of biofuel energy utilization, without big investments, uncertainties or loss of reliability arising from complicated technologies.
Configurations of such power cycles are very flexible and reliable. They provide high electrical efficiency in condensing mode and high total efficiency in CHP mode.

Incinerating MSW as bottoming fuel in hybrid combined cycles relieves communities from the burden of disposing it of in landfills and at the same time provides efficient utilization of its energy content. This is not achievable in simple MSW-fired steam power cycles, due to the low steam generator parameters, hindered by aggressive flue gases. Repowering existing waste incineration units with topping cycles can lead to a remarkable increase in electric efficiency of MSW energy utilization.

The HCC concept for MSW-fired boilers with topping natural gas fired GT is surely the most cost-effective method for increasing the efficiency of MSW energy utilization. Moreover, it is probably the option with the greatest efficiency improvement potential, within the reasonable cost and scale limits.

Internal combustion engines can also be applied as topping cycle for solid-fuel-fired boilers in a HCC configuration. They can be particularly attractive in small scales with biomass fired bottoming cycle in CHP mode, showing good performance, high electric efficiencies, low investment costs and low running costs.

Internal combustion engines still produce more NOx per unit generated power than modern gas turbine combustors, but in HCC configurations reburning of the engine exhaust in the bottoming cycle can be used to substantially reduce NOx emissions without other special equipment. This applies especially to the case when fluidized bed boilers are used as BC.

The outcome of this Literature Survey can be concluded as follows:

- The Hybrid Combined Cycle concept is technically and economically viable. It has been largely demonstrated in various configurations, mostly with natural gas as topping fuel and coal as bottoming fuel. Part-load characteristics of such hybrid power plants are superior to those of separate units.
  Improvements in efficiency due to integration of the TC and BC are easily visible, but strongly depend on specific cycle configuration and parameters and cannot be generalized. The present research project will make an attempt to put more structure into this topic.

- Hybrid concept will be technically, economically and environmentally favourable when utilizing biofuels in the bottoming cycle.

- There are different ways to arrange a hybrid combined cycle. Each arrangement must be carefully evaluated in the given application, with the given topping cycle and bottoming cycle parameters and with the given fuels, in order to assess its performance and attractiveness.

- Both gas turbines and internal combustion engines as topping cycles can be combined with biomass or MSW fired boilers as bottoming cycles and the hybrid configuration can provide promising performance, as stated above.

- Internal combustion engines may show lower investment costs and better performance in small-scale HCC configurations.

- Only a few biomass or MSW hybrid combined cycle units are existing in the world today, despite their large potential. Almost all of them are of the repowered type, where an old steam cycle (or a hot water boiler) has been topped by a new gas turbine (or by an internal combustion engine). The presented configurations have been chosen as the most rewarding ones out of various alternatives. They have proved their advantages in commercial operation and support the conclusions made above.

8. SAMMANFATTNING

I Sverige finns redan en stadig tendens för ökad användning av biobränsle. Det är tydligt att nya stora vatten- och kärnkraftverk inte kommer att byggas, vilket innebär att ökad energiåtervinning av skogsresurser, övriga biobränslen och avfall med högre verkningsgrader och lägre kostnader är absolut nödvändigt.

Inhemska biobränsleresurser i Sverige är omfattande och förnyelsebar, men inte ändlös. De måste användas omsorgsfullt och med högre effektivitet för att behålla sin betydelse i framtidens energisystem. Minskad CO_2 emissioner är ännu en faktor för betydelsen av biobränsle i framtiden.

Många energiomvandlingsteknologier finns för biobränsle, antingen genom direkt förbränning i pannor eller via förädling till brännbara gaser eller vätskor för gasturbiner eller kolvmotorer. Några av dessa avancerade teknologier är kommersiella och redo för användning, medan andra fortfarande testas och inte är sannolika ännu.

För stationär kraftproduktion, speciellt i kraftvärmeverk, är den traditionella förbränningen i pannor den enklaste och mest tillförlitligaste metoden.

En enkel väg till högre elverkningsgrader av biobränsleeldade kraftvärmecykler är att kombinera pannan med en högtemperatur kraftcykel (t.ex. gasturbin eller kolvmotor) eldad med fossilt bränsle. Flera olika kombinationer av gasturbin med ångpanna är möjliga och attraktiva. Den traditionella ångcykel (som en lågtemperaturcykel) kan utnyttja avgasvärmet från gasturbinen (högtemperaturcykeln), vilket innebär att varje hybridcykel är en kombicykel med två olika bränsleinsatser. Inga speciella utrustningar krävs, endast enkla komponenter installeras och kostnaderna blir i allmänhet mindre jämfört med två separata anläggningar av samma storlek (en biobränsleeldad ångcykel och en separat kombicykel med gasturbin). Dessutom är den elektriska verkningsgraden högre än den genomsnittliga verkningsgraden för två separata anläggningar som ger samma totala elproduktion, vilket innebär att bränsleutnyttjandet av båda bränslena är högre.

För närvarande håller intresset för hybridcykler på att växa sig starkt i världen, både för storskaliga koleldade kraftverk och småskaliga biobränsleeldade kraftvärmeverk.

Resultaten från litteraturstudien kan sammanfattas enligt nedan:

- Hybridcykelkonceptet är tekniskt och ekonomiskt genomförbart. Det har demonstrerats i alla storlekar med kol, skogsavfall, hushållsavfall, torv eller träpellets som bränsle till pannorna. Den elektriska verkningsgraden förbättras efter integrering med en högtemperaturcykel (gasturbin eller kolvmotor), även om förbättringen är beroende av specifika parametrar och aktuell konfiguration.

- Hybridcykelkonceptet för biobränsle är fördelaktigt ur teknisk, ekonomisk och miljömässig hänseende.

- Olika konfigurationer av högtemperaturcykler integrerade med fastbränsleeldade pannor i hybridcykel kan konstrueras. Varje konfiguration måste värderas inom dess användningsområde, med bestämda parametrar, bränsle och bränsleproportion. De termodynamiska och ekonomiska förmånerna måste värderas jämfört med separata cykler vid samma bränsle och bränsleproportion.

- Kolvmotorer är också lämpliga för hybridcykler. De kan vara billigare och har en högre elverkningsgrad, speciellt i småskaliga hybridcykler. Kolvmotorer har högre NOx emissioner än moderna gasturbiner, men i seriekopplad hybridcykel minskar NOx efter att rökgaserna tillförts pannan.

- Flera biobränsle- eller avfallseldade kraftvärmeverk med hybridcykel finns i Norden. I nästan alla fall har hybridkonfigurationerna valts som det bäste alternativet och en ny gasturbin har tillsatts till den gamla ångcykeln eller hetvattenpannan. Trots att det numera i Norden är omöjligt att generera el med gasturbinbränsle annan än naturgas, har de existerande hybridcyklerna bevisat dess förmåner i kommersiell drift under många år.

LIST OF REFERENCES

Biomass and MSW resources and utilization:

[2.1] **Alakangas, Eija; Flyktman, Martti (2001)**
"Biomass CHP Technologies. Future Cogen",
VTT Energy Reports, Report Nr. 7/2001, Technical Research Centre of Finland, Jyväskylä, Finland, March 2001.

[2.2] **Angus-Hankin, C.; Stokes, B.; Twaddle, A. (1995)**
"The Transportation of Fuel Wood from Forest to Facility",
Biomass & Bioenergy, Vol.9, Issues 1-5, 1995, pp 191-203.

[2.3] **Bain, Richard L.; Overend, Ralph P.; Craig, Kevin R. (1998)**
"Biomass-Fired Power Generation",
Fuel Processing Technology, Vol.54, Issues 1-3, March 1998, pp 1-16.

[2.4] **Benter, M.M.; Gilmour, I.A.; Arnoux, L. (1997)**
"Biomass-Oil Slurry Fuels: An Investigation into their Preparation and Formulation",
Biomass and Bioenergy, Vol.12, Issue 4, 1997, pp 253-261.

[2.5] **Boman, Ulf R.; Turnbull, Jane H. (1996)**
"Integrated Biomass Energy Systems and Emissions of Carbon Dioxide",
Proceedings of ECOS '96, Efficiency, Cost, Optimization, Simulation and Environmental Aspects of Energy Systems, Stockholm, Sweden, June 1996, pp 477-484.

[2.6] **Braber, K. (1995)**
"Anaerobic Digestion of Municipal Solid Waste: A Modern Waste Disposal Option on the Verge of Breakthrough",
Biomass & Bioenergy, Vol.9, Issues 1-5, 1995, pp 365-376.

[2.7] **Brammer J. G. and Bridgwater A. V. (1999)**
"Drying technologies for an integrated gasification bio-energy plant",
Renewable & Sustainable Energy Reviews, Vol.3, Issue 4, December 1999, pp 243-289.

[2.8] **Bridgwater, A.V.; Peacocke, G.V.C. (2000)**
"Fast Pyrolysis Processes for Biomass",
Renewable & Sustainable Energy Reviews, Vol.4, Issue 1, March 2000, pp 1-73.

[2.9] **Börjesson, P.I.I. (1996)**
"Energy Analysis of Biomass Production and Transportation",
Biomass & Bioenergy, Vol.11, Issue 4, 1996, pp 305-318.

[2.10] **Börjesson, Pål; Gustavsson, Leif (1996)**
"Biomass Transportation",
Renewable Energy, Vol.9, Issues 1-4, September 1996, pp 1033-1036.

[2.11] **Chynoweth, David P.; Jerger, Douglas E.; Srivastava, Vipul J. (1985)**
"Biological Gasification of Woody Biomass",
Proceedings of the 20th Intersociety Energy Conversion Engineering Conference, Miami Beach, Florida, August 1985, Vol. 1, pp 573-579.

[2.12] **Crookes, Roy J.; Kiannejad, Fariborz; Nazha, Marouan A.A. (1996)**
"Systematic Assessment of Combustion Characteristics of Biofuels and Emulsions with Water for Use as Diesel Engine Fuels",

Proceedings of ECOS '96, Efficiency, Cost, Optimization, Simulation and Environmental Aspects of Energy Systems, Stockholm, Sweden, June 1996, pp 647-653.

[2.13] **Dahlroth, Björn (2001)**
"Hur mycket avfall finns det?",
Nytt från STOSEB, 2/01, p.13 & p.15.

[2.14] **Dale, B.E.; Leong, C.K.; Pham, T.K.; Esquivel, V.M.; Rios, I.; Latimer, V.M. (1996)**
"Hydrolysis of Lignocellulosics at Low Enzyme Levels: Application of the AFEX Process",
Bioresource Technology, Vol.56, Issue 1, April 1996, pp 111-116.

[2.15] **Demirbas, A. (2000)**
"Mechanisms of Liquefaction and Pyrolysis Reactions of Biomass",
Energy Conversion and Management, Vol.41, Issue 6, April 2000, pp 633-646.

[2.16] **Dong, Yuanji; Steinberg, Meyer (1997)**
"HYNOL – an Economical Process for Methanol Production from Biomass and Natural Gas with Reduced CO_2 Emission",
International Journal of Hydrogen Energy, Vol.22, Issue 10-11, October 1997, pp 971-977.

[2.17] **Elliott, Douglas C.; Baker, Eddie G. (1985)**
"Hydrodeoxygenation of Wood-Derived Liquids to Produce Hydrocarbon Fuels",
Proceedings of the 20th Intersociety Energy Conversion Engineering Conference, Miami Beach, Florida, August 1985, Vol. 1, pp 586-592.

[2.18] **Forsberg, Göran (2000)**
"Biomass Energy Transport; Analysis of Bioenergy Transport Chains Using Life Cycle Inventory Method",
Biomass & Bioenergy, Vol.19, Issue 1, July 2000, pp 17-30.

[2.19] **Goossens, M.A. (1996)**
"Landfill Gas Power Plants",
Renewable Energy, Vol.9, Issues 1-4, September 1996, pp 1015-1018.

[2.20] **Gregg, D.; Saddler, J.N. (1995)**
"Bioconversion of Lignocellulosic Residue to Ethanol: Process Flowsheet Development",
Biomass & Bioenergy, Vol.9, Issues 1-5, 1995, pp 287-302.

[2.21] **Gustavsson, Leif; Johansson, Bengt (1994)**
"Cogeneration: One Way to Use Biomass Efficiently",
Heat Recovery Systems & CHP, Vol.14, Nr.2, March 1994, pp 117-127.

[2.22] **Gustavsson, Leif; Börjesson, Pål; Johansson, Bengt; Svenningsson, Per (1995)**
"Reducing CO2 Emissions by Substituting Biomass for Fossil Fuels",
Energy, Vol.20, Nr.11, November 1995, pp 1097-1113.

[2.23] **Harder M.K. and Freeman L.A. (1996)**
"A Study of an Integrated Land-Fill and Coppice Power Station",
Renewable Energy, Vol.9, Issues 1-4, September 1996, pp 989-992.

[2.24] **Hill, Jane P. (1991)**
"The Economics of Electricity Generation Using Landfill Methane",
Proceedings of the 5th International Symposium ASME COGEN-TURBO, Budapest, Hungary, September 1991, pp 319-324.

[2.25] **Hillring, Bengt; Ling, Erik; Blad, Bengt (2001)**
"The Potential and Utilisation of Biomass",

included in the book "Building Sustainable Energy Systems", Published by the Swedish National Energy Administration and Svensk Byggtjänst, Edited by Semida Silveira, ISBN: 91-7332-961-4, Stockholm 2001.

[2.26] **Hughes, Evan (2000)**
"Biomass Cofiring: Economics, Policy and Opportunities",
Biomass & Bioenergy, Vol.19, Issue 6, December 2000, pp 457-465.

[2.27] **Hulkkonen, Seppo; Äijälä, Martti; Holappa, Jyrki (1993)**
"Integration of a Fuel Dryer to a Gas Turbine Process",
ASME Cogen-Turbo, IGTI Vol.8, Proceedings of the 7th Congress & Exposition on Gas Turbines in Cogeneration and Utility Industrial and Independent Power Generation, Bournemouth, England, September 1993, pp 61-67.

[2.28] **Hulkkonen, Seppo; Parvio, Erkki; Raiko, Markku (1995)**
"An Advanced Fuel Drying Technology for Fluidized Bed Boilers",
Proceedings of the 13th International Conference on Fluidized Bed Combustion, Orlando, FL, USA, May 1995, Vol.1, pp 399-403.

[2.29] **Jenkins B.M.; Baxter L.L.; Miles Jr T.R.; Miles T.R. (1998)**
"Combustion Properties of Biomass",
Fuel Processing Technology, Vol.54, Issues 1-3, March 1998, pp 17-46.

[2.30] **Jirjis, R. (1995)**
"Storage and Drying of Wood Fuel",
Biomass and Bioenergy, Vol.9, Issues 1-5, 1995, pp 181-190.

[2.31] **Johansson, L.; Westerlund, L. (2000)**
"Energy Efficient Bio Fuel Drying with an Open Absorption System. Parameter Study in order to Reduce Investment Costs",
Applied Energy, Vol.67, Issue 3, November 2000, pp 231-244.

[2.32] **Juste, G. López; Monfort, J.J. Salvá (2000)**
"Preliminary Test on Combustion of Wood Derived Fast Pyrolysis Oils in a Gas Turbine Combustor",
Biomass & Bioenergy, Vol.19, Issue 2, August 2000, pp 119-128.

[2.33] **Knight, J.A.; Gorton, C.W. (1985)**
"Oil Production via Entrained Flow Pyrolysis of Biomass",
Proceedings of the 20th Intersociety Energy Conversion Engineering Conference, Miami Beach, Florida, August 1985, Vol. 1, pp 569-572.

[2.34] **Lehtikangas, Päivi (1999)**
"Lagringshandbok för Trädbränslen",
2:a upplagan, Inst. för skogshushållning, Sveriges Lantbruksuniversitet, Uppsala, 1999, ISBN 91-576-5564-2.

[2.35] **Liang, T.; Khan, M.A.; Meng, Q. (1996)**
"Spatial and Temporal Effects in Drying Biomass for Energy",
Biomass and Bioenergy, Vol.10, Issues 5-6, 1996, pp 353-360.

[2.36] **Linz, David G. (1988)**
"Natural Gas Usage in Municipal Waste Disposal",
Proceedings of the Energy Technology Conference, Vol.15, Washington D.C., February 1988, pp 757-768.

[2.37] **Linz, David G.; Randall Seeker, WM. (1989)**
"Advanced Natural Gas Firing for Pollutant Control from Municipal Solid Waste Combustion Systems",
Proceedings of the American Power Conference, Vol. 51, Chicago, IL, April 1989, pp 865-869.

[2.38] **Lisk, Donald J. (1988)**
"Environmental Implications of Incineration of Municipal Solid Waste and Ash Disposal",
The Science of the Total Environment, Vol.74, Nr.1, August 1988, pp 39-66.

[2.39] **Maly, P.M.; Seeker, W.R.; Klosky, M.K. (1998)**
"Evaluation of Air Toxics Emissions from RDF Slurry Combustion",
Environmental Engineering Science, Vol.15, Issue 2, 1998, pp 107-116.

[2.40] **Mann, Margaret K.; Spath, Pamela L. (1999)**
"Life Cycle Comparison of Electricity from Biomass and Coal",
Proceedings of ACEEE Summer Study on Energy Efficiency in Industry, Washington DC, USA, 1999, pp 559-569.

[2.41] **Minkova, V.; Marinov, S.P.; Zanzi, R.; Björnbom, E.; Budinova T.; Stefanova M.; Lakov L. (2000)**
"Thermochemical Treatment of Biomass in a Flow of Steam or in a Mixture of Steam and Carbon Dioxide",
Fuel Processing Technology, Vol.62, Issue 1, January 2000, pp 45-52.

[2.42] **Miranda, Marie Lynn; Hale, Brack (2001)**
"Protecting the Forest from the Trees: The Social Costs of Energy Production in Sweden",
Energy, Vol.26, Issue 9, September 2001, pp 869-889.

[2.43] **Moreira, Jose R.; Goldemberg, Jose (1999)**
"The alcohol program",
Energy Policy, Vol.27, Issue 4, April 1999, pp 229-245.

[2.44] **Natarajan, V.P.; Suppes, G.J. (1997)**
"Rheological Studies on a Slurry Biofuel to Aid in Evaluating its Suitability as a Fuel",
Fuel, Vol.76, Issue 14-15, November 1997, pp 1527-1535.

[2.45] **Nelson, John (1994)**
"The Environmental and Economical Advantages of Fluidized Bed Combustion for Biomass and Wastes",
Renewable Energy, Vol.5, Nr.5-8, August 1994, pp 824-828.

[2.46] **Ostlie, L. David (1989)**
"The Whole Tree Burner: A New Technology in Power Generation",
Proceedings of the American Power Conference, Vol. 51, Chicago, IL, April 1989, pp 876-879.

[2.47] **Posarac, Dusko; Watkinson, Paul A. (2000)**
"Mixing of a Lignin-Based Slurry Fuel",
The Canadian Journal of Chemical Engineering, Vol.78, Nr.1, February 2000, pp 265-270.

[2.48] **Rezzoug, Sid-Ahmed; Capart, Richard (1996)**
"Solvolysis and Hydrotreatment of Wood to Provide Fuel",
Biomass & Bioenergy, Vol.11, Issue 4, 1996, pp 343-352.

[2.49] **Rocha, J. Dilcio; Luengo, Carlos A.; Snape, Colin E. (1999)**
"The Scope for Generating Bio-oils with Relatively Low Oxygen Contents via Hydropyrolysis",
Organic Geochemistry, Vol.30, Issue 12, December 1999, pp 1527-1534.

[2.50] **Ruth, Lawrence A. (1998)**
"Energy from Municipal Solid Waste: a Comparison with Coal Combustion Technology",

Progress in Energy and Combustion Science, Vol.24, Issue 6, 1998, pp 545-564.

[2.51] **Svensk Energiförsörjning**, 662 31 Åmål, Sweden,
"ENERGIFAKTA".

[2.52] **Svenskt Gastekniskt Center AB**, 205 09 Malmö, Sweden,
"Energigasernas Miljöeffekter. Faktahandbok",
Rapport SGC 075 – tredje utgåvan, November 2000.

[2.53] **Swedish National Energy Administration**, 631 04 Eskilstuna, Sweden,
"Energy in Sweden 2000" Annual Report, December 2000, and
"Energy in Sweden 2001" Annual Report, November 2001.

[2.54] **Szczodrak, Janusz; Fiedurek, Jan (1996)**
"Technology for Conversion of Lignocellulosic Biomass to Ethanol",
Biomass & Bioenergy, Vol.10, Issue 5/6, 1996, pp 367-375.

[2.55] **Tafdrup, S. (1995)**
"Viable Energy Production and Waste Recycling from Anaerobic Digestion of Manure and other Biomass Materials",
Biomass & Bioenergy, Vol.9, Issues 1-5, 1995, pp 303-314.

[2.56] **Thring, Ronald W.; Katikaneni, Sai P.R.; Bakhshi, Narendra N. (2000)**
"The Production of Gasoline Range Hydrocarbons from Alcell Lignin Using HZSM-5 Catalyst",
Fuel Processing Technology, Vol.62, Issue 1, January 2000, pp 17-30.

[2.57] **Tillman, D.A. (2000)**
"Biomass Cofiring: the Technology, the Experience, the Combustion Consequences",
Biomass & Bioenergy, Vol.19, Issue 6, December 2000, pp 365-384.

[2.58] **Wahlund, B; Yan, J.; Westermark, M. (2001)**
"Study of a Total Energy System of Fuel Upgrading by Drying Biomass Feedstock for Combined Heat and Power Production: A Case Study of Skellefteå Bioenergy Combine",
Proceedings of the 12th World Clean Air & Environment Congress and Exhibition, Seoul, South Korea, August 2001.

[2.59] **White, Marshall S.; DeLuca, Peter A. (1978)**
"Bulk Storage Effects on the Fuel Potential of Sawmill Residues",
Forest Products Journal, Vol.28, Nr.11, November 1978, pp 24-29.

[2.60] **www.agrobransle.se**
Agrobränsle AB, 701 17 Örebro, Sweden.

[2.61] **www.folkecenter.dk**
Folkecenter for Vedvarende Energi, DK-7760 Hurup Thy, Denmark.

[2.62] **Wyman, Charles E. (1986)**
"Microbiological Conversion of Renewable Resources into Liquid Fuels",
Proceedings of the 21st Intersociety Energy Conversion Engineering Conference, San Diego, California, August 1986, Vol. 1, pp 217-222.

[2.63] **Älvesand, Lena; Hörnfeldt, Johan; Eriksson, Ola; Andersson, Fredrik; Liedén, Stefan (1997)**
"Storskalig Biobränsleeldning. Förstudie angående möjligheterna att bygga en anläggning för biobränsleeldad energiförsörjning i storstad",
Projektarbete, Inst. för Kraft- och Värmeteknologi, Kungl. Tekniska Högskolan, Stockholm, Sweden, 1997.

Repowering and Coal Hybrid Cycles:

[3.1] **Bauer, G.; Lovis, M.; Märker, W. (1993)**
"Das Verbundkraftwerk – eine neue Variante des kombinierten Kraftwerk",
VGB Kraftwerkstechnik, Vol. 73, Heft 2, 1993, pp 120-124.

[3.2] **Bettocchi, R.; Cantore, G.; Galletti, A.; Gusso, R.; Negri di Montenegro, G. (1989)**
"Comparative Analysis of Gas-Steam Combined Cycles With and Without Supplementary Firing",
Quaderni Pignone, Nr.47, June 1989, pp 5-15.

[3.3] **Boissenin, Y. (1989)**
"Combined Cycle Power Plants: A Practical Guide to the Right Choice",
Proceedings of the 3rd ASME COGEN-TURBO International Symposium, Nice, France, August-September 1989, pp 333-345.

[3.4] **Brander, J.A.; Chase, D.L. (1991)**
"Repowering Application Considerations",
ASME Paper 91-GT-229, Proceedings of the International Gas Turbine and Aeroengine Congress and Exposition, Orlando, Florida USA, June 1991.

[3.5] **Brückner, Hermann; Bergmann, Dietmar; Termuehlen, Heinz (1992)**
"Various Concepts for Topping Steam Plants with Gas Turbines",
Proceedings of the 54th Annual Meeting of the American Power Conference, Chicago, Illinois, April 1992, pp 569-582.

[3.6] **Brückner, H.; Emsperger, W. (1989)**
"Retrofitting Fossil-Fired Power Plants with Gas Turbines as a Means of Increasing Output and Efficiency",
Proceedings of the International Centre for Heat and Mass Transfer, First International Forum on Mathematical Modeling and Computer Simulation of Processes in Energy Systems, Sarajevo, Yugoslavia, April 1989, pp 801-813.

[3.7] **Chase, David L.; Kovacik, John M.; Stoll, Harry G. (1992)**
"Economic Strategies for Repowering Steam Power Plants",
International Power Generation Exhibition & Conference POWER-GEN '92, Orlando, Florida, November 1992, Vol.11&12, pp 166-190.

[3.8] **Davidse, J.; Roukema, J. (1984)**
"Application of Gas Turbine Generators for Preheating Combustion Air with Conventional Boilers or Furnaces",
ASME Paper, Presented at the International Gas Turbine Conference and Exhibition, Amsterdam, The Netherlands, 1984.

[3.9] **Dechamps, P.J.; Mathieu, P. (1991)**
"Repowering Options for the Electricity Generation Mix in Belgium",
Proceedings of the 5th International Symposium ASME Cogen-Turbo, Budapest, Hungary, September 1991, pp 43-50.

[3.10] **Dechamps. P.J.; Pirard, N.; Mathieu, P. (1994)**
"Part Load Operation of Combined Cycle Plants With and Without Supplementary Firing",
ASME Cogen-Turbo, Proceedings of the 8th Congress & Exposition on Gas Turbines in Cogeneration and Utility, Industrial and Independent Power Generation, Portland, Oregon, October 1994, pp 371-380.

[3.11] **Evans, R.H.; McMullan, J.T.; Williams, B.C.; Witthohn, A.; Cerri, G.; Chapman, K. (1996)**
"Improving Coal-Fired Power Generation by Integrating PFBC and Gas-Fired Combined Cycles in a Hybrid Process",
International Journal of Energy Research, Vol.20, Nr.2, February 1996, pp 107-123.

[3.12] **Finckh, H.H.; Pfost, H. (1991)**
"Development Potential of Combined-Cycle (GUD) Power Plants With and Without Supplementary Firing",
ASME Paper 91-GT-227, Proceedings of the International Gas Turbine and Aeroengine Congress and Exposition, Orlando, Florida USA, June 1991.

[3.13] **Gallardy, P.B.; Boulay, R.B.; McLaughlin, B.R. (1992)**
"Combined Cycle Repowering Options",
Proceedings of the American Power Conference, Vol. 54, Chicago, IL, April 1992, pp 560-563.

[3.14] **Galletti, Alessandro (1990)**
"Repowering Power Plants and Cogeneration Stations with Gas Turbines",
Quaderni Pignone, Nr.49, December 1990, pp 29-34.

[3.15] **Gang, Fang; Ruixian, Cai; Rumou, Lin (1990)**
"Off-Design Performance of Fully-Fired Combined Cycle Cogeneration",
ASME Paper 90-GT-370, Proceedings of the International Gas Turbine and Aeroengine Congress and Exposition, Brussels, Belgium, June 1990.

[3.16] **Gelbar, D.E.; Akel, H.R.; Liljedahl, G.N. (1988)**
"Retrofit Utility Power Plant to Combined Cycle Operation",
Proceedings of the American Power Conference, Vol. 50, Chicago, IL, April 1988, pp 57-64.

[3.17] **Gerstin, R.A. (1975)**
"A Technical and Economic Overview of the Benefits of Repowering",
ASME Paper 75-GT-16, March 1975.

[3.18] **Goebel, Konrad (1975)**
"European Approach to Combined Cycles Brings Early Energy Savings",
Energy International, Vol.12, Nr.3, March 1975, pp 20-22.

[3.19] **Hansen, Soeren; Jensen, Svend Erik (2000)**
"Hybrid Units at Map Ta Phut are first of a kind",
Modern Power Systems, Vol.20, Nr.8, August 2000, pp 33-39.

[3.20] **Hebel, G.; Kotschenreuther, H. (1990)**
"Environmental Protection by Improvement of Efficiency",
WFEO/FMOI Energy Conference, Madrid, Spain, October 1990.

[3.21] **Heyen, G.; Kalitventzeff, B (1999)**
"A Comparison of Advanced Thermal Cycles Suitable for Upgrading Existing Power Plant",
Applied Thermal Engineering, Vol.19, Issue 3, March 1999, pp 227-237.

[3.22] **Kakaras, E.; Vigellis, G.; Papageorgiou, N. (1993)**
"Gas Turbine Topping for Existing Brown Coal Fired Power Plants",
Proceedings of the 7th ASME COGEN-TURBO Congress & Exposition, Bournemouth, UK, September 1993, pp 461-466.

[3.23] **Kamminga, P. (1987)**
"Boosting Steam Plant Thermal Efficiency and Power Output by Adding Gas Turbines",

Proceedings of the ASME COGEN-TURBO International Symposium, Montreaux, France, 1987, pp 363-368.

[3.24] **Linnemeijer, M.J.J.; Van Buijtenen, J.P.; Van Loon, A.U. (1987)**
"Boosting Steam Plant Thermal Efficiency and Power Output Through the Addition of Gas Turbines",
ASME Paper 87-GT-4, Presented at the International Gas Turbine and Aeroengine Conference and Exhibition, Anaheim, California, May-June 1987.

[3.25] **Melli, R.; Naso, V.; Sciubba, E. (1992)**
"Modular Repowering of Power Plants with Nominal Ratings Lower than 180 MW: A Rational Design Approach and its Application to the Italian Utility System",
Journal of Energy Resources Technology, Transactions of the ASME, Vol.116, Nr.3, September 1994, pp 201-210, Presented at the 15th ASME Annual Energy-Sources Technology Conference and Exhibition, Houston, Texas, January 1992.

[3.26] **Moore, Taylor (1995)**
"Repowering as a Competitive Strategy",
EPRI Journal, Vol.20, Nr.5, September/October 1995, pp 6-13.

[3.27] **Morikawa, Kengo; Fukumoto, Fumio; Kawauchi, Akihiro (1993)**
"Renewal of Existing Fossil-Fired Plants Incorporating Fully-Fired Combined Cycle",
Hitachi Review, Vol. 42, Nr.1, February 1993, pp 17-24.

[3.28] **Myers, G.A.; Pavel, J. (1989)**
"Combined Cycle Repowering – A Viable Solution to Utility Power Requirements",
Proceedings of the American Power Conference, Vol. 51, Chicago, IL, April 1989, pp 253-260.

[3.29] **Nielsen, Charles (1991)**
"Natural Gas Repowering of Coal-Fired Plants",
Proceedings of the 5th International Symposium ASME COGEN-TURBO, Budapest, Hungary, September 1991, pp 51-59.

[3.30] **Pfost, H.; Rukes, B.; Termuehlen, H. (1997)**
"Repowering with Gas Turbines Utilizing their Exhaust Energy for Feedwater Heating and/or Reheat Steam Generation",
Proceedings of the International Joint Power Generation Conference, Denver, Colorado, November 1997, Vol.2, pp 35-48.

[3.31] **Ploumen, P.J.; Veenema, J.J. (1995)**
"Dutch Experience with Hot Windbox Repowering",
Proceedings of the American Power Conference, Vol. 57, Chicago, IL, April 1995, pp 980-985.

[3.32] **Schobesberger, Peter; Gilli, Paul (1995)**
"The Retrofit of Heat Recovery Systems for Power Increase in Fossil Power Plants",
ASME Paper 95-CTP-62, Proceedings of the ASME Cogen-Turbo Power Conference, Viena, Austria, August 1995.

[3.33] **Seery, D.J.; Sangiovanni, J.J.; Cogswell, F.J.; Bird, C.E. (1997)**
"Engineering Development of a Coal-Fired High Performance Power Generating System",
Proceedings of the International Joint Power Generation Conference, Denver, Colorado, November 1997, Vol.2, pp 49-54.

[3.34] **Smith, Douglas J. (1995)**
"Combined-Cycle Gas Turbines: The Technology of Choice for New Power Plants",
Power Engineering International, Vol.3, Nr.3, May/June 1995, pp 21-25.

[3.35] **Stenzel, William C.; Sopocy, Dale M.; Pace, Stanley E. (1997)**
"Repowering Existing Coal Fired Plants",
Proceedings of the International Joint Power Generation Conference, Denver, Colorado, November 1997, Vol.2, pp 595-604.

[3.36] **Sullivan, T.M.; Briesch, M.S. (1994)**
"Repowering... A Ready Source of New Capacity",
Energy Engineering, Vol.91, Nr.2, 1994, pp 61-80.

[3.37] **Taft, M. (1991)**
"A Comprehensive Classification of Combined Cycle and Cogeneration Plant Part 2: Introducing the Integrated Steam Cycle in Practice",
Proceedings of the Institution of Mechanical Engineers, Part A, Vol.205, Nr.3, 1991, pp 145-159.

[3.38] **Takizawa, M.; Ohara, H.; Morikawa, K.; Kawauchi, A.; Ando, N. (1993)**
"Repowering of Thermal Power Plants as Fully-Fired Combined Cycle Generating Plants",
Proceedings of the 7th ASME COGEN-TURBO Congress & Exposition, Bournemouth, UK, September 1993, pp 473-485.

[3.39] **Taylor, W.G. (1983)**
"Combined Cycles and Alternative Fuels for Gas Turbines",
Quaderni Pignone, Nr.35, 1983, pp 55-65.

[3.40] **Termuehlen, Heinz (1986)**
"Gas and Coal Fired Combined Cycle Plants",
Proceedings of the American Power Conference, Chicago, Illinois, 1986, pp 210-220.

[3.41] **Termuehlen, Heinz; Kraemer, Hans; Achenbach, John A. (1992)**
"Repowering Existing Power Stations with Heavy-Duty Gas Turbines – an Economical Approach",
International Power Generation Exhibition & Conference POWER-GEN '92, Orlando, Florida, November 1992, Vol.11&12, pp 235-244.

[3.42] **Van Der Linden, S.; Fetescu, M.; Burkett, J. (1997)**
"Two Novel Approaches to Repowering",
Proceedings of the American Power Conference, Vol. 59, Part 2, Chicago, IL, USA, 1997, pp 1211-1216.

[3.43] **Walters, Arden B. (1995)**
"Power Plant Topping Cycle Repowering",
Journal of the Association of Energy Engineering, Vol.92, Nr.5, 1995, pp 49-71.

[3.44] **Walters, Arden B. (1995)**
"The Role of Competitive Repowering in Integrated Resource Planning",
Proceedings of the American Power Conference, Vol. 57, Chicago, IL, April 1995, pp 977-979.

[3.45] **Wen, Harvey; Gopalarathinam, R. (1992)**
"Repowering Options for Steam Power Plants",
Proceedings of the American Power Conference, Vol. 54, Chicago, IL, April 1992, pp 583-588.

Biomass Hybrid Cycles:

[4.1] **Bergman, Jackie (1992)**
"HYDRA – projektet",
Report Nr. UB 1992/43, Vattenfall Utveckling AB, Bioenergi, 810 70 Älvkarleby, Sweden, October 1992.

[4.2] **Egard, M.; Steinwall, P.; Håkansson, J.; Assadi, M.; Torisson, T. (2000)**
"Hybridkraftverk för elproduktion",
Report Nr. TERM- 00/7, library listing: STEM-TERM--00-7, Swedish National Energy Administration, Project P10660-1, Eskilstuna, Sweden, August 2000.

[4.3] **Kassem, Nabil; Harvey, Simon (2001)**
"Evaluation of Emerging Hybrid Systems for Heat and Power Generation Using Biomass and Natural Gas, Subject to Technical and Economic Uncertainties",
Arbetsnotat Nr.21, ISSN: 1403-8307, Energy Systems Programme, IKP, Linköping Institute of Technology, Linköping, Sweden, November 2001.

[4.4] **Kunert, Markus (1995)**
"Investigation on Different Types of Combined Cycles with Natural Gas and Biomass Fuel in View on Efficiency and CO2 Emissions",
M.Sc. Thesis, report Nr.453, Division of Heat & Power Technology, Royal Institute of Technology, Stockholm, Sweden, February 1995.

[4.5] **Skoglund, Per (1996)**
"Naturgas för Högre Prestanda vid Biobränsleeldad Kraftvärme",
Mekanisten, Nr.2, 1996, pp 63-66.

[4.6] **Spath, Pamela L.; Overend, Ralph P. (1996)**
"Biomass Utilization in Cofiring Scenarios with a Gas Turbine Integrated Combined Cycle System",
ASME Paper, Environmental Control Division, Proceedings of the 1996 International Joint Power Generation Conference, Part 1, Houston, Texas, October 1996, pp 345-354.

[4.7] **Westermark, Mats (1992)**
"Hybridkraftstationer- teknik, ekonomi och potential",
Report Nr. VU-V 91:32, Vattenfall Utveckling AB, Utveckling och Miljö, Värmeteknik, 810 70 Älvkarleby, Sweden, May 1992.

[4.8] **Wyszkowski, Roberto (2001)**
"Performance of Hybrid CHP Systems in Process Industries",
M.Sc. Thesis, Department of Heat & Power Technology, Chalmers University of Technology, Göteborg, Sweden, February 2001.

MSW Hybrid Cycles:

[4.9] **Bartlett, Michael; Holmgren, Kristina (2001)**
"Waste Incineration in Swedish Municipal Energy Systems – an investigation of the system consequences of waste quantities in Linköping and the conditions for conventional and evaporative hybrid cycle operation"
Arbetsnotat Nr.19, ISSN: 1403-8307, Energy Systems Programme, IKP, Linköping Institute of Technology, Linköping, Sweden, April 2001.

[4.10] **Consonni, Stefano (2000)**
"Combined Cycles for High Performance, Low Cost, Low Environmental Impact Waste-to-Energy Systems",
ASME Paper 2000-GT-24, Proceedings of ASME TURBOEXPO, Munich, Germany, May 2000.

[4.11] **Eber, Stephen M.; Kettler, David J.; Worsley, Carolyn R.; Fedele, Gina (1989)**
"Gas Fired Superheaters in a Resource Recovery Installation",
Proceedings of the American Power Conference, Vol. 51, Chicago, IL, April 1989, pp 858-864.

[4.12] **Holmgren, Robert (1998)**
"Hybridkraftverk för avfallseldning",
M.Sc. Thesis, report Nr.1998:228 CIV, ISSN: 1402-1617, ISRN: LTU-EX--98/228--SE, Division of Energy Technology, Luleå University of Technology, Luleå, Sweden, August 1998.

[4.13] **Ito, K.; Yokoyama, R.; Shimoda, M. (1996)**
"Optimal Planning of a Super Waste Incineration Cogeneration Plant",
ASME Paper 96-GT-384, Proceedings of the International Gas Turbine and Aeroengine Congress and Exposition, Birmingham, UK, June 1996.

[4.14] **Korobitsyn, M. A.; Jellema, P.; Hirs, G. G. (1999)**
"Possibilities for Gas Turbine and Waste Incinerator Integration",
Energy, Vol.24, Issue 9, September 1999, pp 783-793.

[4.14a] **Korobitsyn, M. A. (2000)**
"Enhancing Direct-Fired Power Plants Performance by use of Gas Turbine Technology",
Journal of Propulsion and Power, *AIAA, Vol.16, Issue 4, July 2000, pp 568-571.*

[4.15] **Lowry, William L.; Martin, Charles A. (1990)**
"Economics of Combined Cycle Waste-to-Energy",
Proceedings of the International Joint Power Generation Conference, Boston, MA, USA, October 1990, pp 31-37.

[4.16] **Otoma, Suehiro; Mori, Yasufumi; Terazono, Atsushi; Aso, Tomonori; Sameshima, Ryoji (1997)**
"Estimation of Energy Recovery and Reduction of CO2 Emissions in Municipal Solid Waste Power Generation",
Resources Conservation and Recycling, Vol.20, Issue 2, June 1997, pp 95-117.

[4.17] **Sue, M. (1996)**
"Performance Characteristics of Waste-to-Energy System Utilizing Steam-Injected Gas Turbine",
Transactions of the Japan Society of Mechanical Engineers, Part B, Vol.62, Nr.597, May 1996, pp 2013-2020.

[4.18] **Terasawa, Hideaki; Ogura, Masao (1993)**
"System to Rationalize Energy Utilization at Waste Incineration Plants",
Proceedings of JSME – ASME International Conference on Power Engineering, Tokyo, September 1993, Vol.2, pp 591-594.

[4.19] **Wiekmeijer, Ir. Ted (1990)**
"Improvements in Incinerators by Means of Gas Turbine Based Cogen Systems",
ASME Paper 90-GT-180, Presented at the International Aeroengine Congress and Exposition, Brussels, Belgium, June 1990.

Internal Combustion Engine Combined Cycles

[5.1] **Anonymous (1974)**
"Exhaust from engines under test supplies heating steam at Detroit Diesel",
Power, Vol.118, Nr.7, July 1974, p. 47.

[5.2] **Anonymous (1998)**
"Heat Stroke – Diesel Combined Cycle",
Power Plant Technology Operations & Maintenance, October/November 1998, pp 10-15.

[5.3] **Anonymous (1983)**
"New Diesel – Engine – Powered Total Energy System",
Energy Digest, Vol.12, Nr.4, 1983, pp 10-13.

[5.4] **Aly, S.E. (1988)**
"Diesel Engine Waste-Heat Power Cycle"
Applied Energy, Vol.29, Nr.3, 1988, pp 179-189.

[5.5] **Andersson, Jon; Tomic, Kristjan (1996)**
"Konvertering av Marvikens kraftstation till kombikraftverk med tvåtakts dieselmotorer",
M.Sc. Thesis, ISSN: 0282-1990, ISRN: LUTMDN/TMVK--5270--SE, Dept. of Heat & Power Technology, Lund Institute of Technology, Lund, Sweden, February 1996.

[5.6] **Bailey, M. Murray (1985)**
"Comparative Evaluation of Three Alternative Power Cycles for Waste Heat Recovery from the Exhaust of Adiabatic Diesel Engines",
NASA Technical Memorandum, Report Nr. DOE/NASA/50194-43, NASA TM-86953, July 1985.

[5.7] **Björnsjö, Ann (1996)**
"Kombikraftverk baserat på två dieselmotorer och ett ångkondenskraftverk",
M.Sc. Thesis, report Nr. 483, Division of Heat & Power Technology, Royal Institute of Technology, Stockholm, Sweden, January 1996.

[5.8] **Broberger, Helena (1995)**
"Komplettering av Marvikens kraftstation med hjälp av dieselmotorer",
M.Sc. Thesis, report Nr. 468, Division of Heat & Power Technology, Royal Institute of Technology, Stockholm, Sweden, August 1995.

[5.9] **Cemenska, R.A. (1988)**
"A Rankine Bottoming Cycle System for Heavy-Duty Applications",
Journal of Engineering for Gas Turbines and Power, Transactions of the ASME, Vol.110, Nr.3, July 1988, pp 356-360.

[5.10] **Ferrari, Joseph R.; Shelor, F. Mack; Bianco, Robert (1998)**
"Emissions and Efficiency Characteristics of a 79 MW Diesel Combined Technology Combined Cycle",
ASME Paper 98-ICE-111, Proceedings of the Spring Technical Conference of the ASME Internal Combustion Engine Division, Part 3, Fort Lauderdale, FL, USA, April 1998, pp 91-102.

[5.11] **Finn, Dennis P. (1995)**
"Innovative Combustion of Fuels in Combined Cycle Applications",
Proceedings of the Industrial Power Conference, April 1995, Orlando, FL, USA, pp 23-29.

[5.12] **Hägglund, Thomas; Ahnger, Anders (1997)**
"The Advanced Diesel Combined Cycle Plant",

Wärtsilä Diesel Power News, Issue 1, May 1997, pp 23-26.

[5.13] **Håkansson, Jesper (1998)**
"Hybridkraftverk med gasmotor och fastbränsleeldad panna",
M.Sc. Thesis, ISSN: 0282-1990, ISRN: LUTMDN/TMVK--5302--SE, Dept. of Heat & Power Technology, Lund Institute of Technology, Lund, Sweden, May 1998.

[5.14] **Jonsson, Maria; Yan, Jinyue (2000)**
"Exergy and Pinch Analysis of Diesel Engine Bottoming Cycles with Ammonia-Water Mixtures as Working Fluid",
International Journal of Applied Thermodynamics, Vol.3, Nr.2, June 2000, pp 57-71.

[5.15] **Kuehn, Steven E. (1995)**
"Diesels in Combined Cycle",
Power Engineering, Vol.99, Nr.3, March 1995, pp 35-37.

[5.16] **Kuehn, Steven (1995)**
"Multi-talented Diesels Flexibly Efficient in Combined Cycle",
Power Engineering International, Vol.3, Nr.3, May/June 1995, pp 26-28.

[5.17] **Leising, C.J.; Purohit, G.P.; DeGrey, S.P.; Finegold, J.G. (1978)**
"Waste Heat Recovery in Truck Engines",
SAE Preprints, Paper Nr. 780686, August 1978, pp 2639-2648.

[5.18] **Lindberg, Anders (1998)**
"Indo Rama – Industrial Cogeneration in the heat of India",
Energy News, Wärtsilä NSD Corp., Issue 3, June 1998, pp 16-17.

[5.19] **MacDonald, H.B.; Nash, F. (1987)**
"Waste Heat Recovery to Produce Additional Power at Lerwick 'B'",
Presented at the "Internal Combustion and Steam Plant in Combination" Seminar at The Institution of Mechanical Engineers, London UK, November 1987, pp 83-94.

[5.20] **Makansi, Jason (1994)**
"Repowering: Options Proliferate for Managing Generation Assets",
Power, Vol.138, Nr.6, June 1994, pp 33-40.

[5.21] **Nash, F. (1992)**
"The Development and First Commercial Application of an Innovative Diesel Engine Based Cogeneration System",
Proceedings of the Institution of Mechanical Engineers, Part A, Vol.206, Nr.3, 1992, pp 197-207.

[5.22] **Niemi, Seppo (1997)**
"Survey of Modern Power Plants Driven by Diesel and Gas Engines",
VTT Research Notes, Report Nr.1860, Technical Research Centre of Finland, Espoo, September 1997.

[5.23] **Oomori, Hideyo; Ogino, Shigeru (1993)**
"Waste Heat Recovery of Passenger Car Using a Combination of Rankine Bottoming Cycle and Evaporative Engine Cooling System",
SAE Special Publications, Paper Nr. 930880, International Congress and Exposition, Detroit, MI, USA, March 1993, pp 159-164.

[5.24] **Price, M.E.; Chilton, M.D. (1987)**
"The Fort Dunlop Combined Heat and Power Plant",
Presented at the "Internal Combustion and Steam Plant in Combination" Seminar at The Institution of Mechanical Engineers, London UK, November 1987, pp 43-53.

[5.25] **Shelor, F. Mack (1995)**
"Repower with Diesel Engines to Bolster Feedwater Heating",
Power, Vol.139, Nr.6, June 1995, pp 96-98.

[5.26] **Shelor, F. Mack (1997)**
"Repowering with Diesels: they're not just for black start anymore"
Power Engineering, Vol.101, Nr.2, February 1997, pp 37-40.

[5.27] **Silvonen, Aulis (1996)**
"Hot Combustion Improves Diesel Combined Cycle",
Modern Power Systems, Vol.16, Nr.12, December 1996, pp 61-63.

[5.28] **Sinatov, St. (1998)**
"Turbocharging Helps to Diesel Engine in Cogeneration and Combined Cycles",
Proceedings of the 22nd CIMAC International Congress on Combustion Engines, Copenhagen, Denmark, May 1998, pp 1607-1612.

[5.29] **Steiger, A. (1984)**
"Waste Heat Recovery in Large Diesel Engine with Bottoming Cycle"
VDI Berichte, Proceedings of the International VDI Seminar, Zurich, Switzerland, 1984, pp 483-500.

[5.30] **Stenhede, Thomas (1997)**
"Optimized Heat Production in a Cogeneration Application",
Wärtsilä Diesel Power News, Issue 1, May 1997, pp 10-12.

[5.31] **Sulle, C. (1984)**
"Water Steam Recovery System in a Large E.D.F. Diesel Plant"
VDI Berichte, Proceedings of the International VDI Seminar, Zurich, Switzerland, 1984, pp 515-533.

[5.32] **Verneau, A. (1984)**
"Recovery from Exhaust Gas on a Diesel Engine"
VDI Berichte, Proceedings of the International VDI Seminar, Zurich, Switzerland, 1984, pp 501-513.

[5.33] **Wieser, R. Von (1986)**
"Ein Verbrennungsmotor mit Dampferzeugungsanlage",
Wärme, Band 92, Heft 1, February 1986, pp 1-4.

[5.34] **Wingård, Sture; Åmand, Lars-Erik; Leckner, Bo (1988)**
"Dieselmotor Kombinerad med Fluidiserad Bädd för Kraftvärmeproduktion",
Report Nr. A88-175, Department of Energy Technology, Chalmers Institute of Technology, Göteborg, Sweden, October 1988.

[5.35] **Woodward, J.B. (1994)**
"Evaluation of Brayton and Rankine Alternatives for Diesel Waste Heat Exploitation",
Journal of Engineering for Gas Turbines and Power, Transactions of the ASME, Vol.116, Nr.1, January 1994, pp 39-45.

Other Related References:

[9.1] **Anheden, Marie; Martin, Andrew R. (2000)**
"Thermodynamic Performance Analysis and Economic Evaluation of Externally Fired Gas Turbine Cycles for Small Scale Biomass Cogeneration",
ASME Paper 2000-GT-0016, Proceedings of ASME TurboExpo, Munich, Germany, May 2000.

[9.2] **Barre, Eric (1996)**
"Comparative Analysis on the Use of Gas Turbines or Piston Engines in Small Combined Heat and Power Units"
M.Sc. Thesis Project, Report Nr.488, Division of Heat & Power Technology, Royal Institute of Technology, Stockholm, Sweden, May 1996.

[9.3] **Bem, J. (1985)**
"New Zealand Synthetic Gasoline Plant",
Proceedings of the 20th Intersociety Energy Conversion Engineering Conference, Miami Beach, Florida, August 1985, Vol. 1, pp 517-522.

[9.4] **Bushnell, D.J.; Reistad, G.M.; Bauer, T.; Brynjolfsson, S.; Fox, S. (1985)**
"First and Second Law Evaluations of Wood-Fired Combined Cycle Power Plants",
Presented at the Winter Annual Meeting of the ASME (Advanced Energy Systems Division), Miami Beach, Florida, 1985, pp 121-133.

[9.5] **Eidensten, L.; Yan, J.; Svedberg, G. (1994)**
"Biomass Externally Fired Gas Turbine Cogeneration",
ASME Paper 94-GT-345, Proceedings of the International Gas Turbine and Aeroengine Congress and Exposition, Hague, Netherlands, June 1994.

[9.6] **Ekström, Clas; Amnell, Gösta; Anheden, Marie; Eidensten, Lars; Kierkegaard, Gunilla (2001)**
"Biobränsle från skogen. En studie av miljökonsekvenser och ekonomi för olika användningar",
Report Nr. U 01:30, Vattenfall Utveckling AB, Miljösystem, 810 70 Älvkarleby, Sweden, July 2001.

[9.7] **Horlock, J.H. (1995)**
"Combined Power Plants – Past, Present, and Future",
Journal of Engineering for Gas Turbines and Power, Transactions of the ASME, Vol.117, Issue 4, October 1995, pp 608-616.

[9.8] **Imler, Darell L.; Pollock, Donald C.; Kuhn, Alfred K. (1985)**
"Progress Report from the Great Plains Gasification Project, May 1985",
Proceedings of the 20th Intersociety Energy Conversion Engineering Conference, Miami Beach, Florida, August 1985, Vol. 1, pp 544-548.

[9.9] **Jacoby, Jürgen (1999)**
"Biomass as a Renewable Energy Source – Background and Some Case Studies",
Literature Study, Report Nr. 99/10, Division of Heat and Power Technology, Royal Institute of Technology, Stockholm, September 1999.

[9.10] **Kaikko, Juha (2000)**
"Air Bottoming Cycle, an Alternative to Combined Cycles",
Final Report Nr. EKV-00-28, SNEA Project P10955-1, Division of Heat and Power Technology, Royal Institute of Technology, Stockholm, December 2000.

[9.11] **Legerer, F.J. (1993)**
"Indirectly Fired Gas Turbine Cycle, Suitable for Biomass Combustion",

Proceedings of the 7th ASME COGEN-TURBO Congress and Exposition, Bournemouth, UK, September 1993, pp 511-515.

[9.12] **Näsholm, Ann-Sofi E.; Svedberg, Gunnar; Westermark, Mats O.J. (1993)**
"Exergy Analysis of Biomass-Fired Cogeneration Plant in a Pulp and Paper Mill",
ASME Paper 93-GT-354, Proceedings of the International Gas Turbine and Aeroengine Congress and Exposition, Cincinnati, Ohio, May 1993.

[9.13] **Riccio, G.; Martelli, F.; Maltagliati, S. (2000)**
"Study of an External Fired Gas Turbine Power Plant Fed by Solid Fuel",
ASME Paper 2000-GT-0015, Proceedings of ASME TURBOEXPO, Munich, Germany, May 2000.

[9.14] **Yan, J.; Eidensten, L.; Svedberg, G. (1994)**
"Parametric Analysis of Biomass Externally Fired Gas Turbine Cogeneration",
Proceedings of the Florence World Energy Research Symposium, Florence, Italy, July 1994, pp 877-887.

APPENDIX
A.1.

CHEMICAL ELEMENTS IN BIOMASS

Table : Characteristics of the reference fuels - major elements and ash (from [4.4])

Fuel group	C.V.$_{gross}$ ($MJ/kg_{d.m.}$)	±	C (% d.s.)	±	H (% d.s.)	±	O (% d.s.)	±	N (% d.s.)	±	S (% d.s.)	±	Cl (% d.s.)	±
1. Wood	20.5	0.5	51.9	1.2	6.0	0.3	41.8	1.4	0.12	0.08	0.009	0.003	0.015	0.008
2. Debris	20.6	0.5	51.6	0.5	6.0	0.2	39.0	0.1	0.48	0.16	0.036	0.017	-	-
3. Bark	20.4	0.2	52.5	0.2	5.7	-	39.3	-	0.40	0.14	0.032	0.009	0.02	-
4. Needles	-	-	-	-	-	-	-	-	1.0	0.2	0.077	0.012	-	-
5. Salix	19.7	0.1	48.4	1.6	5.9	0.3	44.2	2.4	0.53	0.10	0.057	0.023	0.008	0.003
6. Timothy	18.6	-	46.4	-	6.3	-	40.0	-	1.2	-	0.16	0.06	0.03	-
7. Lucerne	18.9	-	46.7	-	5.9	-	35.6	-	3.1	-	0.25	0.16	0.6	0.4
8. Rape	18.2	0.5	46.2	0.9	5.7	0.2	38.8	0.7	0.76	0.12	0.17	0.04	0.22	0.06
9. Barley	18.6	0.6	45.8	0.7	5.7	0.4	41.9	1.8	0.52	0.19	0.12	0.04	0.4	0.4
10. Wheat	18.9	0.6	46.2	0.7	5.8	0.3	41.3	2.1	0.59	0.20	0.08	0.03	0.15	0.10
11. RC-summer	18.3	0.5	45.0	1.3	5.7	0.3	46.2	5.0	1.1	0.50	0.19	0.07	0.74	0.21
12. RC-spring	17.9	0.6	44.0	0.9	5.8	0.1	37.5	4.6	0.79	0.56	0.15	0.04	0.11	0.09
13. Peat Ref	21	-	54.5	-	5.6	-	32	-	1.4	-	0.24	-	0.09[1]	0.11
14. Peat-average	22.7	1.6	58.8	2.9	6.1	0.4	32.6	3.8	2.0	1.0	0.26	0.23	0.09[1]	0.11
15. MSW	17.2	-	42.4	-	6.1	-	35.1	-	2.2	-	0.24	-	0.73 (0.5[2])	-
16. Coal 1	28.2	-	74.2	-	4.8	-	11.4	-	1.3	-	0.35	-	0.012	-
17. Coal 2	27.9	1.8	71.7	1.5	4.7	0.6	8.3	1.4	1.3	0.4	0.64	0.22	0.06	0.04

1) from ref. 40
2) average in Swedish MSW

Table : Characteristics of the reference fuels - ash forming elements (from [4.4])

Fuel gr.	Si (%)	±	Al (%)	±	Fe (%)	±	Mg (%)	±	Ca (%)	±	K (%)	±	Na (%)	±	P %	±
1. W	0.0028	0.0018	0.0019	0.0016	0.0026	0.0013	0.016	0.009	0.065	0.027	0.040	0.025	0.018	0.03	0.0075	0.0070
2. d	0.50	0.13	0.11	0.03	0.047	0.047	0.051	0.012	0.42	0.14	0.16	0.07	0.021	0.003	0.043	0.017
3. B	0.14	0.17	0.033	0.027	0.019	0.015	0.062	0.026	0.60	0.33	0.17	0.11	0.036	0.037	0.041	0.024
4. N	-	-	-	-	0.0056	0.0011	0.078	0.010	0.30	0.20	0.51	0.06	-	-	0.15	0.03
5. s	0.038	0.009	0.0055	-	0.0067	-	0.056	0.012	0.37	0.15	0.31	0.07	0.011	0.009	0.076	0.010
6. t	1.4	0.06	0.032	0.022	0.024	0.004	0.13	0.04	0.58	0.25	2.2	0.8	0.056	0.034	0.31	0.13
7. l	0.15	-	0.019	-	0.022	0.005	0.18	0.03	1.6	0.2	2.5	0.6	0.092	0.047	0.29	0.01
8. r	2.1	2.2	0.22	0.20	0.16	0.16	0.10	0.03	1.3	0.2	1.0	0.3	0.14	0.06	0.085	0.017
9. b	1.2	0.4	0.037	0.033	0.026	0.023	0.10	0.03	0.40	0.09	1.1	0.5	0.20	0.17	0.092	0.032
10. w	1.8	0.9	0.023	0.022	0.026	0.027	0.11	0.02	0.40	0.12	0.94	0.25	0.042	0.052	0.075	0.020
11. c1	1.6	0.7	0.019	0.014	0.016	0.010	0.13	0.04	0.38	0.12	1.1	0.5	0.091	0.035	0.18	0.04
12. c2	4.6	2.1	0.11	0.17	0.06	0.09	0.05	0.02	0.20	0.11	0.34	0.09	0.068	0.037	0.10	0.05
13. P.ref[1]	0.52	0.4	0.28	0.15	0.63	0.36	0.086	0.086	0.89	0.47	0.020	0.018	0.014	0.017	0.04	0.02
14. P.av.	0.55	1.3	0.24	0.26	0.67	0.55	0.064	0.025	0.38	0.19	0.025	0.015	0.0092	0.0053	0.068	0.041
15. MSW	0.28	-	0.21	-	4.0	-	0.12	-	4.57	-	0.17	-	0.21	-	2.0	
16. C1	3.4	-	0.88	-	0.22	-	0.052	-	0.27	-	0.058	-	0.087	-	0.015	-
17. C2	4.8	1.2	2.0	0.7	0.47	0.3	0.13	0.13	0.38	0.44	0.15	0.07	0.11	0.12	0.02	0.01

1) Given as oxides (% of ash), recalculated using the average ash content

Table : Trace elements ($ppm_{d.s}$) (from [4.4])

Trace element	Wood[47]	Salix	Wheat	Rape	Barley	Peat	MSW[49,50]	Coal[44]
Cu	1.8 (0.5-6)	3.8±2.4	46±13	62±15	29±18	10±12	1.5 (100-1500)	13 (7-50)
Zn	22 (4-40)	66±19	110±40	110±40	120±60	10±16	1200 (600-1600)	27 (4-180)
Pb	3.6 (0.5-13)	1.5±0.6	86±20	45±17	58±36	2.5±5.0	1200 (50-1600)	13 (2-70)
Cd	0.2 (0.1-0.4)	1.3±0.7	0.02±0.01	0.05±0.07	0.02±0.02	0.2±0.2	3.5 (2-5.5)	0.3 (0.03-0.8)
Hg	0.02 (0.01-0.02)	?	0.01	0.01±0.006	0.01	0.05±0.1	1.5 (0.5-5)	0.1 (0.02-1.6)

Table :
Compositions, heating values, and alkali index for selected fuels (from [2.29])

	Alfalfa stems	Wheat straw	Rice hulls	Rice straw	Switch-grass	Sugar cane bagasse	Willow wood	Hybrid poplar
Proximate analysis (% dry fuel)								
Fixed carbon	15.81	17.71	16.22	15.86	14.34	11.95	16.07	12.49
Volatile matter	78.92	75.27	63.52	65.47	76.69	85.61	82.22	84.81
Ash	5.27	7.02	20.26	18.67	8.97	2.44	1.71	2.70
Total	100.00	100.00	100.00	100.00	100.00	100.00	100.00	100.00
Ultimate analysis (% dry fuel)								
Carbon	47.17	44.92	38.83	38.24	46.68	48.64	49.90	50.18
Hydrogen	5.99	5.46	4.75	5.20	5.82	5.87	5.90	6.06
Oxygen (diff.)	38.19	41.77	35.47	36.26	37.38	42.82	41.80	40.43
Nitrogen	2.68	0.44	0.52	0.87	0.77	0.16	0.61	0.60
Sulfur	0.20	0.16	0.05	0.18	0.19	0.04	0.07	0.02
Chlorine	0.50	0.23	0.12	0.58	0.19	0.03	< 0.01	0.01
Ash	5.27	7.02	20.26	18.67	8.97	2.44	1.71	2.70
Total	100.00	100.00	100.00	100.00	100.00	100.00	100.00	100.00
Elemental composition of ash (%)								
SiO_2	5.79	55.32	91.42	74.67	65.18	46.61	2.35	5.90
Al_2O_3	0.07	1.88	0.78	1.04	4.51	17.69	1.41	0.84
TiO_2	0.02	0.08	0.02	0.09	0.24	2.63	0.05	0.30
Fe_2O_3	0.30	0.73	0.14	0.85	2.03	14.14	0.73	1.40
CaO	18.32	6.14	3.21	3.01	5.60	4.47	41.20	49.92
MgO	10.38	1.06	< 0.01	1.75	3.00	3.33	2.47	18.40
Na_2O	1.10	1.71	0.21	0.96	0.58	0.79	0.94	0.13
K_2O	28.10	25.60	3.71	12.30	11.60	0.15	15.00	9.64
SO_3	1.93	4.40	0.72	1.24	0.44	2.08	1.83	2.04
P_2O_5	7.64	1.26	0.43	1.41	4.50	2.72	7.40	1.34
CO_2 /other	14.80						18.24	8.18
Total	100.00	100.00	100.64	100.00	100.00	100.00	100.00	100.00
Undetermined	11.55	1.82	−0.64	2.68	2.32	1.39	8.38	1.91

	Almond shells	Almond hulls	Pist. shells	Olive pitts	Demol. wood	Yard waste	Fir mill	Mixed paper	RDF
Higher heating value (constant volume)									
MJ/kg	18.67	17.94	15.84	15.09	18.06	18.99	19.59	19.02	
Btu/lb	8025	7714	6811	6486	7766	8166	8424	8178	
Alkali index (as oxide)									
(kg alkali/GJ)	0.82	1.07	0.50	1.64	0.60	0.06	0.14	0.14	
(lb alkali/MM Btu)	1.92	2.49	1.17	3.82	1.41	0.15	0.32	0.32	
Proximate analysis (% dry fuel)									
Fixed carbon	20.71	20.07	16.95	16.28	12.32	13.59	17.48	7.42	0.47
Volatile matter	76.00	73.80	81.64	82.00	74.56	66.04	82.11	84.25	73.40
Ash	3.29	6.13	1.41	1.72	13.12	20.37	0.41	8.33	26.13
Total	100.00	100.00	100.00	100.00	100.00	100.00	100.00	100.00	100.00
Ultimate analysis (% dry fuel)									
Carbon	49.30	47.53	50.20	52.80	46.30	41.54	51.23	47.99	39.70
Hydrogen	5.97	5.97	6.32	6.69	5.39	4.79	5.98	6.63	5.78
Oxygen (diff.)	40.63	39.16	41.15	38.25	34.45	31.91	42.10	36.84	27.24
Nitrogen	0.76	1.13	0.69	0.45	0.57	0.85	0.06	0.14	0.80
Sulfur	0.04	0.06	0.22	0.05	0.12	0.24	0.03	0.07	0.35
Chlorine	< 0.01	0.02	< 0.01	0.04	0.05	0.30	0.19		
Ash	3.29	6.13	1.41	1.72	13.12	20.37	0.41	8.33	26.13
Total	100.00	100.00	100.00	100.00	100.00	100.00	100.00	100.00	100.00
Elemental composition of ash (%)									
SiO_2	8.71	9.28	8.22	30.82	45.91	59.65	15.17	28.10	33.81
Al_2O_3	2.72	2.09	2.17	8.84	15.55	3.06	3.96	52.56	12.71
TiO_2	0.09	0.05	0.20	0.34	2.09	0.32	0.27	4.29	1.66
Fe_2O_3	2.30	0.76	35.37	6.58	12.02	1.97	6.58	0.81	5.47
CaO	10.50	8.07	10.01	14.66	13.51	23.75	11.90	7.49	23.44
MgO	3.19	3.31	3.26	4.24	2.55	2.15	4.59	2.36	5.64
Na_2O	1.60	0.87	4.50	27.80	1.13	1.00	23.50	0.53	1.19

	Almond shells	Almond hulls	Pist. shells	Olive pitts	Demol. wood	Yard waste	Fir mill	Mixed paper	RDF
K_2O	48.70	52.90	18.20	4.40	2.14	2.96	7.00	0.16	0.20
SO_3	0.88	0.34	3.79	0.56	2.45	2.44	2.93	1.70	2.63
P_2O_5	4.46	5.10	11.80	2.46	0.94	1.97	2.87	0.20	0.67
CO_2 /other	17.38	20.12					18.92		
Total	100.00	100.00	100.00	100.00	100.00	100.00	100.00	100.00	100.00
Undetermined	−0.53	−2.89	2.48	−0.70	1.71	0.73	2.31	1.80	12.58
Higher heating value (constant volume)									
MJ/kg	19.49	18.89	18.22	21.59	18.41	16.30	20.42	20.78	15.54
Btu/lb	8378	8119	7831	9282	7916	7009	8779	8934	6679
Alkali index (as oxide)									
(kg alkali/GJ)	0.85	1.75	0.18	0.26	0.23	0.49	0.06	0.03	0.23
(lb alkali/MMBtu)	1.98	4.06	0.41	0.60	0.54	1.15	0.14	0.06	0.54

	Biomass power plant fuel blends						Coal (lvb)[a]	Lignite
	Urban/ (1) Ag wood	Urban/ (2) Ag wood	Urban/ wood	Wood/ pit	Wood/ Al. hull	Wood/ Wheat straw		
Proximate analysis (% dry fuel)								
Fixed carbon	19.79	16.93	15.23	18.10	15.94	16.67	77.00	43.44
Volatile matter	75.89	80.57	79.23	76.77	77.28	75.14	18.49	42.95
Ash	4.32	2.50	5.54	5.13	6.78	8.19	4.51	13.69
Total	100.00	100.00	100.00	100.00	100.00	100.00	100.00	100.08
Ultimate analysis (% dry fuel)								
Carbon	51.44	49.69	48.77	48.62	47.45	47.48	87.52	60.97
Hydrogen	5.67	5.87	5.76	5.78	5.53	5.81	4.26	4.07
Oxygen (diff.)	38.12	41.52	39.53	39.73	39.54	37.92	1.55	18.50
Nitrogen	0.41	0.33	0.27	0.65	0.59	0.35	1.25	1.02
Sulfur	0.03	0.04	0.07	0.06	0.08	0.12	0.75	1.81
Chlorine	0.01	0.05	0.06	0.03	0.03	0.13	0.16	0.04
Ash	4.32	2.50	5.54	5.13	6.78	8.19	4.51	13.69
Total	100.00	100.00	100.00	100.00	100.00	100.00	100.00	100.10

Elemental composition of ash (%)								
SiO_2	39.96	28.81	55.12	52.55	45.60	55.50	37.24	20.93
Al_2O_3	12.03	8.47	12.49	13.15	10.75	9.37	23.73	13.78
TiO_2	0.87	0.83	0.72	0.43	0.54	0.50	1.12	0.41
Fe_2O_3	7.43	3.28	4.51	8.18	4.06	4.77	16.83	12.08
CaO	19.23	27.99	13.53	10.06	18.96	11.04	7.53	16.13
MgO	4.30	4.49	2.93	3.27	4.22	2.55	2.36	4.40
Na_2O	1.53	3.18	3.19	5.90	3.08	2.98	0.81	6.41
K_2O	5.36	8.86	4.78	5.04	6.26	6.40	1.81	0.22
SO_3	1.74	2.00	1.92	2.10	2.06	1.80	6.67	24.27
P_2O_5	1.50	2.57	0.88	1.90	1.47	1.04	0.10	0.00
CO_2/other	6.05	6.07						
Total	100.00	100.00	100.07	100.00	100.00	100.00	98.20	98.63
Undetermined	0.00	3.45	−0.07	−2.58	3.00	4.05	1.80	1.37
Higher heating value (constant volume)								
MJ/kg	20.50	19.49	19.45	19.66	15.89	18.80	35.01	23.35
Btu/lb	8815	8379	8361	8450	6829	8083	15052	10040
Alkali index (as oxide)								
(kg alkali/GJ)	0.15	0.15	0.23	0.29	0.40	0.41	0.03	0.39
(lb alkali/MM Btu)	0.34	0.36	0.53	0.66	0.93	0.95	0.08	0.90

[a]Low volatile bituminous.

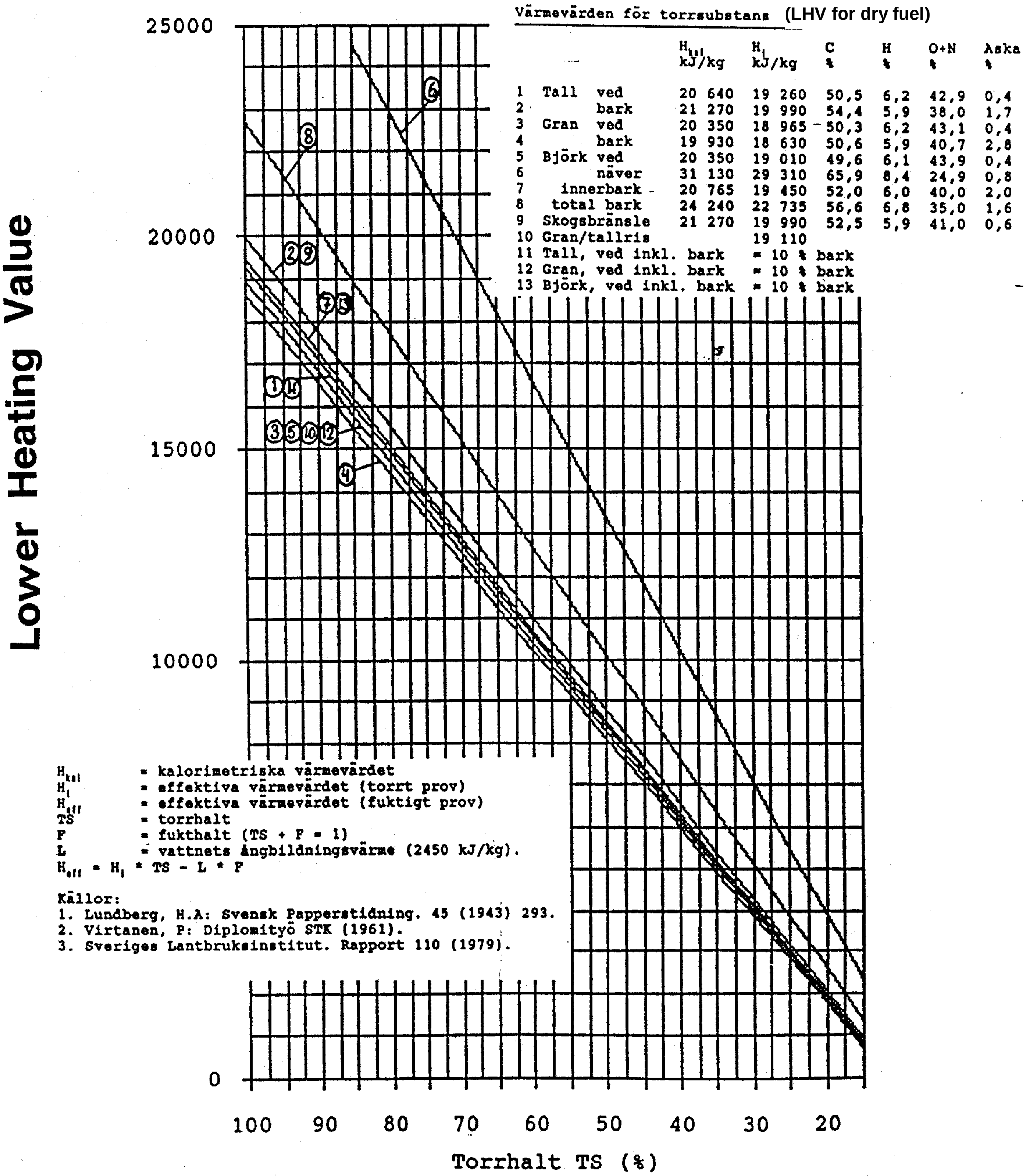
APPENDIX
A.2.

IMPACT OF MOISTURE CONTENT ON LHV OF BIOMASS FUEL

The figure shows the change of LHV for certain types of wood fuels (in kJ/kg) with rising moisture content (**horizontal axis gives percentage of dry mass in total mass**). (from [4.4])

Legend:

1. Pine wood, **2**. Pine bark, **3**. Fir/spruce wood, **4**. Fir/spruce bark, **5**. Birch wood, **6**. Outer white birch bark, **7.** Inner birch bark, **8.** Birch bark total, **9.** Forest residues/woodchips general, **10**. Fir & pine twigs, **11**. Pine total (wood + bark), **12**. Fir/spruce total (wood + bark), **13**. Birch total (wood + bark).



Värmevärden för torrsubstans (LHV for dry fuel)

			H_{kal} kJ/kg	H_l kJ/kg	C %	H %	O+N %	Aska %
1	Tall	ved	20 640	19 260	50,5	6,2	42,9	0,4
2		bark	21 270	19 990	54,4	5,9	38,0	1,7
3	Gran	ved	20 350	18 965	50,3	6,2	43,1	0,4
4		bark	19 930	18 630	50,6	5,9	40,7	2,8
5	Björk	ved	20 350	19 010	49,6	6,1	43,9	0,4
6		näver	31 130	29 310	65,9	8,4	24,9	0,8
7		innerbark	20 765	19 450	52,0	6,0	40,0	2,0
8		total bark	24 240	22 735	56,6	6,8	35,0	1,6
9	Skogsbränsle		21 270	19 990	52,5	5,9	41,0	0,6
10	Gran/tallris			19 110				
11	Tall, ved inkl. bark			≈ 10 % bark				
12	Gran, ved inkl. bark			≈ 10 % bark				
13	Björk, ved inkl. bark			≈ 10 % bark				

H_{kal} = kalorimetriska värmevärdet
H_l = effektiva värmevärdet (torrt prov)
H_{eff} = effektiva värmevärdet (fuktigt prov)
TS = torrhalt
F = fukthalt (TS + F = 1)
L = vattnets ångbildningsvärme (2450 kJ/kg).

$H_{eff} = H_l * TS - L * F$

Källor:
1. Lundberg, H.A: Svensk Papperstidning. 45 (1943) 293.
2. Virtanen, P: Diplomityö STK (1961).
3. Sveriges Lantbruksinstitut. Rapport 110 (1979).

APPENDIX
A.3.

A SHORT DISCUSSION ON CYCLE EFFICIENCIES

Basic considerations for defining Electric Efficiency of combined cycles:

It may be useful to outline in a simple form the definitions for electric efficiency of combined cycles, deriving also formulations for hybrid dual-fuel combined cycles. Some generalized considerations are presented below.

The efficiency of conversion of fuel energy into electric power by any method can be expressed as a ratio of the useful energy output to the raw energy input. This is also the case with thermodynamic cycles. Thus, electric efficiency is defined as:

$$\eta_{el} = \frac{P_{el}}{Q_{fuel}} \qquad (1)$$

where η_{el} is the symbol for electric efficiency of the overall cycle, $\mathbf{P_{el}}$ is the amount of fuel energy converted to electrical energy (useful product) and $\mathbf{Q_{fuel}}$ is the energy input in the cycle in the form of fuel. $\mathbf{Q_{fuel}}$ involves in itself the energy released by combustion of the fuel (which is the major component and can be expressed as the product of fuel mass flow $\mathbf{m_{fuel}}$ times its LHV or HHV), plus any sensible heat that the fuel may carry with it into the combustion chamber, plus possible kinetic or potential energy of the fuel flow. The use of the electric power $\mathbf{P_{el}}$ (being the sole final product) in the numerator of the equation suggests that further definition is needed to state whether all possible energy "losses" and internal consumption from/in the cycle are taken into account, implying the usage of the concept of "net" or "gross" electric efficiency. In order to generalize the discussion, there is no need to consider this distinction here, neither to state the difference between LHV and HHV.

In the special case of a pure straightforward combined cycle ("unfired" CC), there are two points of power production within the cycle (the topping cycle $\mathbf{P_{TCel}}$ and the bottoming cycle $\mathbf{P_{BCel}}$, which add to each-other), while the energy input in the cycle occurs at one point only – the combustion chamber of the gas turbine for example (the topping cycle). The definition of electric efficiency for the overall combined cycle will therefore be:

$$\eta_{elCC} = \frac{P_{TCel} + P_{BCel}}{Q_{TCfuel}} \qquad (2)$$

This definition can be extended to incorporate the specific energy "losses" connected with transfer of energy from the topping (TC) to the bottoming (BC) cycle, namely the fact that not all rejected energy from the TC can be utilized and converted to electric power by the BC. The efficiency equation is expressed usually with the help of the efficiencies for the separate TC and BC as follows:

$$\eta_{elCC} = \frac{\eta_{TCel} Q_{TCfuel} + \eta_{BCel} \eta_{HRSG} (1 - \eta_{TCel}) Q_{TCfuel}}{Q_{TCfuel}} \qquad (3)$$

where η_{BCel} is the efficiency of energy conversion in the bottoming cycle itself and η_{HRSG} is the efficiency of heat transfer from the TC to the BC.

In the special case of supplementary firing between the TC and the BC, the efficiency definition must be further extended to incorporate the fact that energy input into the combined cycle now occurs at two different points. This can be easily accomplished by adding this new source of energy input into the denominator of the equation, together with adding it to the energy supply for the BC in the numerator:

$$\eta_{elCCSF} = \frac{\eta_{TCel} Q_{TCfuel} + \eta_{BCel} \eta_{HRSG} \left\{ (1 - \eta_{TCel}) Q_{TCfuel} + Q_{BCfuel} \right\}}{Q_{TCfuel} + Q_{BCfuel}} \tag{4}$$

where $\mathbf{Q_{TCfuel}}$ and $\mathbf{Q_{BCfuel}}$ are the separate fuel energy inputs in the topping and bottoming cycle, respectively.

It is often helpful to define a ratio of the two energy inputs, namely the fuel energy for supplementary firing (additional fuel input in BC) to the fuel energy input in the TC, as:

$$\phi = \frac{Q_{BCfuel}}{Q_{TCfuel}} \tag{5}$$

Using this ratio to simplify the appearance of the efficiency equation for combined cycles with supplementary firing, we reach the following result:

$$\eta_{elCCSF} = \frac{\eta_{TCel} + \eta_{BCel} \eta_{HRSG} \left\{ (1 - \eta_{TCel}) + \phi \right\}}{1 + \phi} \tag{6}$$

In the case of hybrid dual-fuel combined cycles, the same efficiency equation as that for supplementary-fired cycles is valid, as long as most hybrid cycles can be considered as combined cycles with supplementary firing of different fuel in the BC. The ratio “ϕ” of fuel energy input in the BC to that in the TC readily applies for different types of fuels too, which is exactly the case with hybrid combined cycles. The efficiency of heat transfer in the HRSG η_{HRSG} from Eq.(6) must be substituted with a coefficient giving the amount of energy which is actually transferred from the TC to the BC, denoted for example with “μ”. The boiler efficiency of the bottoming cycle (burning solid fuels) $\boldsymbol{\eta_{boiler}}$ may also appear in the equation.

A generalized efficiency equation for hybrid dual-fuel combined cycles can therefore be presented as:

$$\eta_{elHCC} = \frac{\eta_{TCel} + \eta_{BCel} \left\{ (1 - \eta_{TCel}) \mu + \eta_{boiler} \phi \right\}}{1 + \phi} \tag{7}$$

This equation is valid basically for all hybrid combined cycles, except for some special types of parallel-powered cycles with very low level of “hybridization”, as for example separate GT with HRSG and solid fuel boiler feeding steam to a common steam turbine (in such cycles the TC and BC are not distinguished and there is no real transfer of energy from a TC to a BC). The variations in cycle configuration can be incorporated in the value of "μ" and the ratio "φ".

If we elaborate further on the efficiency concepts that are applicable specifically to hybrid dual-fuel combined cycles, we can try to define the electric efficiency attributable to the bottoming or topping fuel only. This follows from the attempt to compare the performance of a HCC to the performance of a sum of two separate units – one pure combined cycle and one solid fuel fired simple steam cycle – firing the same amount of fuels. A HCC may possibly reach higher electric efficiency than the average efficiency of separate pure combined cycle and solid fuel steam cycle with the same sum of (total) fuel energy input. Accordingly, these efficiencies can be compared in the form of equations as follows:

$$\frac{\eta_{TCel}Q_{TCfuel}+\eta_{BCel}\left\{(1-\eta_{TCel})\mu Q_{TCfuel}+\eta_{boiler}Q_{BCfuel}\right\}}{Q_{TCfuel}+Q_{BCfuel}} \leq\equiv\geq \frac{\eta_{CCel}Q_{TCfuel}+\eta_{BCel}Q_{BCfuel}}{Q_{TCfuel}+Q_{BCfuel}} \qquad (8)$$

where η_{CCel} is the electric efficiency of a pure combined cycle fired with the same fuel energy input as the topping cycle in the HCC arrangement (with the same heat engine and same basic parameters).
Further work on Eq. (8) can lead to the derivation of the efficiency of energy conversion attributable only to the bottoming fuel in a hybrid combined cycle. The following form has been suggested:

$$\eta_{BCel}=\frac{P_{HCCel}-\eta_{CCel}Q_{TCfuel}}{Q_{BCfuel}} \qquad (9)$$

which follows directly from Eq. (8), where the numerator of the left equation is the electric power output of the hybrid combined cycle $\mathbf{P_{HCCel}}$.

Eq. (9) provides a way to calculate how much of the power output is based on the bottoming fuel, by subtracting from the total HCC power output the possible energy conversion of the topping fuel in a high-efficiency pure combined cycle. If the result from Eq. (9) for a given HCC configuration is higher than the efficiency of the bottoming cycle working separately, then this HCC configuration leads to improvement in energy utilization of the bottoming fuel itself. Furthermore, this would also prove that the given HCC configuration provides higher efficiency, compared to the average of separate cycles (one simple cycle fired with the bottoming fuel and one pure combined cycle fired with the topping fuel).

Furthermore, the possibility to "repower" old steam plants by adding a topping cycle to the steam one, calls for a definition of the efficiency attributable to the topping fuel only. This can also be expressed as the efficiency of additional power production by the topping fuel, which comprises the installed power of the topping engine and the increase of power output from the bottoming cycle due to transfer of energy from the TC to the BC. Various authors define this efficiency as "additional power efficiency" or "marginal efficiency" or "incremental efficiency". This can be presented in a form of a generalized equation as follows:

$$\eta_{incremental} = \frac{\Delta P}{\Delta Q_{fuel}} = \frac{P_{TC} + \Delta P_{BC}}{Q_{TC} \pm \Delta Q_{BC}} \tag{10}$$

The incremental efficiency $\eta_{\mathbf{incr}}$ is the efficiency of energy conversion of the additional fuel (the topping fuel plus any change in the bottoming fuel flow) to electric power by the TC itself, plus the increment of power for the BC.

Total Energy Efficiency:

The concept of total efficiency of energy conversion by a thermodynamic cycle takes into account the fact that certain heat output in the form of hot water or steam can also be considered as useful output energy or "product", together with the electrical power. The calculation of total efficiency is similar to that for electric efficiency. If the useful heat output is added in the numerator of Eq. (1) and in every other equation thereafter, the result will be converted directly into total efficiency. If the useful heat energy output from any thermodynamic cycle is in the form of district heating (DH), the basic equation for total energy efficiency can be put as follows:

$$\eta_{TOTAL} = \frac{P_{el} + Q_{DH}}{Q_{fuel}} \tag{11}$$

Or in the form specific for hybrid combined cycles:

$$\eta_{TOTAL} = \frac{\eta_{TCel} Q_{TCfuel} + \eta_{BCel} \left\{ (1 - \eta_{TCel}) \mu Q_{TCfuel} + \eta_{boiler} Q_{BCfuel} \right\} + Q_{DH}}{Q_{TCfuel} + Q_{BCfuel}} \tag{12}$$

Turning to the Second Law approach:

The above stated equations are useful for calculation of efficiencies of hybrid dual-fuel combined cycles in particular and of combined cycles with supplementary firing in general, in terms of basic energy relations. The definitions of incremental power efficiency and bottoming fuel utilization efficiency (Eq. 10 and 9) can be applied as comparison of various combined cycle configurations. The calculation of these efficiencies however is dependent on the availability of exact values for the quantities represented in the equations. These quantities cannot be known, unless the hybrid cycle is simulated and heat-balanced, or is existent and it's performance is tabulated by measurements.

If analytical approach for comparison of different combined cycle configurations is sought, the above presented equations cannot help much, as long as they are a final form of comparison, once the cycle performance is known. The potential for higher electric efficiency of one configuration over another may be derived without simulating the cycles, if the exergy approach is applied, using the second law of thermodynamics. This can again be generalized and simplified by a straightforward procedure based on the already derived equations. As a first view on second law efficiencies of any thermodynamic cycle, the exergetic efficiency of energy conversion from fuel to final energy forms can be stated as follows:

$$\eta_{II} = \frac{Exergy\ in\ product}{Exergy\ in\ fuel} \tag{13}$$

Or if we want to be more specific:

$$\eta_{HCC,II} = \frac{\eta_{TCel}Q_{TCfuel} + \eta_{BCel}\left\{(1-\eta_{TCel})\mu Q_{TCfuel} + \eta_{boiler}Q_{BCfuel}\right\} + \left(1-\frac{T_0}{T_{DH}}\right)Q_{DH}}{Q_{TCfuel} + Q_{BCfuel}} \tag{14}$$

which presents a generalized form of second law efficiency for a hybrid dual-fuel combined cycle in cogeneration mode (electricity and district heating outputs). Eq. 14 taken into account the fact that according to the exergy concept, electricity is a final high-grade product, while any heat output from the cycle should be represented by its exergy value (maximum convertible energy into electricity with a reversible thermo-dynamic cycle). Eq. 14 also uses the simplification of representing the exergy content in fuels with their energy content with good approximation.

The main point of interest for applying the exergy concept to combined cycles and to bottoming cycles in particular, is the possible estimation of the efficiency with which the bottoming cycle accepts and converts into electricity the exergy, which is rejected from the topping cycle and transferred to the bottoming one. Of course, little knowledge is needed to guess that a more "exergetically efficient" bottoming cycle would be a one that utilizes the rejected heat from the topping cycle at the highest possible temperature level. This can be evaluated by tracking the efficiency of exergy transfer of any heat exchangers transferring energy/exergy from the topping to the bottoming cycle, which is the same as tracking any entropy production, irreversibilities and exergy destruction. The pinpointing of places of typical entropy producing processes (heat exchange through large temperature differences, mixing of fluids with different temperatures or with different composition and pressure losses) can accomplish any attempted numerical calculation of a given example.

An attempt to define again a generalized formulation of electric efficiency for a hybrid combined cycle according to the Second Law, would lead to an equation very similar to Eq. 4, 7 and 8:

$$\eta_{HCC,II} = \frac{\eta_{TCel}Q_{TCfuel} + \eta_{BCel}\left\{(1-\eta_{TCel})\mu^{II} Q_{TCfuel} + Q_{BCfuel}\right\}}{Q_{TCfuel} + Q_{BCfuel}} \tag{15}$$

where the generalized coefficient "μ^{II}" contains in itself all irreversibilities (entropy generation and exergy destruction) involved in the transfer of energy/exergy from the topping to the bottoming cycle. The exact value of "μ^{II}" can be found (numerically represented) or simply re-defined by equations for every given exact configuration of an overall combined cycle arrangement…