NEW AND ADVANCED ENERGY CONVERSION TECHNOLOGIES. ANALYSIS OF COGENERATION, COMBINED AND INTEGRATED CYCLES

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The research presented in this thesis was carried out at the Laboratory of Thermal Engineering of the University of Twente and at the Netherlands Energy Research Foundation ECN.

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Prof.dr.ir. G.G. Hirs

Моим родителям

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Summary

At the current status of development in power technology large combined-cycle power plants are exceeding 70% of the maximum theoretical (Carnot) efficiency. These achievements, however, can be obtained only at a large scale for power plants above 300 MW. On a smaller scale, innovative designs are required to reach the same level of performance at feasible costs. Advances within power cycles, integration of cycles, combination of existing technologies – these are the possible ways to improve performance of small- and medium-scale power technology.

Identification and development of new energy conversion technologies and systems for distributed power generation applications are the objectives of the New Energy Conversion Technologies (NECT) programme, being realized by the Netherlands Agency for Energy and Environment (Novem). The part of the programme, which is dedicated to the development of new and improved combinations of existing energy conversion technologies, defines the structure of this thesis.

At the beginning, the basic thermodynamic cycles and their specific features are described. Because no single cycle can offer high efficiency due to the intrinsic limitations and the impossibility to operate within a broad temperature range, combined and advanced cycles are addressed. Combined cycles do not suffer from the drawbacks of the single cycles, since the heat rejected by the topping cycle is utilized by the bottoming one, and better performance can be obtained. The basic cycles are combined according to their temperature level: high-temperature cycles are good candidates for the topping application, and medium- or low-temperature cycles for bottoming. Of the combined cycles considered, each cycle is outlined and its schematic diagram is given. In addition to the combined cycles, improvements within a particular cycle are discussed in the chapter on advanced cycles.

The scope of the NECT programme covers power and heat production, so industrial cogeneration is assessed in various configurations (steam boiler, gas turbine, heat pumps) and operating modes.

Subsequently, several technologies, which are selected for further development within the NECT programme, are analyzed in detail.

One of the configurations is the Joule/Joule combined cycle, which consists of an existing gas turbine and an air bottoming turbine. The bottoming cycle adds 20–30% to the power output, which results in a combined-cycle efficiency of up to 45%. In comparison with a steam bottoming plant, the air bottoming cycle requires no steam/water equipment that enables an operation of small units at high efficiency and low costs without the complexity of the steam cycle. Since the air turbine exhaust is a hot sterile air, the implementation of the cycle in the food processing industries is considered.

Enhancement of direct-fired power plants with the use of gas turbine technology is investigated for a waste incineration plant and a solid fuel fired steam boiler. In the case of the waste incinerator, gas turbine integration adds 12–15% points to the basic plant gross efficiency. The hot windbox configuration shows the best match between natural gas consumption, overall efficiency, and the specific boiler surface area. Combustion with gas turbine exhaust allows exergy losses to be reduced by more than 9%.

In the case of the coal-fired steam boiler, a gas turbine cycle with an external heat exchanger, which substitutes the combustion chamber, is considered. This externally-fired combined cycle shows an overall efficiency of 47%. The use of supplementary firing is employed to increase the turbine inlet temperature and to reduce the costly surface of the high-temperature heat exchanger

An advanced gas turbine cycle with partial oxidation (POGT), which is another subject of the NECT programme, is compared with the reheat gas turbine. Performance of these cycles was found comparable, but some specific features of the POGT deserve further considerations. First, the low level of air excess reduces NO_X forming. Second, in view of its good exergetic efficiency, an implementation of the cycle for combined production of power and synthesis gas is proposed.

The work has resulted in two feasibility projects, which are financially supported by Novem and being carried out by the Energy Research Foundation ECN and the University of Twente. One project is aimed at the development and market introduction of several combined and advanced cycles, outlined in this thesis. The other project assesses the technical feasibility of the gas turbine with partial oxidation. The studies will evaluate the potential future application of the selected cycles and will include optimization, parametric and comparative analysis, experimental work, and a cost estimate. A development team of ECN, UT, an engineering company, manufacturers and end users will be established to introduce proposed systems into the application market.

Samenvatting

Met de huidige stand van de energietechnologie overschrijden grote combined-cycle elektriciteits-centrales al 70% van het maximum theoretisch (Carnot) rendement. Deze prestaties echter worden behaald alleen bij grote eenheden van meer dan 300 MW. Op kleinere schaal zijn er innovatieve cycli nodig om hetzelfde prestatieniveau tegen aanvaardbare kosten te bereiken. Verbeteringen binnen de cyclus zelf, combinaties van bestaande technologiën, integratie van thermodynamische cycli – het zijn de mogelijke methoden om efficiency van klein- en middenschalige energieopwekking te verbeteren.

Identificatie en ontwikkeling van nieuwe energie-conversie technologien en systemen voor decentrale energieopwekking zijn de doeleinden van het Nieuwe Energie-Conversie Technologiën (NECT) programma, dat door de Nederlandse Onderneming voor Energie en Milieu (Novem) wordt gerealiseerd. De structuur van dit proefschrift wordt bepaalt door het deel van het programma, dat zich op ontwikkeling van nieuwe en verbeterde combinaties van bestaande energie-conversie technologiën richt.

Om te beginnen worden de standaard thermodynamische cycli en hun specifieke eigenschappen beschreven. Dit deel omvat de Rankine, Joule-Brayton, Otto, Diesel en Stirling cycli, en ook brandstofcellen. Gecombineerde en geavanceerde cycli worden beschouwd, omdat het met een standaard cyclus niet mogelijk is een hoog rendement te halen vanwege intrinsieke beperkingen en de onmogelijkheid om binnen een breed temperatuurbereik te functioneren. Een gecombineerde cyclus heeft een beter rendement, omdat de restwarmte van de topping cyclus nog eens in de bottoming cyclus benut wordt. De standaard cycli zijn te combineren aan de hand van hun temperatuurniveau: hoge temperatuur cycli voor topping toepassingen, midden en lage temperatuur cycli voor bottoming. Elk van de beschouwde cycli wordt in hoofdlijnen beschreven en in een schema weergegeven.

Verbeteringen binnen standaard cycli woorden in een apart hoofdstuk besproken.

Binnen het kader van het NECT programma valt niet alleen elektrische energieopwekking, ook thermische energie produktie komt aan bod. Daarom wordt ook industriële warmte/kracht koppeling in verschillende configuraties (stoom- en gasturbines, warmtepompen) en bedrijfscondities beoordeeld.

Vervolgens worden de cycli, die geselecteerd zijn voor verdere uitwerking binnen het NECT programma, in detail geanalyseerd.

Een van de configuraties is de Joule/Joule cyclus, bestaande uit een standaard gasturbine en een nageschakelde luchtturbine. De luchtturbine levert 20–30% extra vermogen wat resulteert in overall efficiency van maximaal 45%. Deze cyclus kan ook warme lucht leveren, terwijl hetzelfde prestatieniveau behaald wordt als bij een cyclus met stoombottoming. Een implementatie van deze cyclus bij de voedselindustrie wordt in beschouwing genomen. Een ander voordeel van een dergelijke cyclus is de afwezigheid van een water-stoom systeem. Deze maakt exploitatie van kleine units met hoog rendement en lage kosten mogelijk zonder de complexiteit van de stoombottoming cyclus.

Verbetering van direct-gestookte ketels met gebruik van gasturbine technologie wordt onderzocht voor een afvalverbrandingsinstallatie (AVI) en voor een kolengestookte stoomketel. In het geval van AVI voegt de gasturbine integratie 12–15%

punten bij de bruto efficiency. De hot windbox configuratie toont een goede overeenkomst tussen aardgas verbruik, overall rendement en het specifieke verwarmingsoppervlak van de boiler. Verbranding met het uitlaatgas reduceert de exergieverliezen met meer dan 9%.

In het geval van een kolengestookte ketel wordt een gasturbine met externe warmtewisselaar, die verbrandingskamer vervangt, toegepast. Deze externally-fired combined cycle haalde een rendement van 47%. Er kan bijgestokt worden om de turbine entreetemperatuur te verhogen en het kostbare warmtewissellingsoppervlak te verkleinen.

Een geavanceerde gasturbine cyclus met partiële oxidatie (POGT), die ook bij het NECT programma hoort, wordt vergeleken met de reheat gasturbine. De prestaties van de POGT en de reheat gasturbine zijn vergelijkbaar bevonden. Echter sommige eigenschappen van de POGT cyclus verdienen meer aandacht: ten eerste; een lage luchtovermaat resulteert in NO_X reductie; ten tweede, gezien het goed exergetisch rendement, is een uitvoering van deze cyclus voor gecombineerde elektriciteit- en synthese gas productie mogelijk.

Dit werk heeft geresulteerd in twee haalbaarheidsprojecten die in opdracht van Novem worden uitgevoerd door het Energieonderzoek Centrum Nederland ECN en de Universiteit Twente. Een project richt zich op ontwikkeling en marktintroductie van een aantal gecombineerde en geavanceerde cycli, die in dit proefschrift worden beschouwd. Het andere project beoordeelt de technische haalbaarheid van de gasturbine met partiële oxidatie. Deze projecten evalueren de potentiële toepassingen van de geselecteerde cycli en bevatten optimalisatie, parametrische en vergelijkende analyses, experimentele werk en een kostenbegroting. Een development team bestaande uit het ECN, UT, een ingenieursbureau, fabrikanten en eindgebruikers zal ontstaan om voorgestelde systemen bij de toepassingsmarkt te introduceren.

Краткое содержание

Современные технологии позволяют преобразовывать химическую энергию топлива в электрическую с к.п.д., достигающим 70% от теоретического максимума Карно. Однако такие высокие показатели возможны только для крупных блоков комбинированного цикла мощностью свыше 300 МВт. Подобный уровень экономичности на установках меньшей мощности возможен за счет усовершенствования циклов. Возможными путями улучшения энергетических показателей станций средней и малой мощности являются модернизация и интеграция циклов, а также комбинирование существующих технологий.

Определение и разработка технологий для децентрализованного производства электроэнергии осуществляется в Нидерландах Агенством по Энергетике и Окружающей Среде Novem в рамках программы Новые Технологии Преобразования Энергии (NECT). Часть программы, посвященная разработке новых и усовершенствованных комбинаций существующих технологий, определила структуру настоящей диссертации.

В первой главе описываются основные термодинамические циклы (Ренкина, Джоуля-Брэйтона, Отто, Дизеля, и Стирлинга) и топливные элементы, с указанием их характерных особенностей и рабочих параметров. Вторая и третья глава рассматривают возможности улучшения к.п.д. за счет комбинирования и усовершенствования рассматриваемых циклов, поскольку отдельно взятый цикл не может обеспечить высокого к.п.д. в связи с присущими ему ограничениями и невозможностью работы в широком диапазоне температур. Комбинированные циклы имеют более высокий к.п.д., в связи с тем, что сбрасываемое от верхнего цикла тепло утилизируется в нижнем цикле. Приводятся возможные комбинации циклов, определенные исходя из их рабочих температур. Дается краткое описание и схема рассматриваемых комбинированных циклов. В отдельной главе обсуждаются усовершенствованные циклы.

Программа NECT рассматривает производство не только электрической, но и тепловой энергии, и поэтому четвертая глава посвящена промышленной теплофикации в различных конфигурациях (с тепловыми насосами, паровыми и газовыми турбинами) и режимах.

В последующих главах более подробно анализируются комбинированные и усовершествованные циклы, выбранные для разработки на следующем этапе программы NECT.

Одной из схем является комбинированный цикл типа Джоуль/Джоуль, который состоит из типовой газовой турбины и воздушной турбины, утилизирующей отходящее тепло верхнего цикла. Воздушная турбина увеличивает мощность установки на 20–30%, что позволяет достичь к.п.д. до 45%. Рабочие характеристки этого цикла сопоставимы с парогазовым циклом. Отсутствие пароводяной системы позволяет упростить конструкцию и осуществить эксплуатацию установок малой мощности с высоким к.п.д. Поскольку выхлоп воздушной турбины представляет собой стерильный, горячий воздух, пригодный для технологических нужд, рассматривается работа цикла в теплофикационном режиме.

В шестой и седьмой главах анализируются два коминированных цикла типа Джоуль/Ренкин, предлагаемые для модернизации паровых котлов с прямым сжиганием топлива. Рассматриваются схемы интеграции газовой турбины с мусоросжигательной установкой и схема газовой турбины, камера сгорания которой заменена высокотемпературным теплообменником, расположенным в топке котла. Интеграция мусоросжигательной установки с газовой турбиной привела к росту к.п.д. брутто от 25% до 37–41%. Схема предвключения газовой турбины со сбросом отходящих газов в топку мусоросжигателя и внешним пароперегревом показала наилучшее соотношение между потреблением природного газа, общим к.п.д. и удельной поверхностью теплообмена. Использование отходящих газов турбины в качестве подогретого воздуха горения уменьшило потери эксергии в процессе сжигания на 9%.

Расчеты комбинированного цикла, использующего газовую турбину с внешним теплообменником, показали к.п.д. около 47%. Дополнительное сжигание природного газа для поднятия верхней температуры газотурбинного цикла позволяет уменьшить поверхность дорогостоящего высокотемпературного теплообменника.

В восьмой главе цикл газовой турбины с частичной оксидацией топлива, включенный в программу NECT, сравнивается с циклом ГТУ с промежуточным нагревом. Обнаружены сходные показатели этих циклов, однако некоторые особенности цикла с частичной оксидацией заслуживают дополнительного внимания: во-первых, в связи с малой величиной коэффициента избытка воздуха уменьшается выброс NO_X; во-вторых, возможна комбинированная выработка электричества и синтез-газа с высоким эксергетическим к.п.д.

Настоящая работа выразилась в двух проектах, которые по заказу Novem, выполняются Нидерландским Центром по Энергетическим Исследованиям ECN и унивеситетом Твенте. Первый проект направлен на разработку и внедрение нескольких комбинированных и усовершенствованных циклов, рассмотренных во данной диссертации. Второй проект должен дать техническое обоснование газотурбинного цикла с частичной оксидацей. Проекты включают в себя оптимизацию, параметрический и сравнительный анализ, экпериментальную часть и предварительный технико-экономические расчеты. Для реализации этих проектов будет создана рабочая группа, состоящая из ECN, университета Твенте, инженерной компании и производителей оборудования.

Preface

I am grateful to many people in writing this thesis.

First, my gratitude is expressed to the late Prof. R.E. Leshiner, my teacher who stimulated my interest in thermodynamics and power technology with his excellent knowledge of the subject and a remarkable erudition.

Second, I wish to thank my promotor Prof. G.G. Hirs for allowing me to conduct this research at his Energy Technology chair at the University of Twente. He is the driving force behind the penetration of exergy concept not only in academic, but also in engineering and industrial communities in this country and abroad.

Prof. E.I. Iantovski is thanked for numerous suggestions he made to this thesis.

Novem, in the person of Dr. A.H.M. Kipperman, is acknowledged for permitting to publish parts of the report done on their behalf.

I would like to thank my former colleagues at Stork Engineers and Contractors (then Comprimo) in Amsterdam: Edwin Goudappel and Bas Kerkhof (currently with Tebodin BV) for supporting my exergy studies, and Piet Baaijens for his inexhaustible expertise.

Also my thanks go to all staff of Laboratory of Thermal Engineering. In spite of my rather short and intermittent stay in Enschede, it has been a great pleasure working with you. I thank my roommate, René Cornelissen, for fruitful discussions and his critical approach, which is not limited only by exergy analysis.

Much appreciation deserve the graduate students who contributed to this thesis. I owe a debt to Martijn Wagener, Pieter Jellema, Geert-Jan Wiechers, and Patrick Kers for their enthusiasm and devotion.

I wish to thank my colleagues at the Netherlands Energy Research Foundation ECN for a pleasant and stimulating atmosphere. The management of the business unit ECN Fuels, Conversion & Environment is thanked for permitting me an absence leave and the possibility to complete this thesis.

My appreciation is due to Iris Mayer for her editorial support.

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Introduction

0.1 Advanced and combined cycles

The development of some technologies often follows an S-curve. At the moment of introduction the technology performs poorly and cannot compete with other types. Then comes a break-through period: performance improves very rapidly, costs drop due to the mass production, and more investments are attracted for further research and development. After awhile, the technology enters the saturation phase, where growth slows down and progress is achieved only by increasing the unit size (Fig. 0.1).

The development of the gas turbine aptly illustrates this principle. In the beginning, gas turbines were inefficient, bulky, and unreliable engines. In order to improve their performance, modifications, such as reheat, intercooling, or recuperation, were applied. Another approach was combining the turbines with other, more developed cycles, such as the Rankine cycle. But as soon as the break-through period was achieved in the 1960s, and industrial gas turbines could utilize the achievements obtained in jet engine technology, the use of advanced/combined schemes was no longer required for efficient plant operation. The open-cycle gas turbines offered low capital costs, compactness, and efficiency close to that of the steam plants. Nevertheless, after the oil crisis in the 1970s the efficiency of power plants became the top priority, and combined-cycle plants, first in the form of existing steam plant repowering, and later, as specially-designed gas-and-steam turbine plants, have become a common power plant configuration.

Also, advances within the gas turbine cycle have been reconsidered: in 1948 ABB utilized reheat in a gas turbine power plant built in Beznau, Switzerland (ABB, 1994). The plant with multiple shafts had high efficiency for that time, but low specific power. Forty years later, the reheat concept has been implemented in the ABB GT24/26 series turbine. Recuperation, which was used in the 1940s and 1950s to increase engine efficiency, is now employed in the Rolls Royce/Westinghouse WR21 turbine (Shepard et al., 1995).

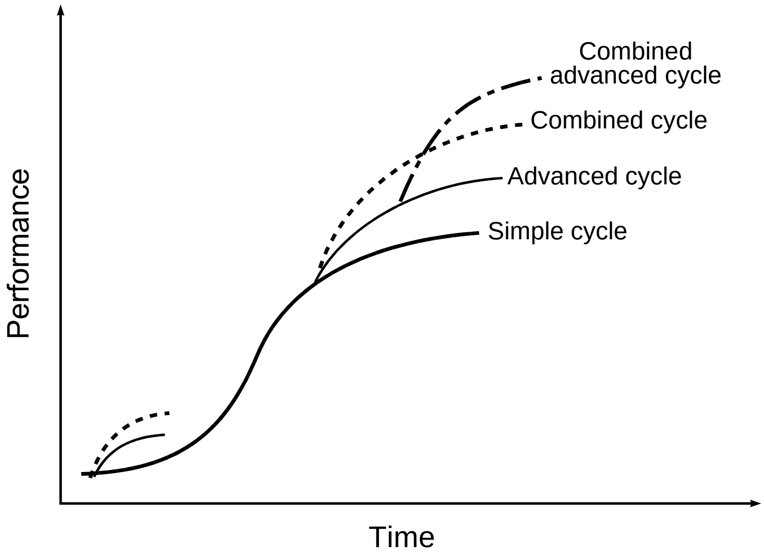

Fig. 0.1. Technology development paths.

Meanwhile, developments in direct-fired steam plants with supercritical parameters have resulted in efficiencies approaching 50%. The best simple-cycle gas turbine offers a value of 42%. Combined together, these technologies generate electricity with up to 59% efficiency. However, combination and integration of two cycles may not always be beneficial. In some cases, a separate generation scheme, especially when it involves the production of two different products (e.g. power and heat), or the use of two different fuels, can provide better performance. The gains and losses of combination/integration should be evaluated as shown in Fig. 0.2. To maintain a performance gain, developments in combined-cycle technology should proceed in the direction of the advance vector, which is perpendicular to the the critical line.

0.2 The New Energy Conversion Technologies programme

Identification and development of advanced and combined cycles is one of the topics of the New Energy Conversion Technologies (NECT) programme, which has been initiated by the Netherlands Agency for Energy and Environment (Novem) in order to identify and to develop the new energy conversion technologies and systems for distributed power generation applications (Novem, 1993). The programme's scope includes projects in the field of research and development, feasibility studies, experiments, and demonstration. The programme consists of four parts: (1) scouting study, (2) combustion, (3) new combinations, and (4) cogeneration. The objective of Part 3 is as follows: to identify and to develop to market introduction new and advanced energy conversion technologies which can provide higher efficiency and lower emissions than existing, competing technologies.

The realization of this part began with an inventory of basic thermodynamic cycles and their combinations, based on existing energy conversion technologies, and feasible within 5–10 years. Of the known energy conversion routes (as outlined by Rogers and

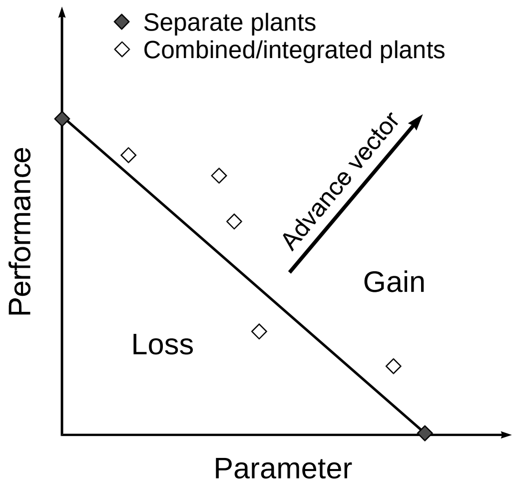

Fig. 0.2. Combined/integrated plants versus separate plants.

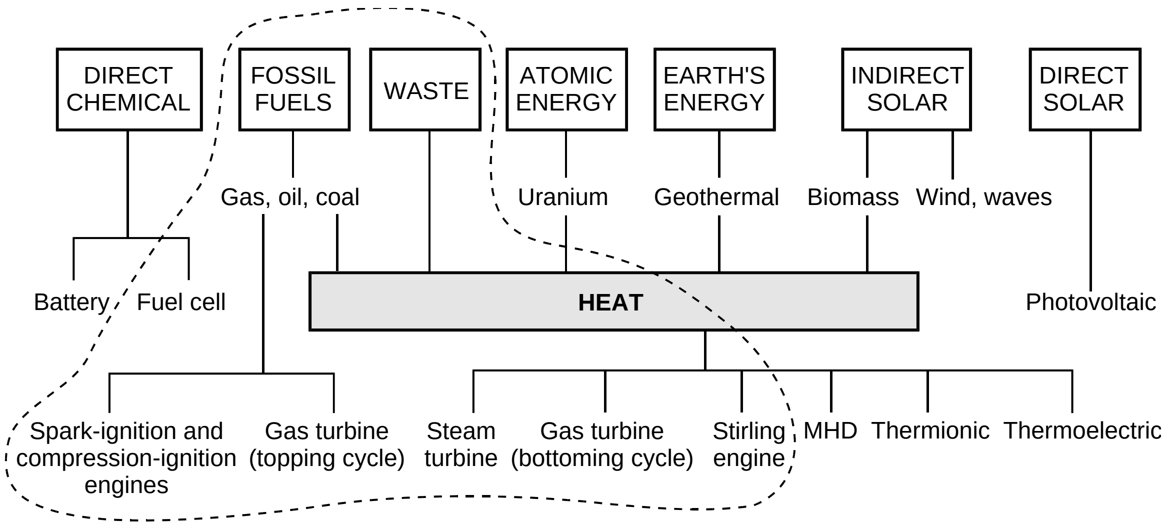

Fig. 0.3. Energy resources and energy conversion technologies.

Mayhew, 1991), a working area was defined, as shown by the dotted line in Fig. 0.3. The Earth's energy, solar and atomic energy sources were excluded from consideration.

The combinations were assessed on the basis of several criteria, the most important of which are efficiency, emissions, and feasibility. A reference system was set for a comparative analysis. A system with the highest efficiency was chosen as the reference. The systems were presented in a two-dimensional matrix, where they were combined according to their thermodynamic parameters: the high temperature cycles were regarded as the most suitable for topping applications, and the low temperature cycles for bottoming. Several combined and advanced cycles were selected for a detailed performance analysis. While performance of power production is easily assessed by, for example, an LHV efficiency, in the case of cogeneration of heat and power, other performance parameters are required.

0.3 Exergy

An objective criteria for assessment offers exergy efficiency. Exergy, also known as availability, is a measure of the maximum useful work that can be obtained when a system is brought to a state of equilibrium with the environment in reversible processes. Due to the irreversibility of thermal processes, the work obtained is always less than the maximum work. Hence, by analyzing work loss within a system, imperfections can be pinpointed and quantified, and possible improvements suggested. Also, different sorts of energy can be directly compared in exergetic terms. Grassmann diagrams, on which the width of a flow line is determined by its exergy value, are used to illustrate the exergy flow across a system.

Exergy method can be applied for the analysis of thermal, chemical, and metallurgical plants (Brodyanski, 1973; Kotas, 1985; Szargut, 1988), technological chains of processes (Szargut, 1991), the life cycle of a product (Cornelissen, 1997), or a whole country (Wall, 1987). Combined with economic analysis, exergy is used in thermoeconomic studies (Tsatsaronis and Winhold, 1985; Valero et al., 1986; Bejan et al., 1996), and to evaluate labour and economic trends (Bandura and Brodyanski, 1996). New methods of exergy analysis have been introduced recently, such as graphic exergy analysis with the use of the energy-utilization diagram (Ishida, 1982), exergy equipartitioning (Tondeur and Kvaalen, 1987) and exergy load redistribution methods (Sorin and Paris, 1995).

0.4 Scope of the thesis

Chapter 1 discusses the basic thermodynamic cycles (Carnot, Rankine, Joule-Brayton, Otto, Diesel, and Stirling cycles) and fuel cells. In Chapter 2 their possibilities and performance in combination are considered. Advances within the basic cycles are reviewed in Chapter 3. Combined generation of heat and power is discussed in Chapter 4.

Of the cycles reviewed in Chapters 1–3, a number are analyzed in more detail. The Joule/Rankine combined cycle is addressed in the form of an externally-fired combined cycle (Chapter 5) and in an integrated scheme of a gas turbine and a waste incinerator (Chapter 6). The Joule/Joule cycle (a gas turbine with an air bottoming cycle) forms the topic of Chapter 7. An advanced gas turbine cycle with partial oxidation is discussed in Chapter 8.

The performance specifications of the analyzed systems are based on manufacturers' claims, developers' evaluations, computer simulations, or are estimated. All efficiency values quoted are based on the lower heating value (LHV) of the fuel, unless otherwise stated.

Materials from the following, previously published papers, were used in this thesis.

1. Korobitsyn, M.A. and Hirs, G.G., "Analysis of Cogeneration Alternatives", ASME Paper 95-CTP-11, 1995.

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Chapter 1
Basic Cycles

Abstract

This chapter gives an overview of basic thermodynamic cycles and fuel cells. The operating parameters, efficiency and power range are outlined for each cycle, and their advantages and disadvantages are discussed. Ways of improving performance are also mentioned, while more advanced developments are covered in Chapter 3.

1.1 Carnot cycle

All the cycles reviewed in this chapter are heat engines, except for the fuel cells. A heat engine is an apparatus that produces work operating between a high-temperature reservoir (source), and a low-temperature reservoir (sink). The overall thermal efficiency of the engine is the proportion of the heat supplied which is converted into mechanical work. A corollary of the Second Law of thermodynamics proves that a heat engine possesses the highest efficiency when operated in the Carnot cycle.

In the Carnot ideal cycle (Fig. 1.1a) heat is added at a constant upper temperature and rejected at a constant lower temperature. It consists of two isothermal processes, 1–2 and 3–4, and two isentropic processes, 2–3 and 4–1. The cycle can be performed on the working fluid within a two-phase region under the saturation line. In the heating process, liquid from state 1 is brought to saturated vapour state 2. The vapour expands isentropically to state 3, and condensation with heat rejection occurs along line 3–4. At state 4, isentropic compression begins, and the fluid is brought back to state 1. Using the general expression for efficiency, the Carnot cycle efficiency can be described as

$$\eta_{CARNOT} = \frac{W}{Q_{ADDED}} = \frac{Q_{ADDED} - Q_{REJECTED}}{Q_{ADDED}} = \frac{T_{MAX}\Delta S - T_{MIN}\Delta S}{T_{MAX}\Delta S} = \frac{T_{MAX} - T_{MIN}}{T_{MAX}} \qquad (1.1)$$

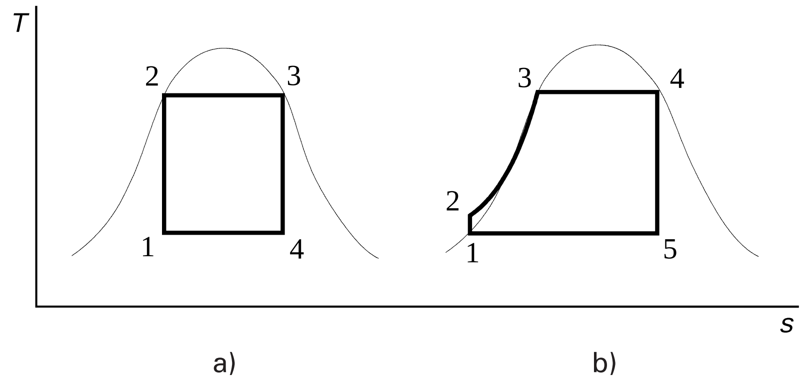

Fig. 1.1. The Carnot and Rankine cycle.

It is evident that the greater the temperature difference the more efficient the cycle. The minimum temperature in a real process depends on the heat sink temperature and the heat transfer characteristics of the condenser unit. A natural heat sink possesses an ambient temperature which is assumed to be 15 °C. The temperature difference in the condenser can vary from 10–15 K for a water-cooled condenser and 25–40 K for an air-cooled unit. Using water as the working fluid, this results in a practical condensing temperature of 25–55 °C at the condensing pressure of 0.032 to 0.16 bar.

The maximum temperature has thermodynamic constraints, such as the critical temperature and pressure. The highest steam temperature in the wet region is the critical temperature of 374 °C. By this means, the efficiency of a Carnot plant with a boiler pressure of 40 bar will lie in the range of 37–43%, depending on the maximum temperature.

1.2 Rankine cycle

Practical difficulties with the compression of a wet vapour in the Carnot cycle (high volumetric flow, non-homogeneous medium) lead to the Rankine cycle, where the working fluid is condensed completely (line 5–1 in Fig. 1.1b) and brought to the boiler pressure by a feed water pump (line 1–2). The work done by the pump on the working fluid is much less than that of the compressor: at low pressures it amounts to 0.3–0.5% of the total cycle work and can often be disregarded. Superheat can be introduced into the cycle in order to raise the maximum cycle temperature, thus improving the efficiency of the cycle (line 4–5 in Fig. 1.2a).

A further improvement can achieved by adding a reheat stage, where steam expands to an intermediate pressure (line 5–a), and then is passed back to the boiler to reach the initial temperature (line a–b in Fig. 1.2b). This arrangement covers even larger area on the T–s diagram, meaning a higher work output. The ideal efficiency that a Rankine plant with reheat can reach is about 55%.

The cycle also gains in performance when the boiler feed water is preheated by the steam extracted from the steam turbine (line c–d in Fig. 1.2c). The average temperature of the heat addition is then higher than in the conventional Rankine cycle, and, therefore, efficiency is increased.

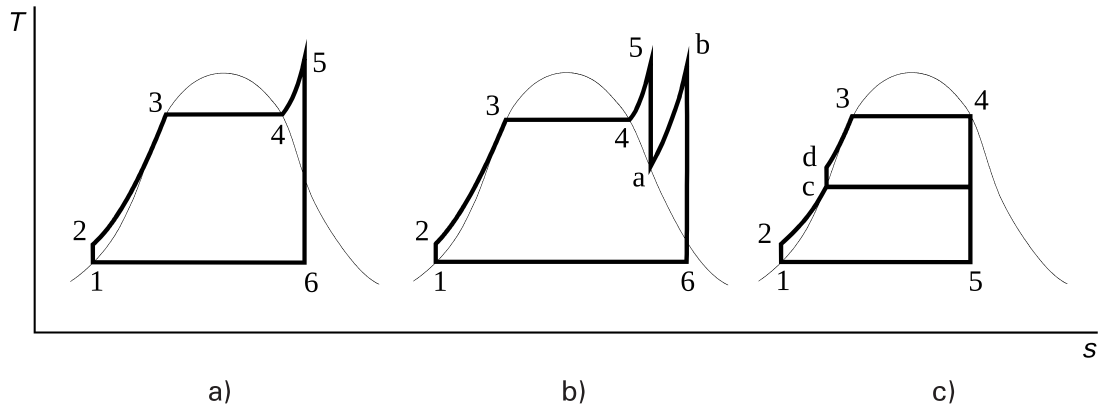

Fig. 1.2. Advanced Rankine cycles:
(a) with superheat;
(b) with superheat and reheat;
(c) with regeneration.

The temperature of the flue gases in the steam boiler approaches 1800 °C, while that of the steam is typically just 500–600 °C. Extremely irreversible heat exchange caused by this large temperature difference indicates an imperfection of this cycle. Modern supercritical Rankine plants with reheat and regeneration that operate at temperatures of 600 °C and pressures of 350 bar obtain an efficiency not higher than 48–49% (Rogers and Mayhew, 1992).

Further developments of the cycle are expected by raising the metallurgic limit with the use of high-temperature materials, or by choosing other working fluids. Using advanced materials, a superheat temperature of 816 °C was obtained at a 4 MW demonstration plant (Rice, 1997). Regarding the other possibility, various media other than steam have been proposed for the Rankine cycle. The choice of a medium depends on the temperature level. At high temperatures (above 500 °C) a fluid with a high critical temperature is required, such as mercury or potassium vapour. At the medium-temperature level, steam remains the most suitable fluid, since it is an inexpensive, readily available, non-toxic and well-known medium. The organic Rankine cycle, based on chlorofluorocarbons (CFC) or other fluids with low critical temperatures, proves to be a workable one at a lower temperature. The organic cycle can be used for waste heat recovery. Regarding the devastating impact of chlorofluorocarbons on the ozone layer, other refrigerants are preferred for use in the organic Rankine cycle, as for example, dimethyl ether (Kustrin et al., 1994). A mixture of more than two components was also suggested in the so-called SMR-cycle (Verschoot and Brouwer, 1994). Further in the text steam will be regarded as a working fluid in the Rankine cycle, unless otherwise stated.

1.3 Kalina cycle

The use of the Rankine cycle in bottoming applications has a serious drawback due to the constant temperature of vaporization. A mixture of fluids has been proposed to substitute steam in a Rankine cycle, such as the Kalina cycle, where a mixed working

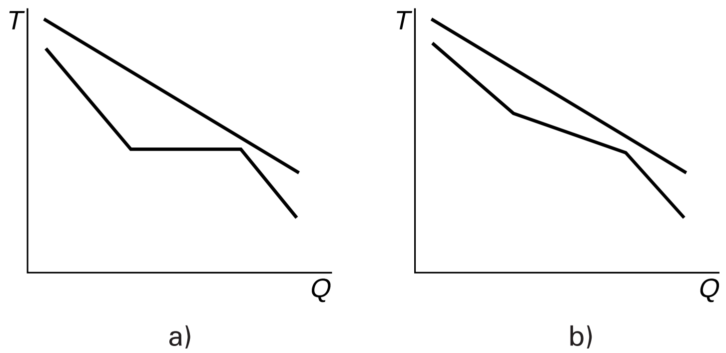

Fig. 1.3. Heat utilization line in: (a) the Rankine and (b) the Kalina cycle.

fluid of variable composition is used to provide a better match between the temperatures of the hot and cold flows (Kalina, 1984). The composition of the fluid is changed in the cycle at different points. In most Kalina cycle studies a mixture of water and ammonia has been used. The ammonia in the mixture begins to vapour first, and as it boils off, its concentration decreases, and the boiling temperature of the mixture increases. This reduces the temperature mismatch between the topper's waste heat and the fluid in the boiler, and allows an efficiency rise in the bottoming cycle. A diagram of heat utilization in the Rankine cycle (Fig. 1.3a) versus the Kalina cycle (Fig. 1.3b) clearly shows the advantage of the latter.

A basic Kalina cycle consists of a heat recovery vapour generator (HRVG), a steam-ammonia turbine, and the distillation condensation subsystem (DCSS), as shown in Fig. 1.4. In the DCSS the stream from the turbine is cooled in the recuperator, and then mixed with a lean solution of ammonia in order to raise the condensing temperature. The resulting basic solution is condensed in the absorber and brought to the recuperator under pressure. Part of the flow is sent to dilute the ammonia-rich stream coming from the separator. The main flow passes the recuperator and is flashed in the separator. The vapour is mixed with the basic solution, condensed and pressurized before entering the vapour generator.

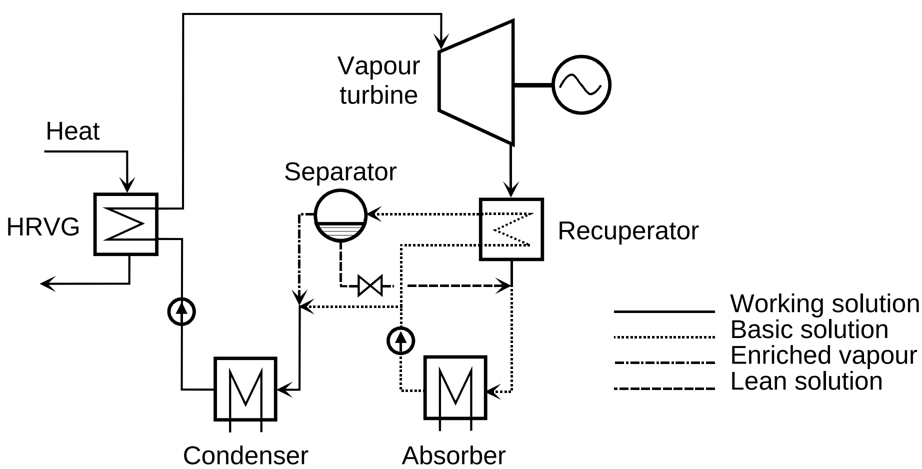

Fig. 1.4. Flow scheme of the Kalina cycle.

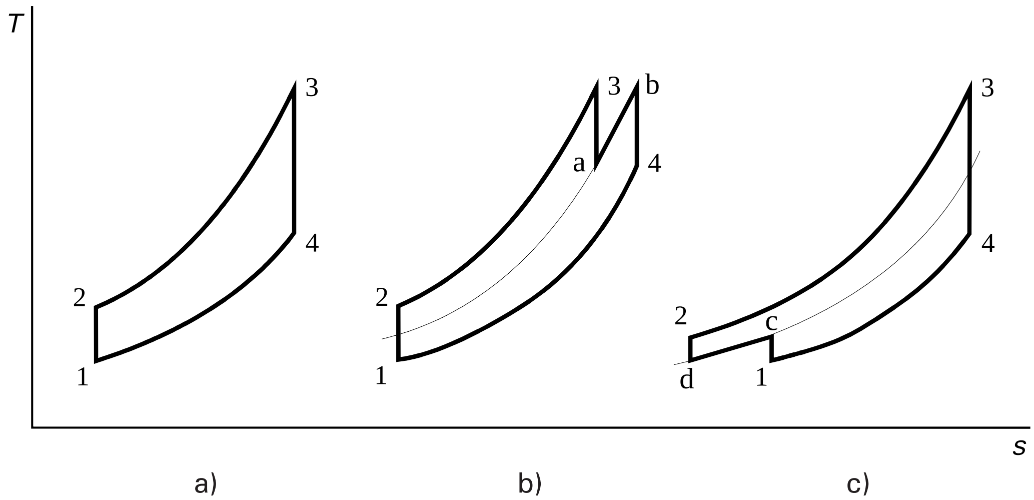

Fig. 1.5. Basic and advanced Joule cycles:
(a) basic cycle;
(b) with reheat;
(c) with intercooling

Several investigations showed a favourable performance of the Kalina bottoming with various topping cycles, such as the gas turbine (Marston and Hyre, 1995), direct-fired power plants (Kalina, 1991), and in geothermal (Marston and Sanyal, 1994), solar and low temperature heat recovery applications (Olsson et al., 1994), as well as for cogeneration (Olsson et al., 1991). Modifications within the Kalina cycle have also been analyzed (Glasare et al., 1993).

1.4 Joule-Brayton cycle

The Joule, or Brayton cycle is a gas power cycle that consists of the following processes: isentropic compression of the working fluid to the maximum working pressure (line 1–2 in Fig. 1.5a), heat addition at constant pressure along line 2–3, isentropic expansion to initial pressure (process 3–4), and, finally, the heat release at constant pressure (line 4–1). The Joule cycle is realized in a gas turbine plant, where the fuel is burned directly in the working fluid, which eliminates the heat transfer area. The waste heat is rejected into the atmosphere, which makes the cooler unnecessary. Both these factors make the gas turbine plant compact and less expensive than a steam plant with the same power output.

Although the exhaust is released at temperature of 400 to 600 °C and represents appreciable energy loss, modern gas turbines offer high efficiency (up to 42%) and a considerable unit power output (up to 270 MW). Some typical modern gas turbines are listed in Table 1.1 (Gas Turbine World Handbook, 1997).

The high air excess ratio needed for turbine cooling is responsible for a relatively high oxygen content in the exhaust gases (14–16%) and a larger mass flow than in a comparable gas engine. These specific features of the Joule cycle can be utilized by means of supplementary firing and waste heat recovery.

Table 1.1. Performance specifications of some modern gas turbines.

Turbine	Manufacturer	Power, MW	Efficiency, %
LM1600PA	General Electric	13.8	35.5
OGT15000	Mashprom/Orenda	17.1	34.2
LM2500PE	General Electric	22.8	36.8
GT10	ABB	24.6	34.2
FT8	United Technologies	25.5	38.1
RB211	Rolls-Royce	27.2	35.8
LM5000PC	General Electric	34.5	37.2
LM6000PC	General Electric	43.9	41.9
Trent	United Technologies	51.2	41.6
PG6101FA	General Electric	70.2	34.2
701DA	Westinghouse	136.9	34.0
PG9311FA	General Electric	226.0	35.7
701F2	Westinghouse	253.7	37.1
V94.3A	Siemens	255.0	38.5
GT26	ABB	265.0	38.5
701G1	Westinghouse	271.0	38.7

One important disadvantage is that a gas turbine does not perform well in part-load operation. For example, at 50% load, the gas turbine achieves around 75% of the full-load efficiency, and at 30% load this drops to 50% of the nominal efficiency (Kehlhofer, 1991). Therefore, arrangements, such as the controlled inlet guide vanes and multi-shaft designs, are employed to improve the part-load performance.

Other modifications of the cycle include reheat, intecoling and recuperation. The expansion work can be increased by means of reheating (Fig. 1.5b); moreover, this makes it possible to provide full-load efficiency within a broader load range by varying reheat fuel flow. Because of the increased specific work output due to reheat, the plant becomes compacter: for example, the ABB GT26 is three times lighter than the

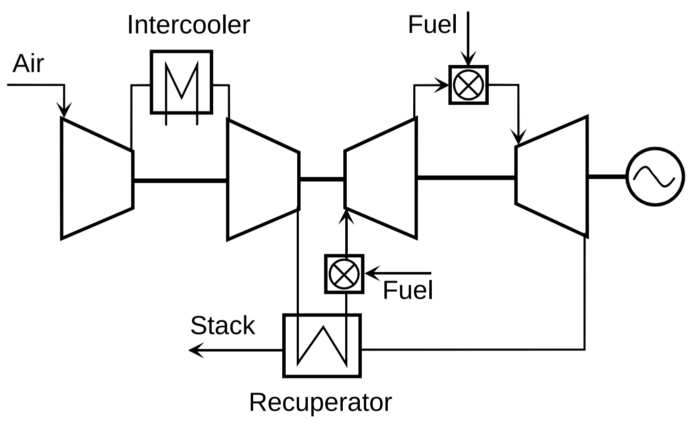

Fig. 1.6. The Heron gas turbine.

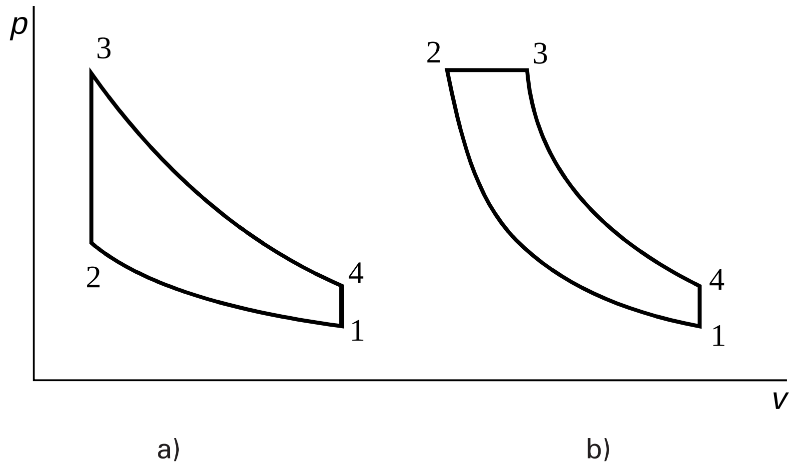

Fig. 1.7. Otto and Diesel cycles.

General Electric PG9311FA with approximately the same power output. Another technique to increase the specific work output is intercooling, which diminishes the work required by the compressor (Fig. 1.5c). The compressor outlet air becomes colder and, if air cooling is applied, this allows higher turbine inlet temperatures.

Efficiency can be raised when recuperation of the exhaust flow is employed. This concept is used in several gas turbines, such as the 1.4 MW Heron (Fig. 1.6) described by Poolman (1993), the 21 MW Westinghouse/Rolls-Royce WR–21 (Shepard et al., 1995), or ATS series Solar turbines in the 1 to 25 MW size range (Stambler, 1995).

The use of recuperation is limited, however, by the compressor outlet temperature T_2, and to decrease it intercooling may be applied. An in-depth analysis of the regenerated gas turbine cycle can be found in Beck and Wilson (1996).

The recuperated turbines are expected to obtain efficiencies from 39% (WR–21) to 43% (Heron) and even as much as 50% (the ATS Solar prototype expected to be available by the year 2000), which are significantly higher compared to 25–32% for other turbines of this power range size.

1.5 Otto and Diesel cycles

In the Otto and Diesel cycles the work is obtained in a sequence of non-flow processes. The Otto cycle (the spark-ignition cycle) consists of isentropic compression, constant volume heat addition, isentropic expansion of the heated gas to produce work, and constant volume heat rejection to the atmosphere (Fig. 1.7a).

The Diesel compression-ignition cycle resembles the Otto cycle, but heat addition occurs here instead at a constant pressure (line 2–3 in Fig. 1.7b). Since both engines operate in a reciprocating manner, so that the cylinder is exposed to high temperature for only a short period during the whole cycle, the maximum permissible temperature in the cycle can be as high as 2500 °C. The engines have a rather high efficiency and provide good performance at part-load conditions. Gas and Diesel engines are commonly used to generate electric power at power ranges below

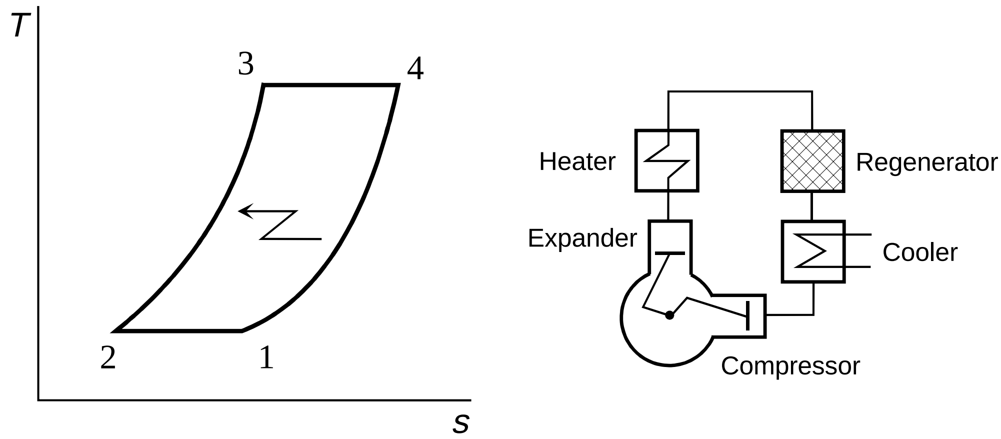

Fig. 1.8. The Stirling cycle.

70 MW with efficiency up to 50%, whereas the gasoline engines have smaller power output (up to 500 kW) and lower efficiency (Anonymous, 1995; Diesel and Gas Turbine Worldwide Handbook, 1997).

In Otto (Diesel) cycle waste heat is generated at two temperature levels: by a low-temperature flow of coolant (90–125 °C) and by a medium-temperature flow of exhaust gas (200–400 °C). The latter makes it possible to incorporate a heat recovery arrangement as a bottoming cycle.

The main disadvantage of reciprocating engines is the slow work rate, which makes the equipment bulky and heavy when large power output is required. For example, the 51 MW MAN K 98 engine weights more than 1500 tons, while the ABB GT8C gas turbine with the same output just 95 tons. (Gas Turbine World Handbook, 1997).

Further improvements of Otto (Diesel) engine can be achieved by increasing the compression ratio, turbocharging and intercooling.

1.6 Stirling cycle

A cycle consisting of two constant volume processes and two isothermals is called the Stirling cycle (Fig. 1.8). Heat is added from an external source at constant temperature in process 3–4, and rejected isothermally in process 1–2. Heat exchange occurs between processes 2–3 and 4–1. Therefore, the cycle possesses the Carnot efficiency $(T_{ADDED} - T_{REJECTED})/T_{ADDED}$.

One end of the regenerator must be continuously maintained at the upper temperature which is only limited by metallurgical restrictions. This makes the engine suitable for heat recovery over a broad range of temperatures. Along with the crankshaft arrangement, as shown in Fig. 1.8, a free-piston design can be impemented, allowing a compact design and low noise level. Different gases may be employed as the working fluid. Of air, methane, helium and hydrogen, the latter offers the highest power density, which is important in mobile applications. For stationary applications the engine may use helium or less costly methane. Just as an Otto (Diesel) engine, the Stirling engine generates waste heat at two levels: the lower one of the cooling circuit,

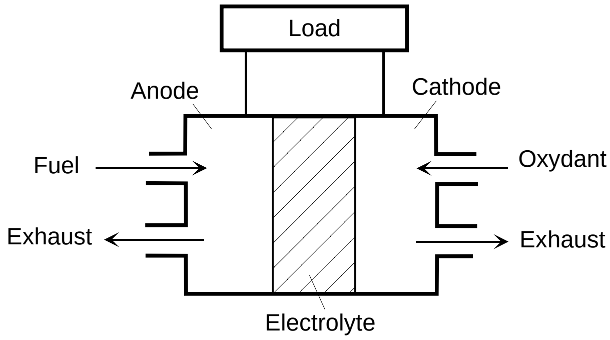

Fig. 1.9. Schematic of a fuel cell.

and the higher-temperature one of the heater's exhaust. An important advantage of the Stirling cycle is a flat part-load characteristic: the engine maintains nominal efficiency even at 50% load (Penswick, 1997).

However, modern Stirling engines have not reached such a mature phase of development as Otto or Joule cycles, and commercial units are only available at a relatively low power level of several hundred watts.

In spite of difficulties in designing an efficient and compact heat exchanger, this externally-fired engine can provide an efficiency comparable to that of the Otto (Diesel) cycle and much lower emissions. An efficient coal-fired plant can be implemented using the Stirling cycle for power systems less than 10 MW, where the Rankine cycle's hardware becomes costly and inefficient. Besides, the reversed Stirling cycle has been successfully used for refrigerating, as for example, in an air liquefaction plant.

1.7 Ericsson cycle

To improve the efficiency of the Joule cycle the infinite number of intercoolers and reheat stages together with a heat exchanger between lower- and higher-pressure streams can be applied to the cycle to provide heat addition and rejection at constant temperature, thus approaching the Carnot cycle. Because an infinite number of the intercoolers seems unrealistic, in practice, it is seldom greater than two. The same holds true for the reheat stages. An arrangement with water injection at each compressor stage can approximate compression at a constant temperature (see Section 3.4.3). Nevertheless, the Ericsson cycle has never gained acceptance as a heat engine.

1.8 Fuel cells

Fuel cells convert the chemical energy of the fuel directly into electricity, and are not restricted by the Carnot efficiency. The chemical reaction is nearly reversible, and therefore, this form of energy conversion can be considered the most effective one.

Table 1.2. Current and projected performance specifications of fuel cells (from Van Barkel et al., 1993)

Type of cell	Fuel	Power range MW	Efficiency present, %	Efficiency in 2015, %
Phosphoric Acid	Methane, methanol	0.1–5	40–45	45–50
Molten Carbonate	Methane, coal gas	1–100	45–55	55–60
Solid Oxide	Methane, coal gas	1–100	45–50	55–65

Type of cell	Lifetime present, hrs	Lifetime in 2015, hrs	Capital cost present, NLG/kW	Capital cost in 2015, NLG/kW
Phosphoric Acid	35 000	80 000	4200	2300
Molten Carbonate	15 000	35 000	3900	1800
Solid Oxide	22 000	35 000	3900	1800

A fuel cell consists of electrodes put into an electrolyte. The fuel (hydrogen, methane) is brought to the anode, and the oxidant (air, oxygen) to the cathode. In the ionizing reactions electric current is generated (Fig. 1.9). Different fuels such as hydrogen, hydrocarbons, or coal-derived fuel can be used in a fuel cell. However, due to the high degree of irreversibilities of the anodic oxidation reactions of hydrocarbons, hydrogen has proved to be the most suitable fuel (Srinivasan et al., 1993). Other fuels (methane, methanol) are processed in a reforming reaction to form hydrogen and carbon monoxide.

The type of the electrolyte determines the type of fuel cell. The most developed cells types are the Alkaline Fuel Cell (AFC), Solid Polymer Fuel Cell (SPFC), Phosphoric Acid Fuel Cell (PAFC), Molten Carbonate Fuel Cell (MCFC), and Solid Oxide Fuel Cell (SOFC). Basically, these can be divided into two groups: low-temperature (AFC, SPFC, PAFC) and high-temperature (MCFC and SOFC) fuel cells. The former operate at a temperature under 200 °C, the latter above 600 °C. This determines the application of a fuel cell in a combined cycle; the low-temperature systems can be used as a bottoming cycle, and the high-temperature ones as a topping one. AFC and SPFC operate at rather low temperatures of 60–100 °C and are best suited for transport or domestic applications.

While the low-temperature cells are fed with hydrogen which is produced externally, the high-temperature ones can supply enough heat for internal reforming, thus utilizing the waste heat.

Fuel cells have the useful characteristics that their efficiency increases at part-load and that the efficiency is independent of scale because of their modular construction. Low maintenance, low noise and vibration, very low pollutant emissions, and minimal water use are among other important advantages, which permit a fuel cell plant to be located in urban areas.

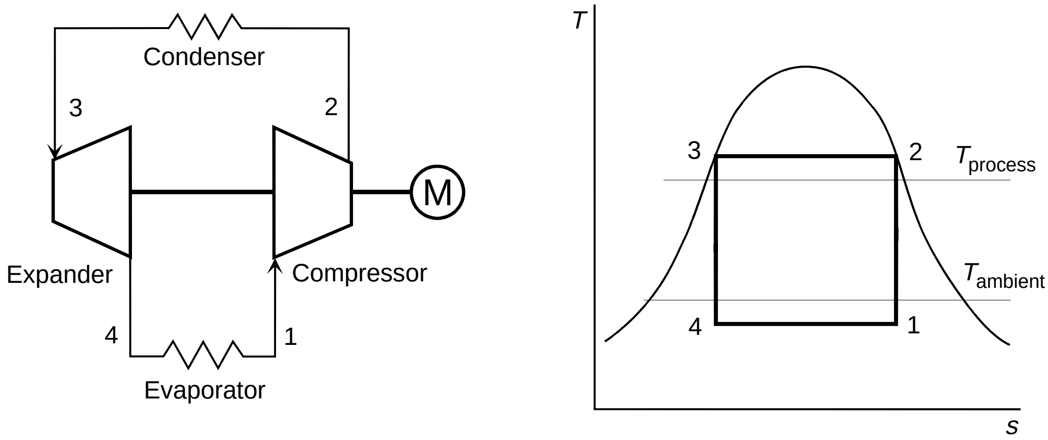

Fig. 1.10. A Carnot cycle-based heat pump.

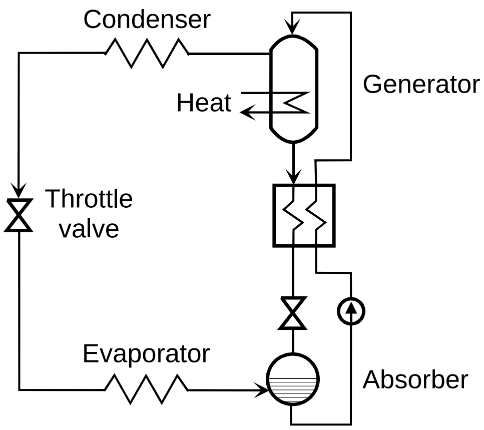

Fig. 1.11. An absorption heat pump.

Considerable improvements in power density and operational lifetime are expected in the future development of fuel cells. Capital costs around 3000 NLG/kW for the year 2000 and 1500–2000 NLG/kW in 2030 have been given; these figures are projected to be greatly reduced with increasing production volume (Van Brakel et al., 1993; Barclay, 1995; Hirschenhofer and McClelland, 1995). The results of the study by Van Brakel et al. are presented in Table 2.

A 2 MW demonstration direct carbonate fuel cell plant has been opened in Santa Clara. The plant is designed to operate with an overall efficiency of 50% (Anonymous, 1996).

1.9 Heat pumps

Heat pumps use a reversed cycle (Carnot, Rankine, Joule or Stirling) to supply heat at an elevated temperature at the expense of work done on the working fluid. In a reversed Carnot cycle a wet vapour is compressed isentropically and passed to the

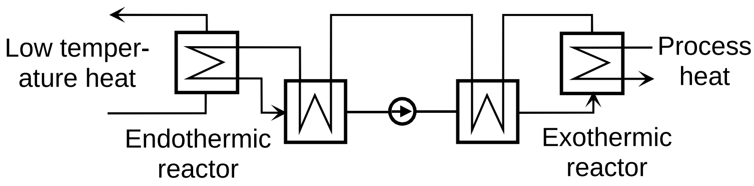

Fig. 1.12. A chemical heat pump.

condenser. Then the fluid is expanded to its original pressure and evaporated at constant pressure to the initial state (Fig. 1.10).

The ratio of output to input expresses the performance of a heat pump, namely heat supplied Q_{ADDED} to work input W. The coefficient of performance (COP) of a Carnot heat pump can be defined as

$$COP = \frac{Q_{ADDED}}{W} = \frac{Q_{ADDED}}{Q_{ADDED} - Q_{REJECTED}} = \frac{T_{MAX}}{T_{MAX} - T_{MIN}} \tag{1.2}$$

The amount of work is always less then the heat supplied, and COP is thus always greater than unity. It is also obvious that the smaller is temperature difference between the upper and the lower temperatures, the better the COP. A typical temperature range for a heat pump lies between 5 and 60 °C, therefore, the ideal COP of the Canot heat pump in this case is about 6.

Some simplification can be made in the expansion part when the expander is replaced by a throttle valve. This introduces irreversibilities in the cycle and diminishes the amount of heat extracted in the evaporator, but allows a compact construction.

The work done by the compressor can be considerably reduced when the working vapour is first dissolved in a liquid before compression. Then, the vapour can be drawn off, condensed, expanded, evaporated, and dissolved again. This principle is used in an absorption heat pump (Fig. 1.11).

The pump work required to bring a liquid to the desired pressure is much less than the work required to compress a vapour, but on the other hand, there is some heat needed in the generator to drive off the vapour from the solution.

The waste heat can be also brought to a higher temperature by a chemical heat pump, where a combination of endothermic and exothermic reactions is employed (Fig. 1.12). A number of reactions have been proposed: dehydrogenation-hydrogenation, hydrolysis-dehydrolysis (Murata et al., 1993; Yamashita and Saito, 1993), depolymerization-polymerization (Kawasaki et al., 1995), etc. A concept of CO_2 conversion chemical heat pump using activated ceramics was described by Tamaura and Tsuji (1993). A chemical heat pump enables heat transportation over long distances without the need for insulation due to a negligible amount of sensible heat in comparison to the latent heat contents of a fluid (Hasatani, 1995).

The heat pumps can be applied in virtually any cycle to supply heat at a low-temperature level for industrial or domestic needs.

1.10 Conclusions

The basic thermodynamic cycles can be summarized as follows:

The Rankine cycle (as a steam plant) is the most developed cycle covering the power range of 10–1000 MW, and offering a maximum efficiency at a level of 47%. It performs rather well in part load, but requires a bulky steam boiler, steam turbine and condenser with the associated cooling system. Due to intrinsic steam properties, a high operating temperature (up to 650 °C) inevitably involves the use of very high pressure (up to 350 bar), whereas the temperature is still much lower than that of the fuel flame (2000–2400 °C). Another drawback is a constant temperature of evaporation, which increases the mean temperature difference in the boiler. Nevertheless, again thanks to the water properties heat rejection can easily be obtained at a temperature close to ambient. Regarding fuels, any sorts and qualities are suitable for this cycle, since heat addition occurs from an external heat source.

The Kalina cycle offers a variable-temperature profile of evaporation, and thus a higher efficiency without the need for costly multiple-pressure and reheat arrangements. The working parameters of this cycle are close to that of the Rankine's. The Kalina cycle, however, is still in development and only one demonstration plant has been built so far.

The Joule-Brayton cycle operates at temperatures up to 1500 °C and moderate pressures up to 40 bar. It covers a power range between 0.5 and 280 MW and operates at best with an efficiency of 42%. The cycle is well-developed, but requires a high-quality fuel such as natural gas or oil distillate. Also, poor part-load operation of the gas turbine remains a major drawback.

The Otto (Diesel) represents another mature energy conversion technology of high efficiency (up to 50%) and a broad power range (up to 0.5 MW for gasoline engines, and up to 70 MW for gas and Diesel engines). Maximum cycle temperature can be as high as 2000 °C, but the power density is lower than that of the Joule cycle. The Otto (Diesel) engine has a good part-load performance.

The Stirling cycle is well-known, but nonetheless, has not yet been developed to the levels of the Otto (Diesel) or Joule cycle. It performs quite well within different temperature ranges (efficiency up to 45%) and in part load, but the power size of the engine is limited by several kilowatts. External firing makes it possible to use any fuel, to provide an easier emission control and operation at a low noise level.

The fuel cells are, just as the Stirling cycle, still in the development phase. They promise, however, high efficiency (better than 50%), a compact design, modularity, low emissions, good part-load performance, and the possibility of integration with other cycles. The projected power plant size may be as high as 100 MW.

The heat pump technology is reaching its mature state, and further implementation of heat pumps in industries and domestic application is expected. The heat pumps that employ thermodynamic cycles with a constant temperature of heat addition and rejection, such as the Rankine or Stirling cycles, offer better COP values.

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Chapter 2
Combined Cycles

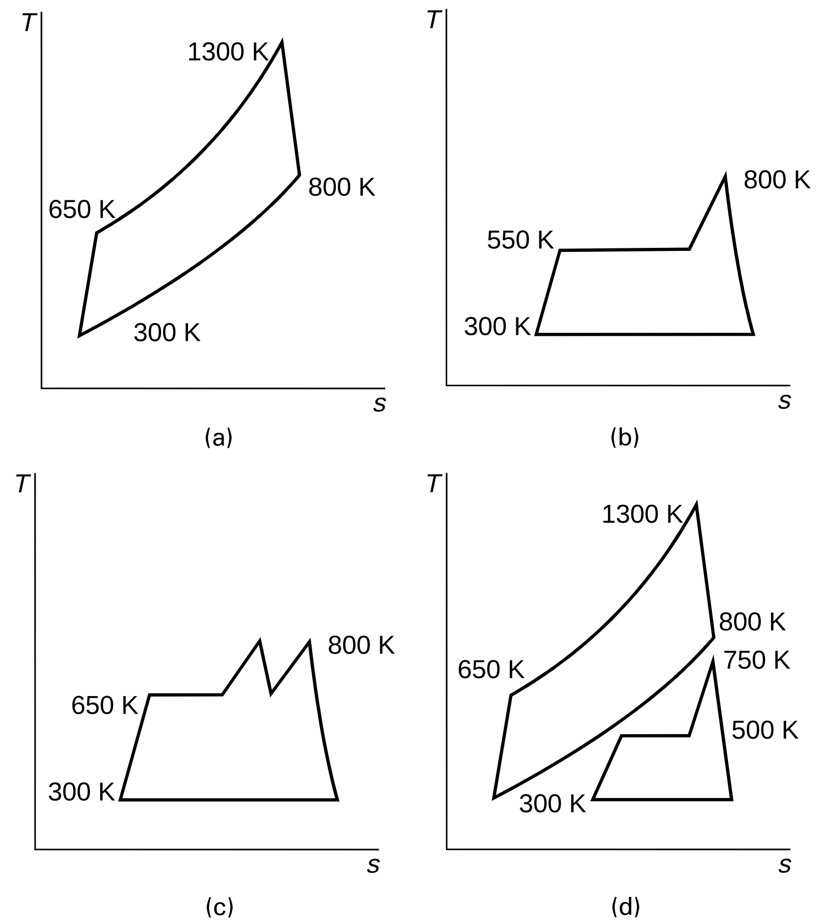
Abstract

This chapter describes possibilities and potentials in combining basic thermodynamic cycles in order to obtain better performance. Being divided into topping and bottoming cycles according to their temperature range, a number of possible combinations were defined. The resulting combined cycles were evaluated for potential performance gains and feasibility. Of the combined cycles, a gas turbine cycle with external firing and one with air bottoming were selected for the further consideration within the NECT programme. Performance of these cycles is analyzed in Chapters 5 and 7. Another combination, the integrated concept of a gas turbine and a waste incinerator, is addressed in Chapter 6.

2.1 Introduction

As shown in the previous chapter, some basic cycles have a maximum temperature closer to the fuel flame temperature, but a rather high temperature of heat rejection; others reject heat at nearly ambient temperature, but possess a moderate maximum temperature. Therefore, in order to approach higher efficiency, a combined cycle is required with a high-temperature topping cycle and a medium- or low-temperature bottoming cycle.

The advantage resulting from combining cycles was illustrated by Kehlhofer (1991) for three single cycles (a gas turbine plant, a steam boiler plant, and a steam plant with reheat) and a combined one. As seen on a T–s diagram, a gas turbine plant operates within a broad range of temperature, from 300 to 1300 K, but rejects heat at quite a high temperature level of 800 K (Fig. 2.1a). The steam turbine plants (Fig. 2.1b and Fig. 2.1c) reach a maximum working temperature of only 800 K, but their waste heat is released at a temperature close to that of the environment. The combination of these cycles (Fig. 2.1d) covers the largest area on the T–s diagram, and this results in



Plant	**(a)**	**(b)**	**(c)**	**(d)**
Average temperature of heat supplied, K	1000	600	680	1000
Average temperature of heat rejected, K	520	300	300	300
Carnot efficiency, %	48	50	56	70

Fig. 2.1. Comparison of single cycles with a combined cycle (after Kehlhofer, 1991).

the highest Carnot efficiency of 70%, whereas the best single-cycle efficiency approaches a value of 56%.

The drawback of one cycle may become a benefit when combined with another cycle. The high exhaust temperature of a gas turbine indicates its low efficiency, but is advantageous for the steam bottoming cycle. The highly efficient aero-derivative gas turbines provide worse performance in a combined-cycle configuration than industrial turbines with lower efficiency and higher exhaust temperatures. These considerations should be taken into account when searching for the optimal parameters of combined cycles. Comprehensive thermodynamic and economic analysis of combined cycles together with practical examples are given by Horlock (1992).

To determine what cycle is better suited for topping or bottoming application, they can be ranked according to their operating temperature range, as shown in Fig. 2.2.

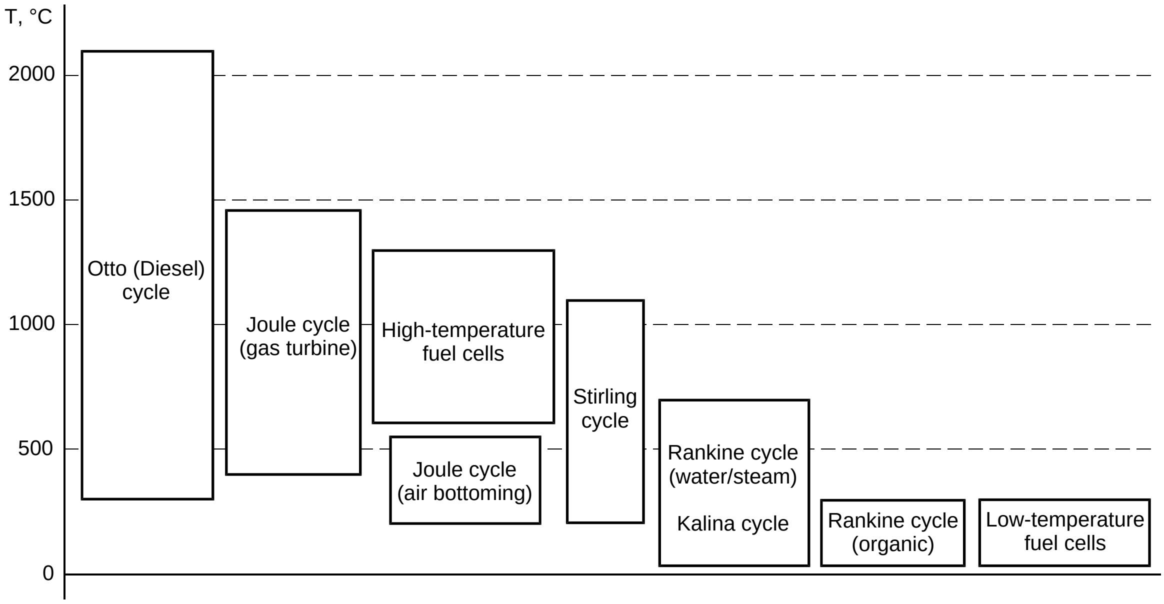


Fig. 2.2. Thermodynamic cycles arranged according to their temperature range.

High-temperature cycles are good candidates for topping, and medium- and low-temperature cycles for bottoming.

When these cycles are put into a matrix, a number of combinations can be defined (Table 2.1). The Rankine cycle is suitable both for topping and bottoming, just as the Stirling cycle. The Joule (gas turbine) cycle along with the Otto and Diesel cycles can be better applied as topping cycles. The Otto and Diesel cycles are comparable in terms of operating parameters, so they were regarded as the same cycle. For the reference fuel cell system a high-temperature scheme was chosen, such as the Solid Oxide Fuel Cell (SOFC), and the Molten Carbonate Fuel Cell (MCFC).

In some configurations, both cycles are integrated so that it is ambiguous which is the topper and which is the bottomer. This is the case, for example, in a Joule cycle with external firing, which makes use of a heat exchanger located in the furnace of a

Table 2.1. Combined cycles considered.

		Topping cycles			
		Rankine	**Otto/Diesel**	**Joule**	**Fuel cell**
Bottoming cycles	**Rankine**	◆	◆	◆	◆
	Kalina	◆	◆	◆	◆
	Joule		◆	◆	◆
	Otto/Diesel			◆	◆
	Stirling	◆	◆	◆	◆
	Fuel cell				◆
	Heat pump	◆	◆	◆	◆

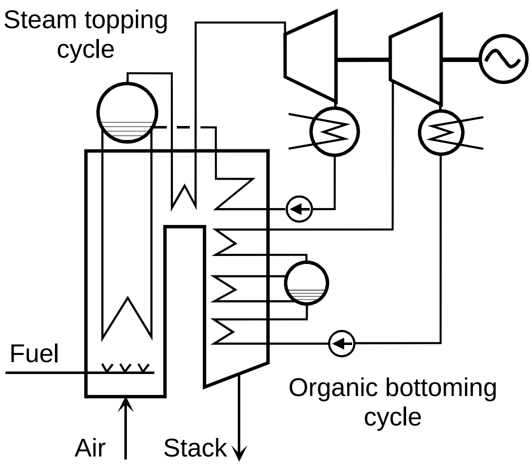


Fig. 2.3. A steam topping/organic bottoming (Rankine/Rankine) cycle .

steam boiler, or when a fuel cell is installed before the combustion chamber of a gas turbine. In the following description of combined cycles mostly dual cycles are considered, since the complexity of triple-cycle combinations, such as the potassium-steam-organic Rankine cycle, is undesirable.

2.2 Rankine/Rankine

For the topping cycle a working fluid should be selected with a high critical temperature so that heat transfer takes place at the maximum allowable temperature under the saturation line.

A pilot 40 MW plant was developed and built by the General Electric Company in the late 1940s that used mercury vapour for the topping application. The thermodynamically attractive properties of mercury (critical temperature of 1500 °C, saturation pressure at 550 °C of 14 bar) made it possible to combine the mercury topping cycle with the steam bottoming. Plant efficiency of 37% was mentioned in the literature (Haywood, 1980). Potassium vapour can also be considered as a fluid for the topping cycle, but its very low saturation pressure at top temperature (0.1 bar at 560 °C) and other disadvantageous properties inhibit its use for this application.

The concept of steam topping at high pressure (350 bar) and temperature (above 700 °C) with flow splitting was outlined by Rice (1997). In the case of steam topping at moderate conditions, ammonia or an organic medium can be applied as the working fluid in the bottoming Rankine cycle (Horn and Norris, 1966) (Fig. 2.3).

2.3 Rankine/Kalina

The Kalina cycle can provide a better match between the temperature of the flue gases and that of the working fluid, due to the variable chemical composition of the latter (see Section 1.3). The application of this bottoming cycle in direct-fired power plants (Fig. 2.4) can improve the performance of the whole system by 23% compared to the

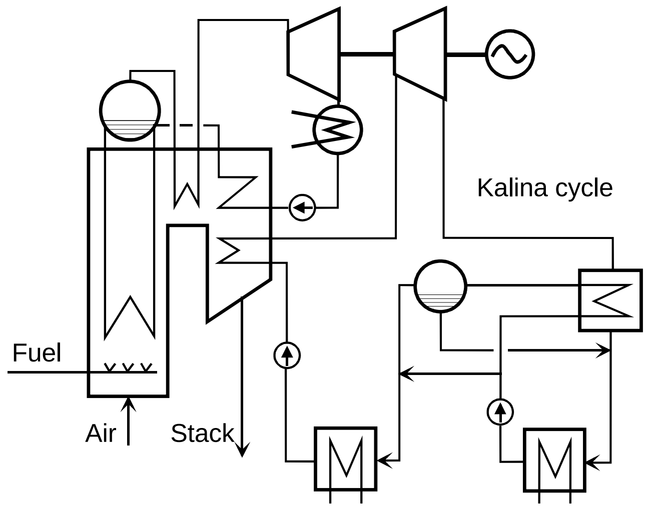


Fig. 2.4. The Rankine/Kalina combined cycle.

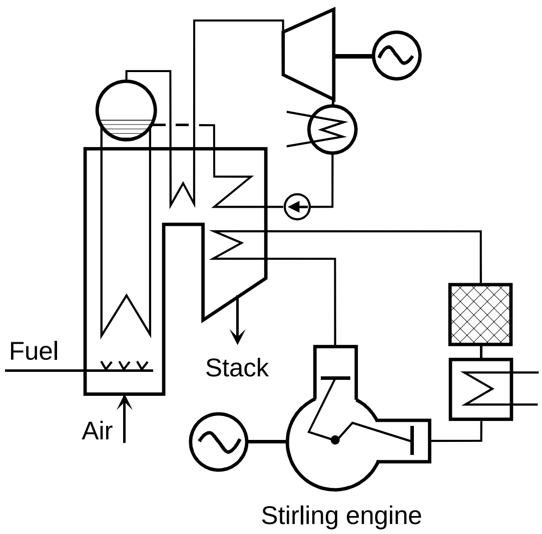


Fig. 2.5. The Rankine/Stirling combined cycle.

Rankine bottoming cycle (Kalina, 1991; Kalina and Tribus, 1992). No part of the Kalina system operates at vacuum conditions, which simplifies operation and maintenance. The steam boiler can be completely replaced by the Kalina-cycle vapour generator. Then, at the same top temperature of 650 °C, the Kalina cycle will obtain approximately 20% greater efficiency than the triple-pressure Rankine cycle, while the former requires no extremely high pressure (Davidson et al., 1996). The use of the Kalina cycle has been recently licensed by ABB for direct coal-fired power plants (Stambler, 1995).

2.4 Rankine/Stirling

The use of the Stirling engine in combination with the Rankine cycle is theoretically attractive, but limited by the small power size of the Stirling engine. An arrange-

Table 2.3. Possible Joule/Rankine combinations.

		Joule topping cycle	
		direct-fired	indirect-fired
Rankine bottoming cycle	**HRSG**		
	unfired	◆	◆
	with supplementary firing	◆	
	Steam boiler (repowering)		
	gas-, oil-, coal-fired	◆	◆
	waste-fired	◆	◆
	Integrated with gas turbine		
	steam and water injection	◆	◆
	blade cooling	◆	◆
	combustor cooling	◆	

ment where the Stirling engine's heater is mounted in a steam boiler is shown in Fig. 2.5. Another position of the heater, for example, before the superheater, can be considered in order to obtain a higher maximum temperature in the Stirling cycle.

2.5 Joule/Rankine

Of the various combined cycles, the Joule/Rankine combined cycle is the most developed and wide-spread. Inexpensive and readily available media, such as air for the topping cycle, and water/steam for the bottoming, and well-developed technologies (gas turbine, waste heat boiler, steam turbine) have led to wide acceptance of this scheme. Possible combinations of these two cycles are presented in Table 2.3. The bottoming cycle can be implemented either as a heat recovery steam generator (HRSG) behind a gas turbine, or in a repowering scheme, where the gas turbine exhaust is fed into the existing steam boiler's furnace. The use of steam as a working or cooling medium within the gas turbine is realized in integrated schemes.

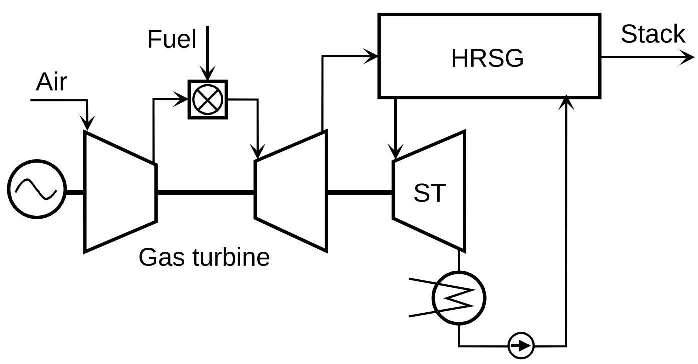
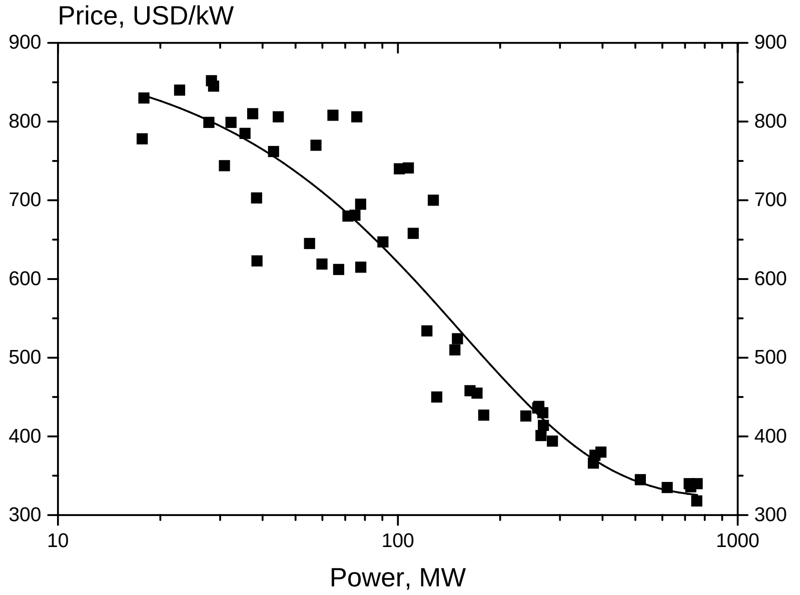


Fig. 2.6. The Joule/Rankine combined cycle.



Power range, MW	Efficiency, %	HRSG type	Price, USD/kW
15–50	40–50	double pressure	650–850
51–100	44–52	double pressure	600–800
101–250	50–52	double- and triple-pressure	400–750
above 250	56–58	triple-pressure with reheat	350–450

Fig. 2.7. Prices and efficiencies versus power output of gas turbine/HRSG combined-cycle plants. Based on the data from the 1997 Gas Turbine World Handbook.

2.5.1 Conventional scheme (GT/HRSG)

In a typical scheme, exhaust heat from the open gas turbine circuit is recovered in a steam generator (Fig. 2.6). In order to provide better heat recovery in the steam boiler, more than one pressure level is used. With a single-pressure HRSG typically about 30% of the total plant output is generated in the steam turbine. A dual-pressure arrangement can increase the power output of the steam cycle by up to 10%, and an additional 3% can result by choosing a triple-pressure cycle (Marston and Hyre, 1995). Modern gas turbine plants with a triple-pressure HRSG with steam reheat can reach efficiencies above 55% (see also Section 3.3.1). ABB claims 58.5% efficiency of a combined cycle plant built around their GT24/26 reheat gas turbines; the same efficiency is cited for the Westinghouse steam-cooled W701G gas turbine (Gas Turbine World Handbook, 1997). General Electric expects even higher efficiencies with the new 'H' technology. A gas turbine with steam cooling of the turbine blades and nozzles combined with an advanced HRSG is expected to operate at a efficiency level of 60% in the near future (Corman, 1995). However, these high efficiency values can be achieved at large units above 300 MW.

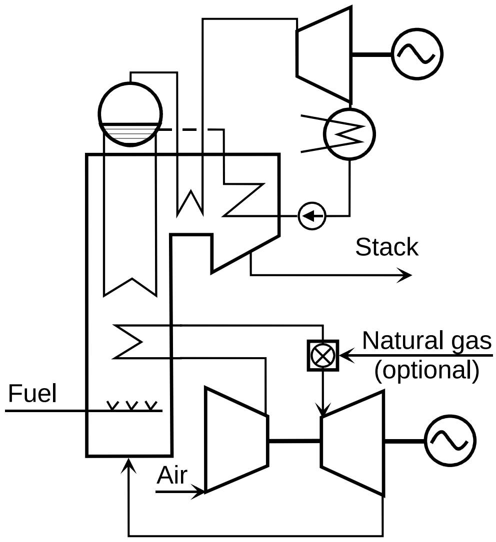


Fig. 2.8. The Joule cycle with external firing combined with the Rankine cycle.

An overview of equipment-only prices and efficiencies based on The Gas Turbine World Handbook (1997) is given in Fig. 2.7. The table under the figure indicates that for an alternative combined-cycle to be attractive in the under 50 MW power range, the cycle should provide an efficiency better than 50% and/or a price lower than 600 USD/kW.

2.5.2 Repowering

In addition to the classical Joule/Rankine scheme (Fig. 2.6), this combination is employed to repower an existing steam plant. (1) The gas turbine exhaust can be utilized as preheated combustion air in the steam boiler or a process furnace. Such a configuration is analyzed for the case of a solid waste incinerator in Chapter 6. (2) Exhaust heat can alternatively be recovered by preheating the boiler feed water for the existing steam plant. Here, the degree of waste heat utilization is lower than in an HRSG, since the heat recovery occurs only within the liquid state. Thus, the maximum temperature that the feed water at subcritical plants can reach is just 374 °C.

2.5.3 Externally-fired gas turbine

A combination of an externally-fired gas turbine with a direct-fired boiler allows combustion of low-grade fuels to drive turbomachinery and to generate steam in the steam boiler. After compression, air is heated indirectly in a heat exchanger, and then expands in the turbine to the atmospheric pressure. The turbine exhaust is blown into the boiler at elevated temperature and serves as a hot windbox (Fig. 2.8). A performance analysis of such a system can be found in the literature (Trumpler, 1985; Huang and Naumowicz, 1992). In the current state of the technology, air cannot be heated to the required gas turbine inlet temperature, 1200–1300 °C, so the maximum practicable temperature is considered 950 °C (Seery et al., 1995). A further increase of the air temperature up to 1300 °C can be accomplished by the use of supplementary firing of

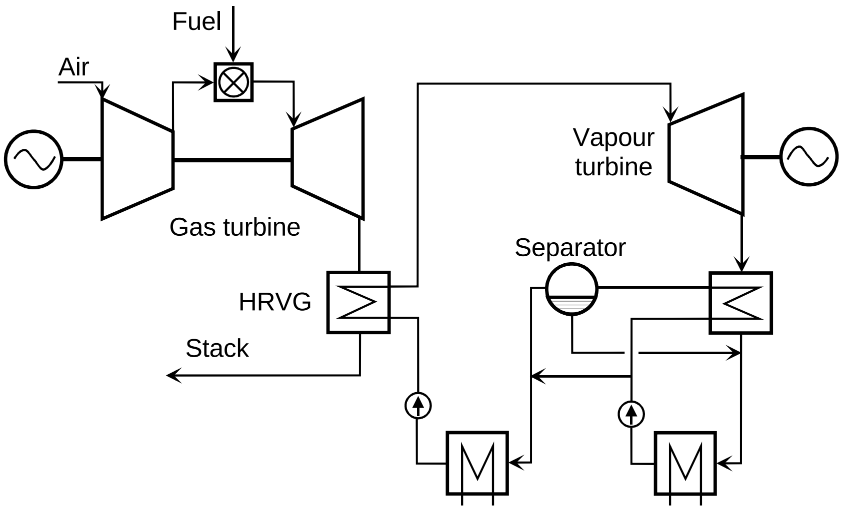


Fig. 2.9. The Joule cycle with the Kalina bottoming.

clean fuels (natural gas, coal-derived gas), or by the use of advanced ceramic materials in the heat exchanger. The externally-fired cycle with and without supplementary firing is analyzed in Chapter 5.

This concept is now being developed under the Combustion 2000 programme of the US Department of Energy by a team of manufacturers, engineering companies and universities. The High Performance Power System (HIPPS) is planned to be commercially available by 2005 with the following specifications: thermal efficiency (based on HHV) of greater than 50%, NO_X emissions less than 26 g/GJ, SO_2 emissions less than 26 g/GJ, and the cost of electricity 10% lower than current coal-fired plants (Ruth, 1995).

2.6 Joule/Kalina

As a bottomer for a gas turbine, the Kalina cycle has been investigated in several studies (El-Sayed and Tribus, 1985; Olsson et al., 1991; Kalina et al., 1992; Rumminger et al., 1994; Marston and Hyre, 1995; Bjorge, 1995). In these studies was found that the Kalina cycle can produce 10–30% more power than a Rankine cycle. For example, in combination with a General Electric MS9001FA gas turbine, the Kalina combined cycle rates 412 MW and 59.6% efficiency versus 396 MW and 57.3% in the conventional triple-pressure combined cycle (Bjorge et al., 1997).

Because the exhaust pressure of the vapour turbine in the Kalina cycle is above atmospheric pressure, no vacuum is needed to be maintained in the condenser during operation, or stand-by periods. Therefore, the start-up procedure can be performed in a much shorter time. The working fluid composition can easily be changed in order to obtain the optimal performance in respect to alterations in load or ambient conditions. Another advantage is the smaller size of the whole unit. The footprint of the Kalina plant is about 60% of the size of a Rankine plant design (Stambler, 1995).

The operation of the 3 MW pilot plant built in Canoga Park, California, was demonstrated in 1992-1994 (Anonymous, 1992; Corman et al., 1995). More than

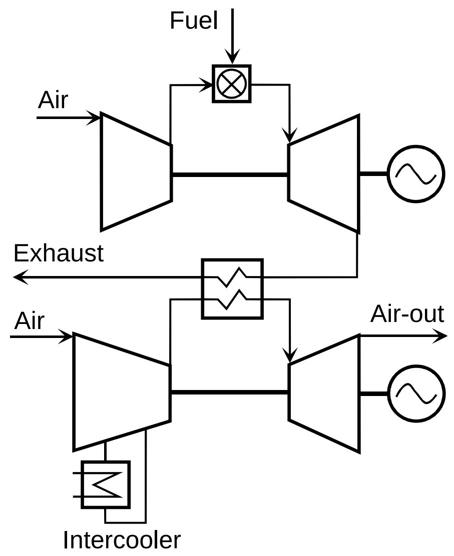


Fig. 2.10. A gas turbine with air bottoming (the Joule/Joule cycle).

4000 hours of operation confirmed the high degree of the heat recovery and system operability under different work conditions. No traces of corrosion were noticed in samples from different parts of the unit.

In 1993 General Electric signed an exclusive world-wide licensing agreement with the Kalina cycle patent owner (Exergy, Inc., Hayward, California) to design and market gas turbine powered Kalina combined cycle plants. A General Electric demonstration plant is planned to be put into operation by 1998. The plant is expected to be in the 100 MW size range (Bjorge et al., 1997).

2.7 Joule/Joule

Two Joule cycles can be combined by an air-gas heat exchanger (Fig. 2.10). The exhaust of the primary gas turbine is sent to a heat exchanger, which, in turn, heats the air in the secondary gas turbine cycle. Air is expanded in the turbine to generate additional power. Intercooling in the air compressor reduces the required expander work. In comparison to the Joule/Rankine combined cycle, this scheme does not require bulky steam equipment (boiler, steam turbine, condenser), or a water processing unit, and allows unmanned operation.

Recent studies (Bolland et al., 1995; Hirs et al., 1995) showed the feasibility of this configuration. These reported an increase of power by 18 to 30% depending on the number of intercoolers, and an efficiency growth of up to 10% points. For example, for the Allison 571K topping gas turbine, introduction of the air bottoming cycle with two intercoolers led to an increase in power from 5.9 to 7.5 MW and in efficiency from 33.8 to 43.2%. Comparable results were obtained with the General Electric LM2500 topping turbine.

The scheme can also be applied for cogeneration. The exhaust air, leaving the cycle at 200–250 °C, can be used for process needs that require heat of such temperatures.

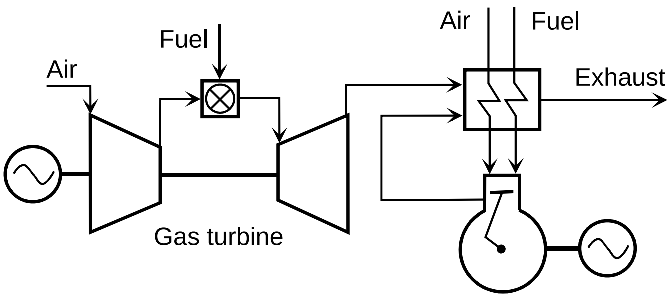
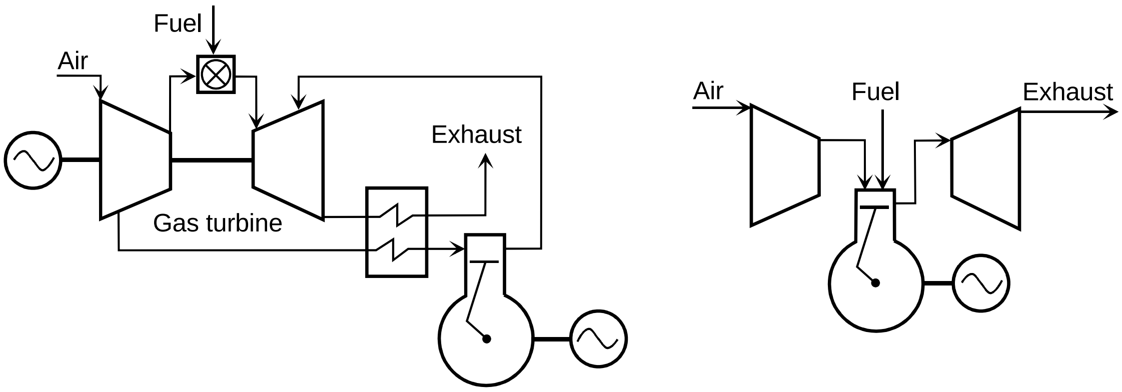


Fig. 2.11. The Otto (Diesel) engine inlet air preheating by a gas turbine.



a) b)

Fig. 2.12. Joule/Otto (Diesel) combined cycles:
(a) pressurization by a gas turbine;
(b) turbocharging.

Uncontaminated hot air is of special interest for food processing or pharmaceutical industries.

Because the bottoming cycle has a low temperature ratio, expansion work insignificantly exceeds the work consumed by the compressor. Therefore, the cycle is sensitive to turbomachinery efficiency, and necessitates the use of the most efficient equipment. This sensitivity can be diminished by burning additional fuel and raising the inlet temperature in the bottoming cycle to 1000–1200 °C. When the outlet air is used in food processing industries, the firing should proceed indirectly.

The performance analysis of this cycle is given in Chapter 7.

2.8 Joule/Otto(Diesel)

Preheating of the inlet air of an Otto (Diesel) engine can sufficiently improve its performance. The gas turbine exhaust can be applied in order to increase the inlet flow temperature. The exhaust heat can be recovered either in a heat exchanger (Fig. 2.11), or directly, when the exhaust gases, containing 14–16% oxygen, are passed into the Otto engine's combustion chamber. Another possibility offers a pressurization scheme, where air is extracted from the compressor and fed into the Otto (Diesel)

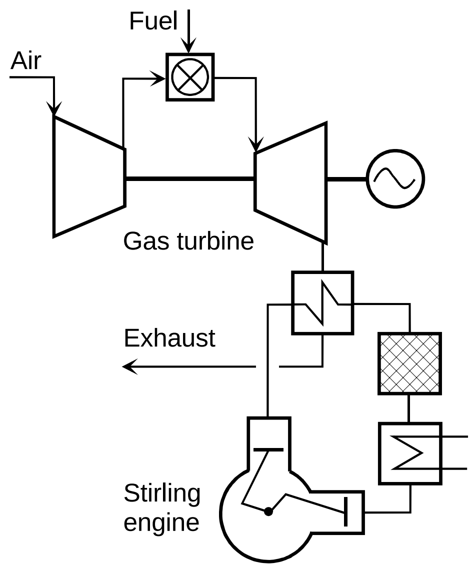


Fig. 2.13. Gas turbine waste heat recovery by the Stirling engine.

engine. Subsequently, the engine outlet flow expands through the low-pressure stage of the gas turbine (Fig. 2.12a). Simple turbocharging and utilization of engine exhaust is covered in Section 2.13 on the Otto (Diesel)/Joule combined cycle.

2.9 Joule/Stirling

In a combination of a gas turbine and a Stirling engine, the heater of the latter can be placed either in the combustor of the turbine, or after the expander in the exhaust flow (Fig. 2.13). The arrangement is determined by the optimal performance of the combined cycle and by the materials used in the Stirling heater's head. As much as 9 MW can be recovered by a bottoming Stirling cycle from the exhaust of a Rolls-Royce RB211 gas turbine of 27 MW (Walker et al., 1994). Such a combined-cycle plant will obtain an efficiency of 47.7%. Just as the Joule/Joule cycle, this combination provides a compact and simple heat recovery scheme.

2.10 Otto(Diesel)/Rankine

The exhaust gas from the Otto/Diesel engine has a temperature of 300–500 °C and, thus, this heat can be recovered in a waste heat boiler (Fig. 2.14a). However, the engine releases a smaller amount of exhaust gases than a comparable gas turbine, because it does not require a considerable air flow for cooling: the air excess is typically 30–40%, compared to 200–350% for a gas turbine. Hence a 10 MW engine can provide about 1–1.5 MW of electric power that can be generated in the bottoming steam or organic Rankine cycle (Vernau, 1984). Moreover, performance of the Rankine cycle rapidly declines when the exhaust temperature is decreased (Woodward, 1994).

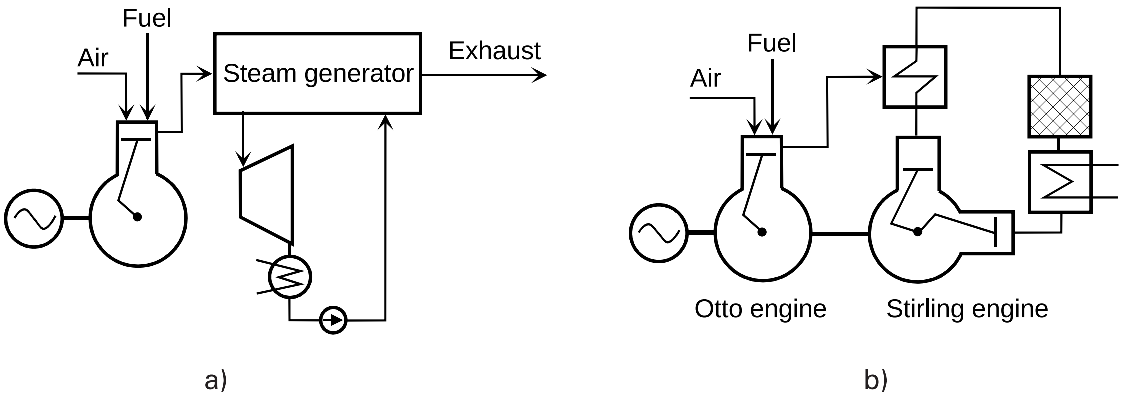


Fig. 2.14. Otto (Diesel) engine waste heat recovery:
(a) by the Rankine cycle;
(b) by the Stirling cycle.

2.11 Otto(Diesel)/Kalina

A combination of the Otto and Kalina cycles can be considered comparable with the Otto/Rankine combination. The use of the ammonia-steam mixture may improve heat recovery, but because the exhaust flow of the Otto engine is not significant, the improvement may not be justified by the complexity of the Kalina bottoming unit.

2.12 Otto(Diesel)/Stirling

Yet another Otto engine heat recovery configuration is presented in Fig. 2.13b. The gas engine exhaust provides heat to the Stirling engine's heater, where it is recovered to drive the piston. Taking into consideration the small size of the Stirling engine, this concept seems most suitable when applied in a vehicle or other small size applications.

2.13 Otto(Diesel)/Joule

The implementation of an expander in the Diesel cycle allows to expand the exhaust to the atmospheric pressure and thus provide additional power to the cycle (Fig. 2.12b). Comparing the use of the Joule-Brayton cycle versus the Rankine cycle in the bottoming configuration, Woodward (1994) noted that high cylinder exhaust pressure and low temperature favour the Joule alternative. On the other hand, high cylinder gas temperature and low pressure make the Rankine cycle more attractive for heat recovery.

An enhancement of a turbo-charged Diesel engine by means of inlet air humidification has led to an increase in efficiency (from 33.3 to 36.5%), and power output (from 185 to 205 kW), as well as in the reduction of NO_X emissions (from 12 to 3 g/kWh), as reported by Rosen and Olsson (1996).

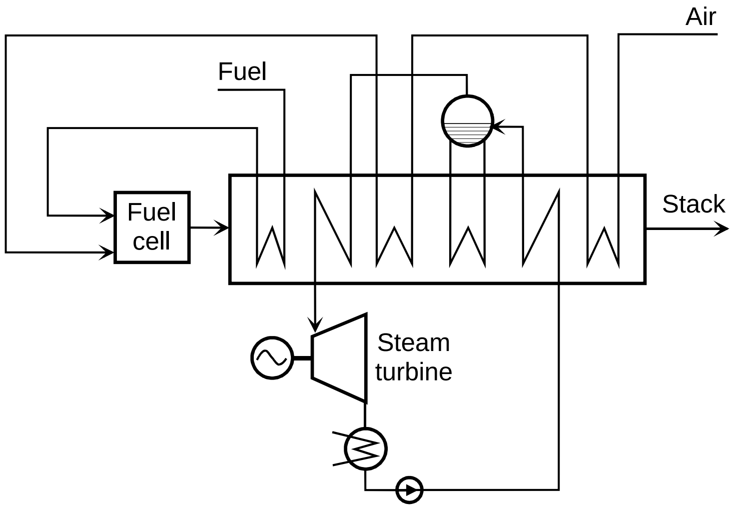


Fig. 2.15. SOFC with steam bottoming (Fuel cell/Rankine).

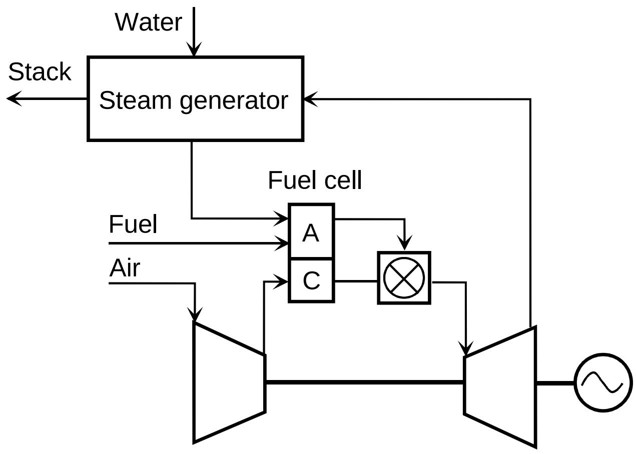


Fig. 2.16. Fuel cell in the Joule cycle.

2.14 Fuel cell/Rankine

An evaluation study of a fuel cell system with steam bottoming (EPRI, 1992) indicated that the plant had the potential to maintain high efficiencies over a broad range of load conditions. The schematic diagram of the plant is given in Fig. 2.15.

The study, based on Westinghouse atmospheric tubular solid oxide fuel cell technology, showed that a large-scale plant (300 MW) would have 49% efficiency in full load and around 52% in 25%-load operation. For a smaller-scale unit of 20 MW, the figures were 46% and 47%, respectively. The capital requirement for the combined cycle plant was estimated to range from 800 to 1200 USD/kW for the large unit, and 1500 to 2100 USD/kW for the small one.

In another study (Mozaffarian, 1994) an efficiency of 55% was reported, assuming fuel utilization of 80%, average stack temperature of 950 °C, and the local cell impedance of $0.7 \mathrm{W \cdot cm^2}$.

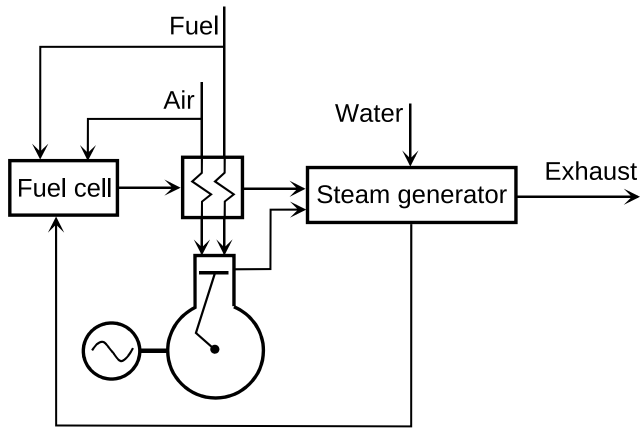


Fig. 2.17. Fuel cell combined with an Otto engine and a waste heat boiler.

2.15 Fuel cell/Kalina

As already mentioned in Section 5.6, the Kalina bottoming cycle offers 10–30% more power and 3–9% higher efficiency than the Rankine bottoming cycle. Therefore, the findings of the reports cited in that section can be extrapolated for the fuel cell/Kalina combination with a correction for higher efficiency.

2.16 Fuel cell/Joule

A fuel cell system, operating at elevated pressure allows to integrate a gas turbine within the system, thus improving performance. The schematic of the system is presented in Fig. 2.16. Several studies showed an advantageous performance of such a concept. In the Argonne National Laboratory's report (Minkov et al., 1988) the optimal fuel utilization in a MCFC-gas turbine combined plant was discussed. The lowest cost of electricity was obtained at a value of fuel utilization of 55%. A study by Westinghouse (Parker and Bevc, 1996) reported on a 3 MW SOFC combined cycle plant based on a Heron turbine. By the year 1999 the natural-gas fuelled plant is expected to have efficiency above 60%, NO_X emissions lower than 5 ppm, a turndown ratio of 4:1, and installed costs of 1000 USD/kW. The use of the fuel cells integrated with combustors allows efficiency to approach 70%.

Other advanced concepts with anode/cathode gas recirculation, steam injection, intercooling and recuperation (Harvey and Richter, 1994; Jansen and Mozaffarian, 1995; Jansen et al., 1996) were reported to obtain efficiencies around 70%. The specific costs of a fuel cell/gas turbine combined cycle from 1600 NLG/kW to 2500 NLG/kW were cited.

This combination is claimed to have the highest efficiency of any options discussed in this chapter, and can, therefore, be seen as a choice for the future power plants.

In addition to the series scheme, a parallel concept can be considered. The gas mixture, resulting from a reforming reaction of methane, is separated by the use of a membrane into hydrogen and carbon dioxide. Then, hydrogen is fed to a low temperature fuel

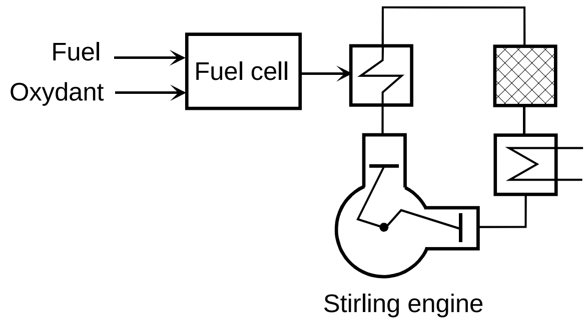


Fig. 2.18. The Fuel cell/Stirling combined cycle.

cell, while CO is burned in the combustor of the gas turbine. The most effective polymer membranes cannot be applied here directly, since they operate at temperatures below 150 °C. The use of other materials, such as ceramic, glass, or metal, will allow separation of the reformed gas at higher temperatures. However, such membranes are costly and less effective, and the implementation of this scheme may be hindered by economic reasons.

2.17 Fuel cell/Otto(Diesel)

A number of configurations can be considered, regarding the combination of a topping fuel cell and a bottoming Otto (Diesel) engine. High-temperature waste heat from the fuel cell could preheat the inlet air of the Otto (Diesel) engine, and the engine's exhaust would be recovered to generate steam for the reforming process (Fig. 2.17). Alternatively, the fuel, partially utilized in the fuel cell, could be passed through a steam boiler directly into the Otto engine.

2.18 Fuel cell/Stirling

This combination can provide high performance on a small scale, determined by the size of the Stirling engine, for example, in automotive vehicles (Fig. 2.18). Since both represent a relatively new technology, this concept is not expected to be realized in the immediate future.

2.19 Fuel cell/Fuel cell

Partially utilized fuel and oxidant from a topping fuel cell can be directed to a bottoming cell to complete oxidation. For example, the SOFC anode flow containing unused fuel and H_2O/CO_2 mixture are passed to MCFC, and the cathode flow of unused air and H_2O is sent to the low-temperature PAFC. Such a scheme allows operation at even higher efficiency than a combination with a heat engine, although its realization does not seem to be feasible within the next 10 years.

2.20 Conclusions

Combined cycles offer better performance than single cycles. Moreover, shortcomings of the topping cycle can often be compensated by the bottoming cycle. Combining high-temperature cycles with those of medium- and low temperature provides the most effective way in approaching Carnot efficiency, and thus better utilization of the fuel exergy. However, the possibilities for combination may be limited by various factors, such as the status of development, power output, fuel requirements, or part-load characteristics.

Considering these factors the following combined cycles were found of interest for future development.

The Joule/Rankine cycle, in the form of an externally-fired combined cycle, can provide high efficiency and low emission release, while burning low-grade fuels. This cycle is considered in Chapter 5. In addition, the gas turbine can be used to repower an existing steam plant, or to enhance a solid waste incinerator. The latter option with possibilities for superheating in an HRSG behind the gas turbine is discussed in Chapter 6.

The Joule/Joule cycle, a gas turbine with air bottoming, offers a performance comparable to that of the steam bottoming cycle without the use of steam-water equipment. The air, leaving the bottoming cycle at a temperature above 200 °C, can be utilized in food processing industries. Analysis of this cycle is given in Chapter 7.

The Joule cycle with a topping high-temperature fuel cell represents a highly effective combined cycle that, furthermore, has low emissions and a broad turn-down ratio. The development of this cycle falls under the scope of another project within the NECT programme, and therefore, will be not covered in the following work.

Several other promising cycles have not been selected for further development due to a small power size (the Stirling cycle), or because of the advanced state of development elsewhere (Kalina cycle).

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Chapter 3
Advanced Cycles

Abstract

This chapter gives an overview of improvements that can be implemented within the basic and combined cycles described in the previous chapters. Since the Rankine and Joule cycles represent the most developed technologies, the modifications covered here are primarily related to these two cycles. These include multi-pressure steam boilers, gas turbines with steam injection, air humidification, chemical recuperation, and partial oxidation. Main performance parameters are given along with a discussion on advantages and disadvantages of these schemes. Based on conclusions of this chapter, four cycles were selected for the next phase of the NECT programme for technical and economic evaluation.

3.1 Introduction

The last generation combined-cycle plants based on gas and steam turbines can operate at efficiencies up to 58.5% (Gas Turbine World Handbook, 1997). All major gas turbine manufacturers, including General Electric, ABB, and Westinghouse-Siemens, offer a combined-cycle package of similar efficiency at a power range above 250 MW. In spite of such a high efficiency value (with a TIT of 1350 °C, the Carnot factor is 0.78), innovative cycles are being proposed in order to rise this level of efficiency even higher, and to obtain comparable efficiency at a smaller scale, where performance values are not so spectacular. At the scale under 30 MW combined-cycle efficiency is typically about 42%, which is caused by technical and economic constraints. Because the New Energy Conversion Programme is aimed at distributed power generation, the cycle innovations are interesting for an implementation within just this power range.

Research and development work, which is being carried out by gas turbine manufacturers, engineering companies, research organizations and universities, offers various way of improving cycle performance.

Table 3.1. Advanced cycles reviewed in other studies.

	GT with intercooling	GT with reheat	GT with thermal recuperation	STIG	HAT	Evaporative cooling
Stecco, 1992				◆	◆	
Stambler, 1992	◆	◆		◆	◆	
Hodrien et al., 1994	◆	◆	◆	◆	◆	◆
Bannister et al., 1995	◆		◆			◆
Chiesa et al., 1995	◆			◆	◆	
Stambler, 1995	◆		◆			
Yang, 1995	◆		◆			
Davidson et al., 1997	◆	◆	◆	◆	◆	

	Kalina cycle	Chemical recuperation	Partial oxidation	Fuel cell-gas turbine	Ceramic gas turbine
Stecco, 1992	◆				
Stambler, 1992		◆			
Hodrien et al., 1994	◆	◆	◆	◆	◆
Bannister et al., 1995		◆			
Lior, 1995			◆	◆	
Stambler, 1995		◆			◆
Yang, 1995		◆	◆		
Davidson et al., 1997	◆	◆			

3.2 Background

Several studies on the advanced energy conversion technologies have been performed in the recent years. The California Energy Commission evaluated some promising new gas turbine-based technologies (Stambler, 1992). Hodrien and Fairbairn (1994) gave a general overview of power generation systems from steam power plants to fuel cell combined cycle systems. Stecco (1992), Chiesa et al., (1995) and Yang (1995) described advances in gas turbine technology, and Lior (1995) presented some novel energy conversion approaches. Another two studies resulted from the Collaborative Advanced Gas Turbine programme (Davidson et al., 1997) and the Advanced Turbine System programme (Stambler, 1995), which are jointly sponsored by DOE, utilities, and manufacturers. The programmes aim to accelerate the commercial availability of higher efficiency, natural-gas-fuelled advanced gas turbines and cycles for use in utility and distributed applications in the next decade (Cohn et al., 1994). The technologies covered in these studies are summarized in Table 3.1.

The aforementioned reports allow some conclusions to be drawn. Gas turbine-based power plants will remain the most efficient energy conversion technology in the coming decade. Advances in fuel cell development will allow the fuel cells to compete with gas turbine systems, or to enhance the latter.

Regarding the gas turbine technology, considerable investments in material research are aimed on increasing the firing temperature, which is the major constraint on the turbine efficiency. However, a comparable increase in efficiency can be

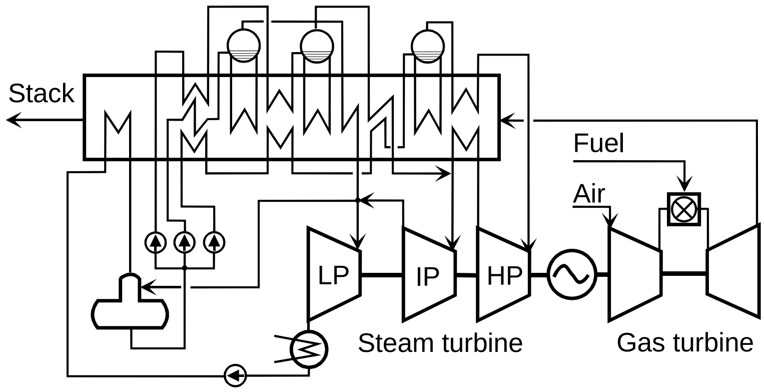

Fig. 3.1. Triple-pressure heat recovery steam generator with reheat.

achieved at lower cost by the use of intercooling, reheat, partial oxidation, and/or recuperation.

As can be seen from Table 3.1, the cycles with intercooling, chemical and thermal recuperation have attracted most of the interest. The most advanced cycles, the HAT cycle, the chemically-recuperated and the partial oxidation gas turbines, were reported to have a potential to reach efficiencies around 60% in a five-year period and are expected to enter commercial service shortly after 2000 (Hodrien and Fairbairn, 1994).

3.3 Advanced Rankine cycles

3.3.1 Multi-pressure steam boiler

The drawback of the Rankine cycle in a bottoming application due to heat addition at a constant temperature has already been noted in Chapter 2. In modern power plants it is solved by the introduction of the second or third pressure level in order to reduce the mean temperature difference in the boiler. An example of such an arrangement is given in Fig. 3.1.

The highest efficiency of a commercially-available advanced combined-cycle plant (ABB GT-26 gas turbine with a triple-pressure reheat HRSG) of 58.5% was reported (Gas Turbine World Handbook, 1997). To obtain a higher value, a steam bottoming cycle with five and even more pressure levels with steam turbo-chargers was proposed by Jericha et al. (1997). Such an arrangement results in heat utilization with a minimum temperature difference and still avoiding the use of mixed working fluid, as in the Kalina cycle. Operating within the subcritical region with pressures up to 180 bar and a superheat temperature of 542 °C, a combined-cycle plant was reported to obtain overall efficiency of about 60%.

3.3.2 Water flashing

The multi-pressure system's complexity inhibits its use at small scale. In order to simplify the heat recovery steam boiler the concept of water flashing was proposed by

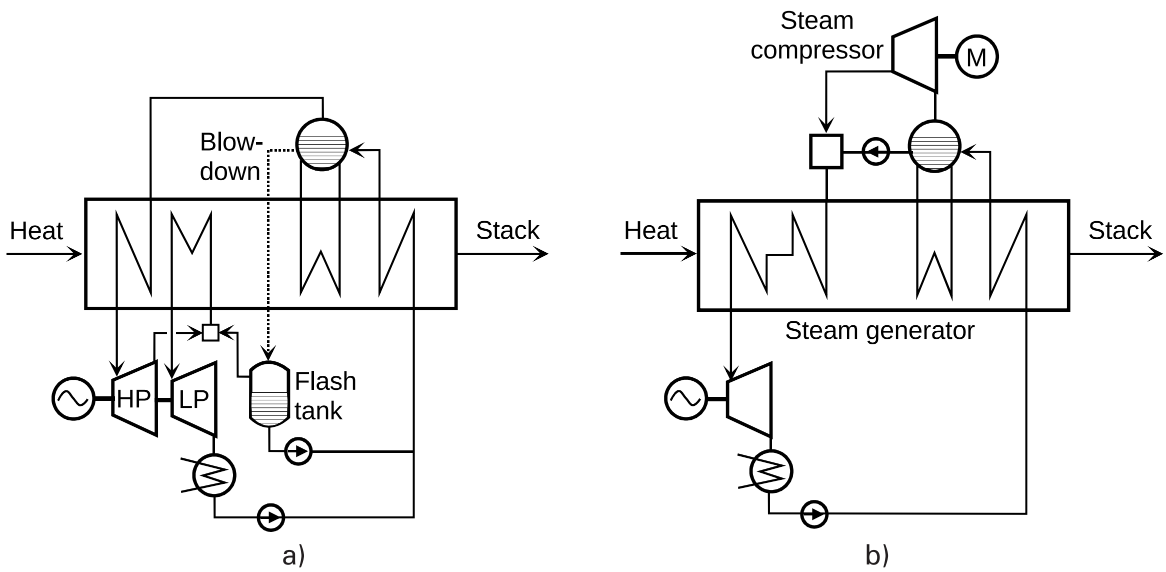

Fig. 3.2. (a) Water flashing; (b) Steam recompression

Dechamps (1994). This scheme creates the second, lower pressure level by means of extracting the saturated, pressurized water from the evaporator's drum and flashing it in the flash tank (Fig. 3.2a). This arrangement adds up to 2 percentage points to the single-pressure combined cycle efficiency, and can be comparable with the dual-pressure HRSG schemes. Dechamps reported 51.5% efficiency for a typical combined cycle plant based on the boiler with water flashing.

3.3.3 Steam recompression

Another concept to simplify the heat recovery steam generator was proposed by Cheng (1978). Instead of a separate high pressure section in the steam generator, the pressure in the boiler is raised by means of the steam compressor (Fig. 3.2b). This is particularly useful for small power plants where a low steam/water flow rate makes impractical splitting of the flow into different pressure levels.

Since the same flow is fed through all tube bundles of the boiler, heat recovery is improved. A study by Grimaldi and Manfrida (1992) showed good thermodynamic performance of the plant. At a high-pressure level of 35 bar and a low pressure of 10 bar, a combined-cycle gas turbine plant will operate at 57% efficiency. In the following study Grimaldi and Desideri (1994) reported that steam recompression bottoming cycle is comparable to the conventional dual-pressure steam bottoming one, while requiring a 20% smaller surface area of the boiler at the expense of the steam compressor.

3.3.4 Steam flow splitting

In a cogeneration plant special attention is paid to the plant's ability to operate under various heat loads. Normally, it is accomplished by several means, such as expanding of the excess steam through the condensing part of a steam turbine, dumping the excess steam in a dump condenser, or by using steam injection in a gas turbine.

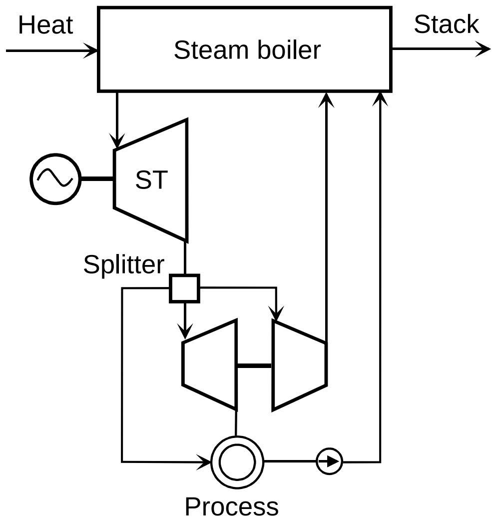

Fig. 3.3. Steam flow splitting in a cogeneration plant.

However, each of those has its drawbacks. The use of a condensing turbine requires an expensive low-pressure section and a condenser. Moreover, at full heat load the condensing section presents a parasitic loss. The dump condenser appears even a worse solution. Steam injection in a gas turbine is possible to a certain extent, unless the turbine is specially designed to work in this mode.

Another way to provide operating flexibility is the concept of steam flow splitting and recompression (Lievense and Hirs, 1986). The excess steam is recompressed to initial pressure by a steam compressor. The compressor is driven by a back-pressure steam turbine. A simplified scheme of the concept is presented in Fig. 3.3. Maintaining a constant flow through the main steam turbine allows operation at full-load efficiency at different heat load conditions.

At combined-cycle cogeneration plants, heat load can be controlled by supplementary firing, by steam injection (see also the following section), and/or by putting the gas turbine in part-load operation. A combination of these methods with steam splitting will permit an effective and flexible operation of the cogeneration plant.

3.4 Advanced Joule cycles

3.4.1 Water and steam injection

The positive effect of the steam or water injection on the performance of a gas turbine is well-known. Water injection has been used for power augmentation in aircraft engines since the 1950s, and in industrial gas turbines since the 1960s (Foster-Pegg, 1989). The injection increases the mass flow and the specific heat of the working fluid, which gives additional power to the cycle. Along with this, it helps to lower NO_X formation in the combustion chamber and to cool the blades more effectively than air.

Steam injection is more effective than water injection, since the steam generated in the HRSG is fed into the turbine, thus improving the heat recovery. Such a cycle is

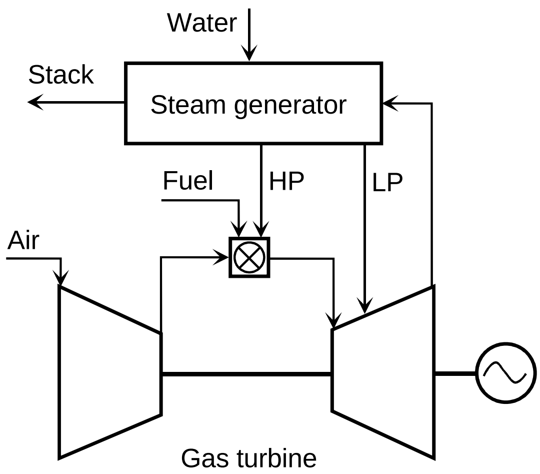

Fig. 3.4. Steam injection for power augmentation and NO_x reduction.

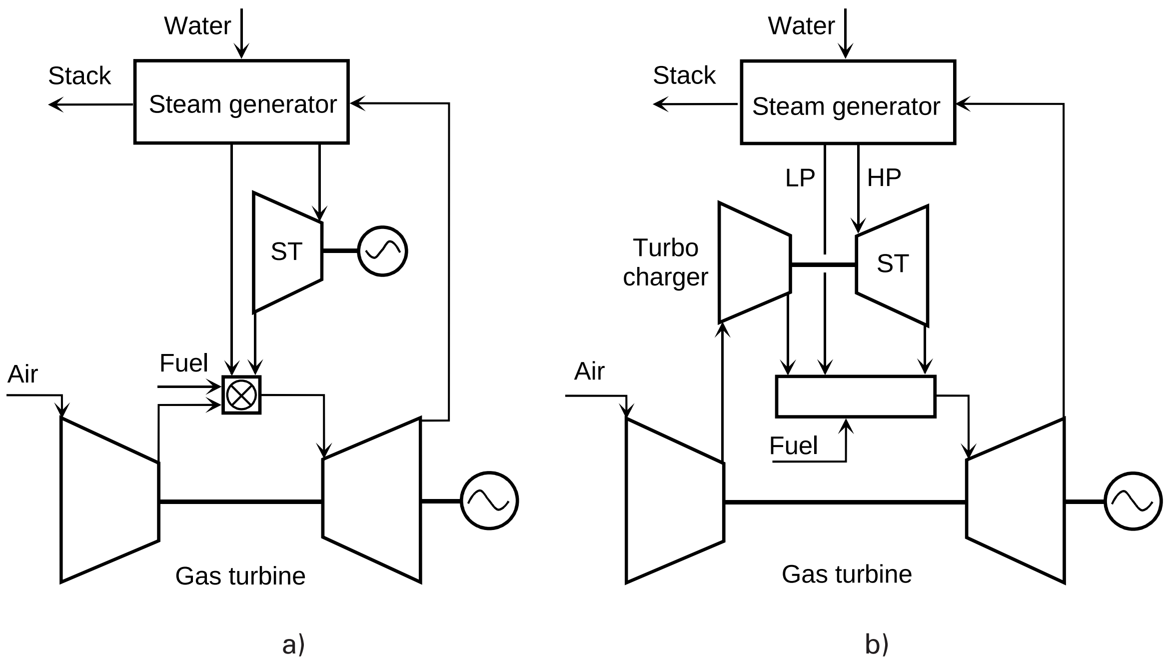

Fig. 3.5. Steam-injected gas turbine configurations.

often designated as the STIG (STeam Injection Gas turbine) cycle. High-pressure steam can be injected into the combustor, while intermediate-pressure and low-pressure steam are often expanded in the first gas turbine stages, as shown in Fig. 3.4 (Cheng, 1978; Larson and Williams, 1987). There are several gas turbines specially-designed for steam injection, such as the General Electric LM2500 and LM5000 STIG series, Allison 501-K, or Ruston TB5000.

In the LM5000 STIG turbine about 7% steam by weight of the air flow is injected at high pressure, and up to 6.5% of the air flow at low-pressure. By this means the power output is increased from 34 MW to 52 MW and efficiency from 37 to 43% (Tuzson, 1992; Rice, 1995). When compared with the combined cycle, the STIG plant proves to be economically competitive in power range under 150 MW (Van Laar et al., 1988).

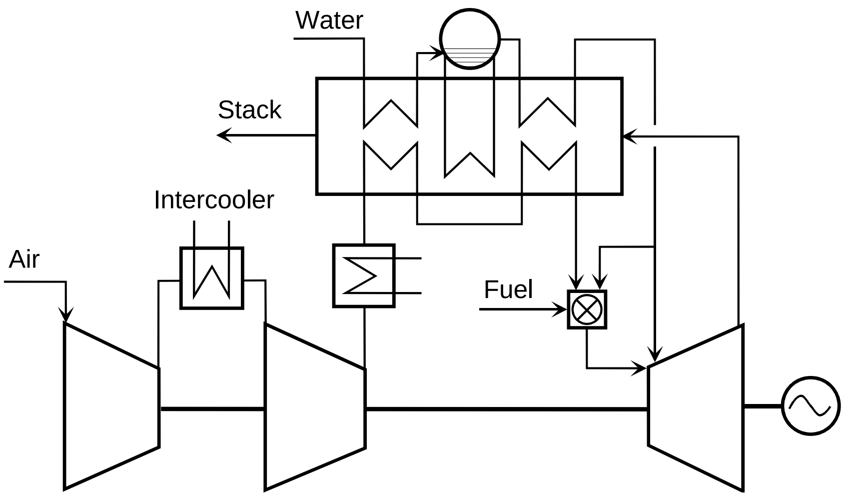

Fig. 3.6. Dual-recuperation intercooled aftercooled steam-injected cycle (DRIASI).

It needs, however, to be recognized that expansion of steam in the gas turbine proceeds to the atmospheric pressure and in a less efficient manner than in the steam turbine. Whereas in the combined-cycle plant steam leaves the steam turbine at much lower pressures, thus providing more power. Therefore, a gas turbine with steam injection will always have a lower efficiency than that in combined-cycle operation.

The introduction of intercooling and reheat in a STIG turbine allows to reduce the power consumed by the compressor from 50% of the total output for modern engines down to 30%. Therefore, efficiency of the gas turbine becomes less dependent on the compressor characteristics and the work ratio is considerably increased. The power of a conceptual intercooled LM5000 STIG was reported to grow to 110 MW, and efficiency to 55% (Cohn, 1984).

Another study on the intercooled STIG (Chiesa et al., 1995) reported 52.2% efficiency at a TIT of 1370 °C and a pressure ratio of 34, and 53.2% efficiency at a TIT of 1500 °C and a pressure ratio of 45. An additional increase in efficiency by about 3 percentage points was expected with the introduction of a reheat stage (Macchi et al., 1991).

Other STIG modifications include: a STIG turbine with a topping steam turbine (Fig. 3.5a), where HP steam is first expanded in a back-pressure steam turbine and then is injected into the combustor (Rice, 1995); the Turbo-STIG, the turbocharged steam injected gas turbine (Fig. 3.5b), proposed by Foster-Pegg (1989). His calculations, based on the data of the existing turbines from 3 to 47 MW, showed that the Turbo-STIG configuration would result in an average increase in power of 95% and an average efficiency rise from 30% to 42.6%.

A power plant that combines steam injection, recuperation, and water injection is described by Bolland and Stadaas (1995), and presented in Fig. 3.6. The analysis of this concept for different classes of gas turbines (industrial and aero-derivatives) showed that the dual-recuperated intercooled aftercooled steam-injected (DRIASI) cycle can provide comparable or superior efficiencies to those of combined cycles for small systems up to 30 MW. For larger systems, the performance of the DRIASI cycle was found to be inferior both to combined cycles and to steam-injected turbines.

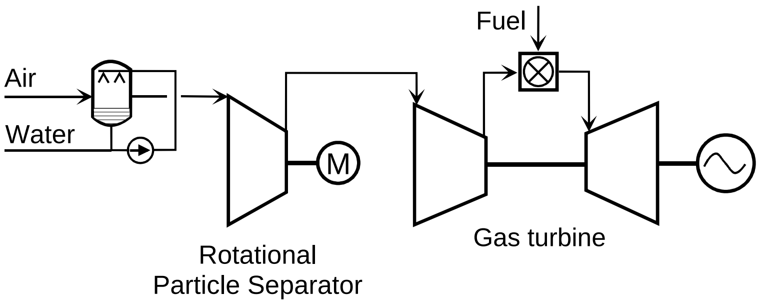

Fig. 3.7. Supercharging a gas turbine with the Rotational Inlet Air Filter and an evaporative cooler.

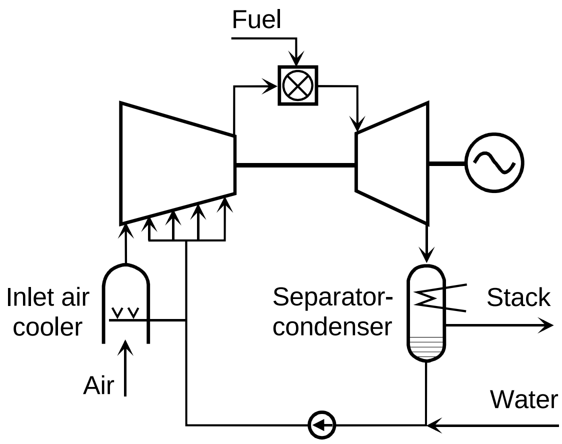

Fig. 3.8. Gas turbine with water injection in the compressor.

3.4.2 Supercharging and evaporative cooling

The pressure losses in the inlet air filter and in the heat recovery steam generator lead to a certain decrease in efficiency and power. For a single-pressure combined cycle every 10 mbar inlet loss will result in a 0.8–1.0% decrease in efficiency and 1.6% reduction in power output. Every 10 mbar outlet loss will decrease efficiency by 0.7–0.9% and power output by 0.6% (Gas Turbine World Handbook, 1997). In addition, a 10 mbar loss will cause a 1–2 °C temperature rise in the stack.

By means of supercharging, the inlet pressure in the system can be increased and that results in additional power generated in the gas turbine. The use of the rotating particle separator can serve two purposes, as an inlet air filter, and as a supercharger (Fig. 3.7). The rotating separator permits an effective separation of solid and liquid particles of diameter 0.1 μm and larger from gases (Brouwers, 1996). An enhancement, introducing an evaporative cooler, allows even more power to be produced. The advantages of air cooling for power augmentation are described by several authors (De Lucia et al., 1993; Kolp et al., 1995).

However, additional fuel consumption will be required in order to bring the working fluid to the turbine inlet temperature, and a rather small increase in efficiency is expected. In Laagland et al. (1996) a 3.27% increase in power and a 0.68% increase

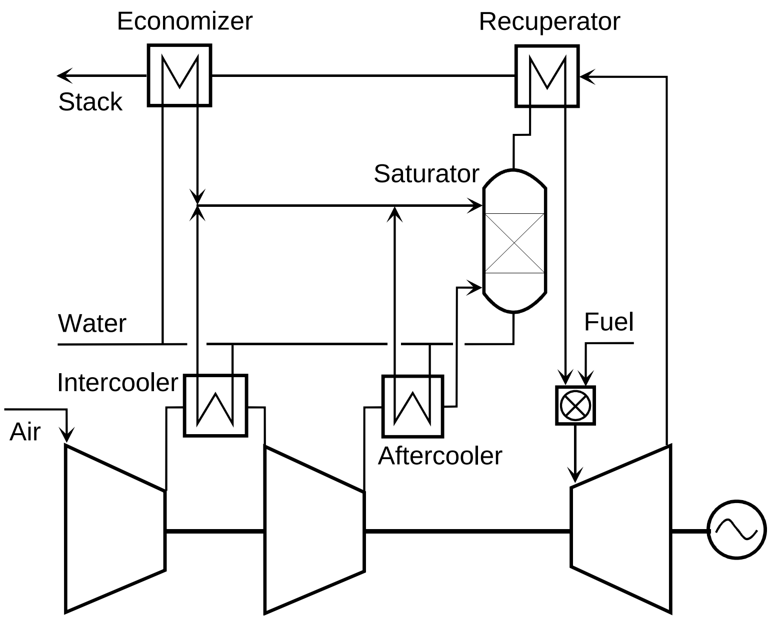

Fig. 3.9. The Humid Air Turbine cycle.

in efficiency of a gas turbine are reported. In the case of the combined cycle the improvements were 2.67% and 0.11%, respectively.

3.4.3 Wet compression

The benefits of intercooling have already been discussed in the previous chapter. However, the use of heat exchangers for this purpose will result in a re-design of the existing engine, and an increase in size and complexity. A much simpler approach, known as wet compression, can be applied instead. The intercooling is accomplished by injecting water at the compressor stages, which results in nearly isothermal compression (Fig. 3.8). Water in the exhaust is recovered in a separating-condensing unit.

Assuming a turbine inlet temperature of 1100 °C, a pressure ratio of 30, and an isentropic efficiency of compressor and expander of 0.85, the efficiency of a plant can reach 43% (Poletavkin, 1980). Introduction of a reheat stage and an increase in the pressure ratio to 100 improve efficiency to 48% in a simple cycle. The wet compression was found feasible in experiments carried out on a Kawasaki S1A-02 gas turbine, as reported by Qun et al. (1997).

This concept is employed in more complex configurations, such as the one discussed below in section 3.4.7 on gas turbine cycles with multi-stage combustion.

3.4.4 Humid Air Turbine cycle

Originally proposed as the evaporative-regenerative cycle (Mori et al., 1983; Anonymous, 1987), the humid air turbine (HAT) cycle provides a substantial power boost of the system and an efficiency rise of several percentage points. A more advanced concept with intercooling (Fig. 3.9) can provide even higher efficiency. The

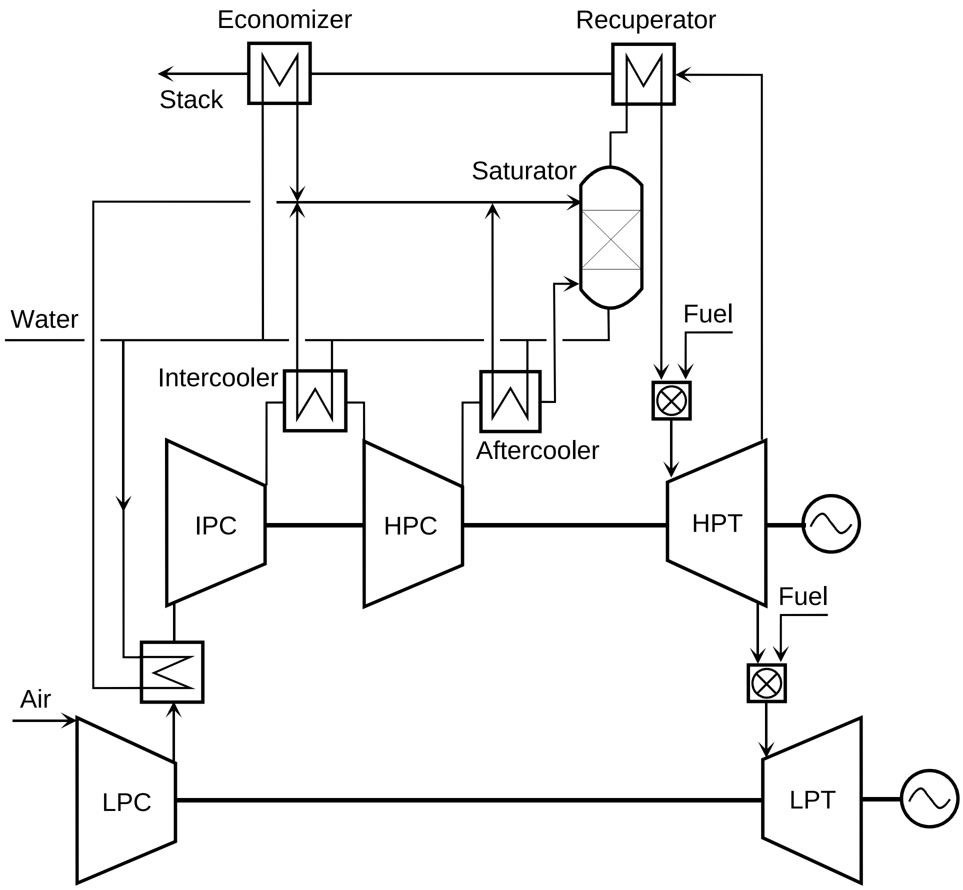

Fig. 3.10. The Cascaded Humidified Advanced Turbine (CHAT).

air is first compressed in the low-pressure compressor of the combustion turbine and then enters the intercooler. The heat of compression is recovered for air saturation by circulating water and makeup water which is passed to the saturator. The cooled compressed air is further compressed in the high-pressure compressor, cooled in the aftercooler and then fed to the air saturator. The air is contacted over packing with water heated by the various heat sources. The humid air leaving the saturator, after preheating in the gas turbine exhaust, is fed to the combustor.

The hot gas exiting the combustor expands through the gas turbine driving the compressors and providing power. The exhaust heat is then recovered in the recuperator and in the economizer to preheat the water for air saturation (Rao, 1991). The water content increases the mass and the specific heat of the flow, which leads to additional power, and the use of the recuperator gives higher efficiency. By varying the water content the HAT-plant can be put in partload operation without penalizing efficiency, and can also be started up in much shorter time than a conventional combined cycle plant (Nakhamkin and Gulen, 1995).

Several authors reported the net electric efficiency of the HAT cycle. It varies from 54% for a low-pressure ratio turbine (Stecco et al., 1993) to 57% (Chiesa et al., 1995) for a high-pressure cycle, whereas a study by Horlock (1997) suggests that the optimum pressure ratio is rather low, being in the range of 8 to 10.

The cycle does not require expensive steam/water equipment that simplifies the scheme and lowers operating and maintenance costs. For a 300 MW HAT-plant specific capital costs of 400 US dollars per kW have been cited (De Biasi, 1995).

Due to the high moisture content in the working fluid a modified design of expanders is required. Turbo Power and Marine together with Flour Daniel are evaluating FT4000 gas turbine options in the HAT cycle at 200 MW and 55.3% efficiency (Day and Rao, 1993). Further development of the HAT-cycle is being carried out by Westinghouse for W501D5 and W501F gas turbines. Being proposed as the Cascaded Humidified Advanced Turbine (CHAT) cycle, the plant employs proven industrial components, avoiding the complications that high turbine inlet temperatures imply on hardware (Fig. 3.10). Nevertheless, Nakhamkin et al. (1995) reported an efficiency of 55%. When an advanced turbine with a TIT of the LP expander of 1500 °C and an overall pressure ratio of 80 is employed, the cycle efficiency of 63–65% is projected for a CHAT plant of 500 MW (Nakhamkin et al., 1997).

A small-scale CHAT plant of 12 MW is expected to be ready for commercial installation in 1998 with an efficiency of 44.5% and a capital cost of USD 800/kW (Anonymous, 1997).

3.4.5 Semi-closed gas turbine

In a semi-closed gas turbine cycle the gas turbine exhaust is cooled down to the condensation temperature of water vapour, and part of the exhaust gas is recirculated through the compressor (Gasparovich, 1968; Facchini et al., 1996; Nemec and Lear, 1996). The flow scheme is similar to the wet compression cycle (Section 3.4.3), with the addition of a heat recovery steam generator before the condenser-separator, the water injection is optional. Advantages of the semi-closed cycle include combustion at nearly stoichiometric conditions, low emissions, high specific power, good part-load efficiency, and smaller size relative to a conventional combined-cycle plant with an open-cycle gas turbine. Efficiencies reported were 2–3% points lower than those of the reference combined-cycle plants.

3.4.6 Water recovery

The previously reviewed cycles with water/steam injections consume large quantities of water. The typical steam to fuel ratio lies in the range of 2:1 to 4:1. That means a water consumption of 3–5 ton/h for a gas turbine of 25 MW. In addition to the injected water in the gas turbine exhaust, there is also some water in the inlet air and some water formed in the combustion process. If necessary, this water can be reclaimed in a recovery unit where some of the moisture content is condensed, separated from the exhaust gases, treated, and reused (as shown in Fig. 3.8).

To provide a continuous supply of water without the use of external sources, the exhaust must be cooled to 30–40 °C (Nguyen and Den Otter, 1992). Due to the acid content in the flue gases the condensing surfaces should be protected from corrosion, for example by a Teflon coating. A condenser with direct contact or indirect heat exchange can be considered. The performance of a direct contact counter-current con-

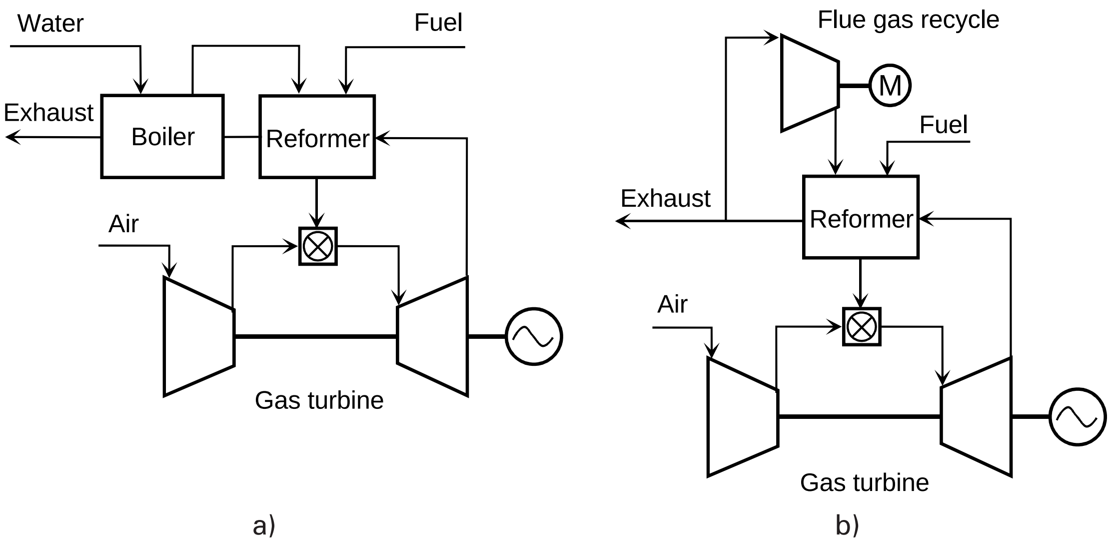

Fig. 3.11. The chemical recuperation in gas turbines:
(a) with steam reforming;
(b) with flue gas recycling.

denser in relation to various factors (water input temperature, water content in the gas turbine exhaust flow, a pinch-point difference of the condenser) can be found in the study by Bettagli and Facchini (1994).

3.4.7 Chemically-recuperated gas turbine

The chemically recuperated gas turbine (CRGT) uses a reforming process to convert methane, water, and sometimes CO_2 into a hydrogen and carbon monoxide fuel mixture that can be burned in the combustor. This endothermic reaction absorbs heat at a temperature lower than the combustion temperature and in this manner increases the fuel's heating value. Recuperation that proceeds thermally and chemically results in a higher degree of heat recovery than in standard recuperation schemes. Moreover, the hydrogen-rich fuel has greater flammability than methane and supports combustion at a lower flame temperature, which potentially reduces NO_X formation. Janes (1990) estimated NO_X production as low as 1 ppm. However, the gas turbine exhaust temperature is not high enough for a complete reforming reaction. At 550 °C only 20% of the fuel is reformed. In order to increase the temperature some additional firing can be applied.

Different reforming schemes have been proposed. In the scheme with steam reforming, steam generated in an HRSG is mixed with natural gas in a reformer (Fig. 3.11a). An analysis of a simple CRGT without intercooling or reheat (Kesser et al., 1994) showed that a power plant based on a LM5000 gas turbine could reach an efficiency of 47% in comparison to 39.5% in simple cycle. This is a little less than that of the conventional combined cycle (47.9%). For the same gas turbine with steam injection efficiency of 46.7% was reported. This indicates that 10–20% of the conversion rate in the reformer is not sufficient to utilize the advantages of chemical recuperation, and it makes the turbine operate in a steam injection mode. Nevertheless, this option was proposed for exhaust heat recovery in remote compressor station applications (Botros et al., 1997).

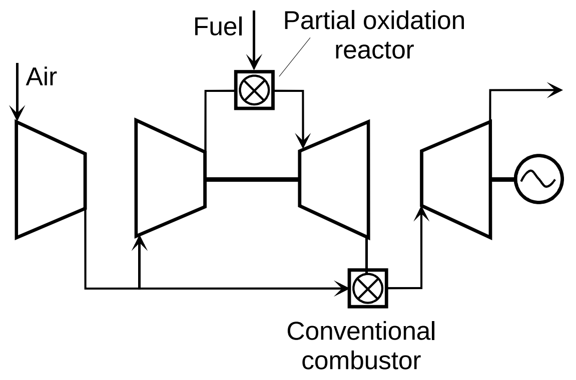

Fig. 3.12. Schematic of the gas turbine with partial oxidation.

Another scheme uses exhaust gas recuperation (Briesch et al., 1995), a portion of the flue gas is compressed, mixed with natural gas, heated with exhaust heat from the combustion turbine, mixed with the air from the compressor, and sent to the combustor (Fig. 3.11b). When the mixture is heated in the presence of a nickel-based catalyst, hydrogen and carbon monoxide are produced. The reaction is accelerated at low excess oxygen, low pressure, and high mass ratio of recycled exhaust gas to methane. Therefore, the best results are achieved when the fuel is burned at the stoichiometric ratio, and the exhaust gas is used to reduce the turbine inlet temperature. Briesch indicates the typical value of recirculation as over 50% of turbine flow. This means that both the air compressor flow and the exhaust flow are less than half that of conventional cycles with the same turbine size.

In a recent paper (Newby et al, 1997) a comparison was given between simple, advanced and combined cycle plants based on the Westinghouse 501F turbine. Efficiency figures were: 35.7% in the conventional simple cycle, 38.7% in the flue-gas reforming simple cycle, 45.6% in the STIG cycle, 48.7% in the steam reforming cycle, 56.8% in the conventional combined cycle, and 57.1% in the combined cycle with flue-gas reforming.

3.4.8 Gas turbines with multi-stage combustion

Approaching isothermal heat addition in a gas turbine can be accomplished by increasing the number of expansion stages, as done in the reheat cycle. Usually, air cooling is used in to maintain the working temperature within the metallurgical limits. This can be alternatively obtained by means of substoichiometric combustion, i.e. in a reheat cycle with fuel excess. The fuel-rich mixture is first partially burned in one or more combustors, and the final oxidation takes place in the last stage combustor (Fig. 3.12). Then, the exhaust heat is recovered in a bottoming cycle. Cancelling the excess air results in a decrease in work required by the compressor, reduced dimensions of the turbomachinery, and low NO_X emissions.

The concept of partial oxidation is implemented in the ‘Chemical’ gas turbine (Arai et al., 1995; Yamamoto et al., 1995). After substoichiometric combustion in the first burner, the hot gas containing unburned fuel expands through the high temperature turbine to the atmospheric pressure. The reduced atmosphere in the first stage allows the

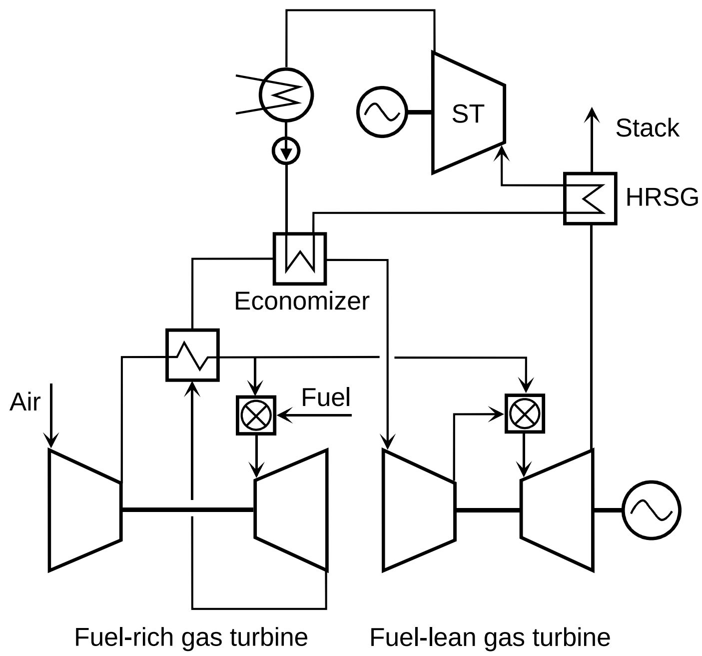

Fig. 3.13. The Chemical gas turbine.

use of carbon-reinforced composites, which can withstand temperatures up to 1700 °C. The gas is then recompressed and passed on to the second, lean-mixture burner. The gas is expanded through the second turbine, and subsequently used to generate steam in the boiler of the Rankine cycle (Fig. 3.13). Lior (1995) found that such a system could reach efficiency level of 66%, assuming a turbine inlet temperature of 1500 °C and an effectiveness of the air-gas heat exchanger of 90%.

Harvey et al., (1995) investigated a three-stage turbine with partial oxidation in the first two stages and chemical recuperation with recycled exhaust gas and water injected between compressor stages. Realistic assumptions made for the equipment characteristics (turbomachinery efficiency of 0.9, a turbine inlet temperature of 1260 °C, a pressure ratio of 20, and minimum ΔT in heat exchangers of 25 K) result in a 65% efficiency for an optimized scheme.

Other multi-stage combustion turbine cycles were proposed in an attempt to approach isothermal heat addition. El-Masri and Magnusson (1984) described a multi-reheat gas turbine that performs the combustion/work extraction at a nearly constant temperature. However, the cycle requires high pressure ratio in the range of 40–100. Calculations made by El-Masri and Magnusson for a plant consisting of a gas turbine with a TIT of 1300 °C, a pressure ratio of 100, and a steam cycle at 650 °C, showed 65% efficiency.

The combustion of hydrogen and oxygen produces steam of high temperature. When this reaction proceeds in a large number of stages nearly isothermal superheating of steam can be achieved. A plant with 15 stages, the maximum temperature of 525 °C, the maximum pressure of 180 bar, and the condensing temperature of 33 °C obtained 49.2% efficiency in calculations by Cicconardi et al. (1998).

The gas turbine cycle with partial oxidation is covered in detail in Chapter 8.

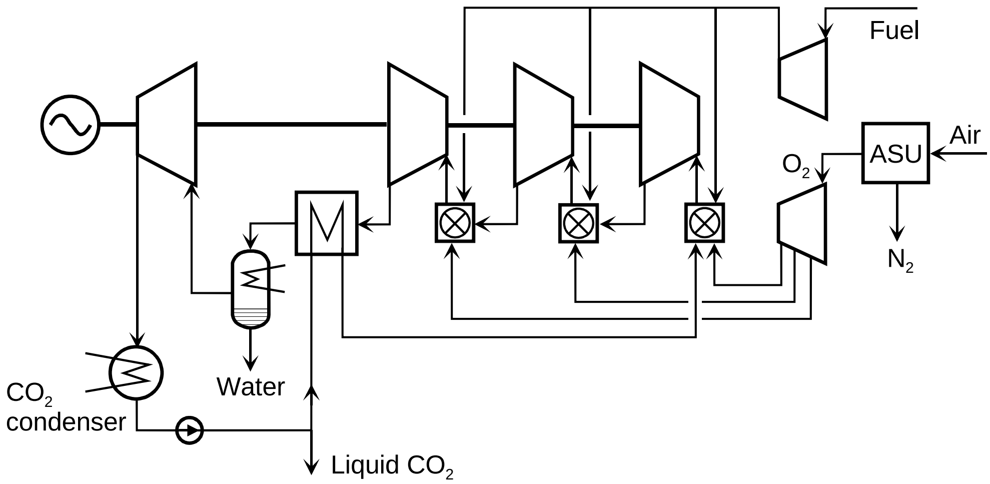

Fig. 3.14. Zero-emission power cycle with CO_2 recirculation.

3.4.9 CO_2 gas turbine

A power plant where CO_2 acts as a working fluid in a gas turbine was proposed by several authors (Yantovski et al., 1992; De Ruyck, 1992; Mathieu et al., 1995). In the example by Yantovski, oxygen from the air separation unit is delivered to the combustors at a pressure of 240 bar, along with the fuel and recycled CO_2. The working fluid expands in the double reheat turbine (240/60/15/4 bar), the exhaust is cooled down in a heat exchanger, and water separation occurs (Fig. 3.14).

In the multi-stage intercooled compressor the fluid is brought to a pressure of 60 bar, and at 20 °C liquefaction takes place. The liquid CO_2 is pumped to a storage area and partially returned to the cycle. Assuming a turbine inlet temperature of 1300 °C, isentropic efficiencies of the gas turbine stages of 80%, 85% and 90%, and an isentropic efficiency of the oxygen compressor of 80%, Yantovski (1994) reported a total plant efficiency of 55%. Since a gas turbine operating at 240 bar and 1300 °C is not feasible, a simplified option with a working pressure of 60 bar was proposed, which can achieve 52% efficiency.

The scheme by De Ruyck employs an evaporative water injection and that by Mathieu a heat recovery steam generator. They operate at pressures of 50–70 bar and have efficiencies of 45–47%. CO_2 storage, such as depleted gas reservoir, must be considered for these cycles.

3.5 Other cycles

3.5.1 Ejector topping power cycle

Instead of expanding hot gases in a turbine, this cycle uses the gases to compress another gas in an ejector. In spite of relatively low efficiency, the ejector can tolerate higher temperatures than a turbine, and it can use working fluids that have better ther-

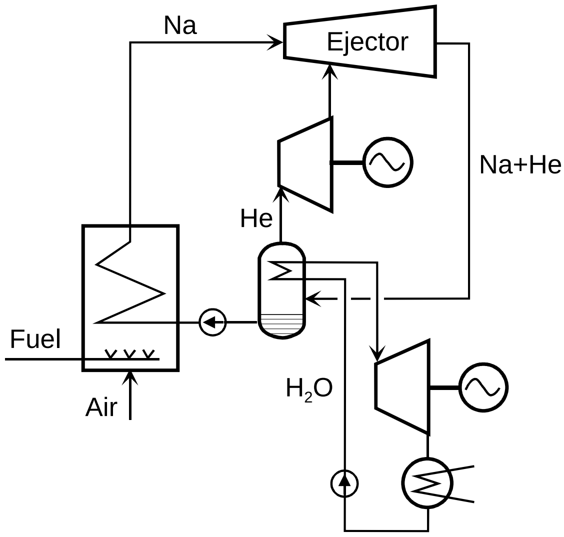

Fig. 3.15. High-temperature ejector topping cycle.

modynamic properties than steam or air. The cycle proposed by Freedman and Lior (1994) uses sodium as the primary fluid, and helium as the secondary one (Fig. 3.15). The system has no moving parts and operates at intermediate pressure. The components can be constructed from ceramic or graphite composites. System efficiency varies with topping boiler conditions. Assuming a steam bottoming efficiency of 40%, at 1700 °C and 50 bar the total efficiency approaches 51%.

3.6 Discussion and conclusions

Among the concepts of the advanced Rankine cycles described in this chapter, three technologies seem of interest in future considerations: the steam recompression and the water flashing as a means of creating a second pressure level in a boiler, and the steam flow splitting as a means of a flexible control at cogeneration plants. The first two are especially attractive at small combined-cycle plants, where a conventional double- and triple-pressure boiler is not economically justified. A performance analysis of such systems showed their competitiveness and feasibility.

The use of the steam flow splitting allows to extend the heat load range without a significant loss in efficiency. This is achieved by recompressing the excess process steam back to initial boiler pressure. The advantages of such an arrangement need to be evaluated in order to make the final conclusions about this concept.

As for the Joule cycle, some enhancements of the gas turbine technology were considered. The steam-methane reforming process, which is well-known in the process industry, can also be applied in a power cycle. The Heron turbine can be considered as a good candidate for chemical recuperation, since it has relatively high temperature of the power turbine exhaust (above 600 °C) which is favorable for the reforming process. Moreover, reforming with CO_2-recycle appreciably diminishes the discharge of this gas to the atmosphere. These chemically-recuperated cycles are regarded as optional within the NECT programme.

The HAT cycle has acquired much attention since 1990s, and several dedicated gas turbine designs have been developed by gas turbine manufacturers (Day and Rao, 1993; Nakhamkin et al., 1995). The cycle features the absence of the bulky steam equipment (boiler, steam turbine, condenser), a good part-load performance, and a very low level of NO_X emissions. Nevertheless, the HAT cycle is not considered in the future work because of the advances in the cycle development reached by other research groups

A less developed cycle is the cycle with partial fuel oxidation in a multi-stage gas turbine. The process of substoichiometric combustion can approach isothermal heat addition, reduce NO_X formation, and provide a compact design due to the absence of the excess air. Reported high efficiency of 60–66% suggests that the cycle deserves further consideration. A performance analysis of this cycle is given in Chapter 8.

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Chapter 4
Industrial Cogeneration

Abstract

Combined generation of heat and power employs various schemes to satisfy district heating or process needs. A number of performance assessment parameters are used to compare the schemes with each other and with separate generation plants. Criteria such as the fuel utilization efficiency, fuel saving, total fuel consumption, and exergetic efficiency are discussed. Several typical industrial cogeneration schemes, which provide process steam of 210 °C and 10 bar, are assessed to quantify the differences between the options using different methodologies. The plants under consideration include conventional boiler, gas turbine schemes with a heat recovery steam generator and a back-pressure steam turbine, and electric and heat-activated heat pumps. In the case of the gas turbine-based schemes, additional options with supplementary firing to double steam production are also examined. The gas turbine-based schemes were found the most efficient in fuel-saving and exergy terms. The fuel savings of the case with the lowest heat-to-power ratio appeared the most susceptible to an improvement in utility efficiency. A comparison between the fuel energy utilization efficiency and exergetic efficiency for the cases considered is given.

4.1 Introduction

Combined production of heat and power (CHP) is generally regarded as an environmentally and economically attractive energy technology and is widely used for process and space heating needs. The last decade has witnessed a rapidly growing number of cogeneration units. Current CHP penetration in the European market averages 9%, and the European Commission would like this to reach 18% by 2010 (Anonymous, 1997). In The Netherlands the amount of electricity produced at the CHP installations has grown from 9% of the total generated power in 1980 to 30% in 1997, providing 6500 MWe (Holwerda, 1997). Further increase of the cogeneration share in the total power production is expected. Obviously, this considerable amount of power should be generated efficiently. To evaluate the performance of a CHP plant an objective criterion is required.

4.2 Performance criteria

Combined generation of heat and power implies production of two different kinds of energy. A common denominator should first be established to analyze the efficiency of a cogeneration plant. A number of criteria can be used to assess the plants's performance, as summarized in Huang (1996) and Horlock (1997). Of those parameters, the most frequently used are: (1) fuel utilization, or energy efficiency, (2) 'factored' efficiency; (3) exergetic efficiency, and (4) avoided fuel costs, or fuel energy savings.

The first, the most straightforward criterion is based on the First Law, which deals only with the quantitative side of energy. It is defined as:

$$\eta_{FU} = \frac{Q_{CG} + P_{CG}}{F_{CG}} \tag{4.1}$$

Since the fuel utilization efficiency gives the same value to work and heat, it cannot provide an adequate performance figure. This was shown, for example, by Huang (1990) and Sarabchi (1992) for a basic gas turbine cogeneration plant. This drawback has led to the introduction of 'factored' efficiency, where a weight factor is given to heat. The factor is determined by economics as the price ratio of heat to power, which is usually taken as 1/3 (Timmermans, 1978; Sarabchi and Polley, 1995); or by politics, such as the PURPA efficiency (derived from the Public Utility Regulatory Policy Act) that gives heat a factor of 0.5 (Huang, 1996). The price ratio reflects the present day economy, in which heat is mainly produced by direct combustion boilers. The increasing number of cogeneration plants, which consume less fuel per unit of heat, in a free market would result in a different heat-to-power price ratio.

Thermodynamics suggests the use of the exergetic factor which exactly indicates the quality of heat in terms of its work potential. An even more correct performance value is obtained if the exergy content of fuel is also taken into account:

$$\eta_{EX} = \frac{P_{CG} + E^{Q}_{CG}}{E^{F}_{CG}} \tag{4.2}$$

Introducing the Second Law analysis has proved to be a valuable methodology for evaluation (Kotas, 1985; Habib, 1994; Horlock, 1997) and optimization of cogeneration plants (Pak and Suzuki, 1997). After first being used as an academic criterion, exergetic efficiency is now becoming more often used by engineering companies and industries. Exergy was used, for example, in a study on global CHP integration in an area where various industries and dwellings for 600 000 inhabitants were located (Haskoning, 1994); as well as to compare steam of different parameters in evaluation of internal steam prices within a chemical concern (Donders, 1998).

The fourth criterion, the fuel savings, deals with the fuel consumption of the plants. The demand for heat and power at an industrial site can be satisfied either by

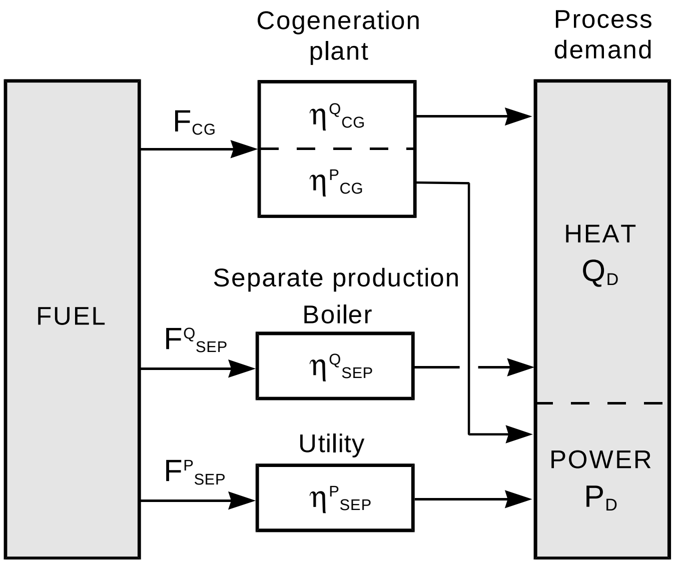


Fig. 4.1. Cogeneration versus separate production

a cogeneration plant, or by a separate production scheme, or both, if the cogeneration plant cannot meet heat or power demands (Fig. 4.1). Here, thermal efficiency η^Q is the ratio of heat produced to the fuel input, and electric efficiency η^P is the ratio of power to the fuel input of a plant. Although the thermal efficiency does not reflect the quality of heat produced, it is still suitable for comparison studies, if all plants compared generate heat of the same quality. The general expression for the total fuel consumption is

$$F_{TOTAL} = (F^P_{SEP} + F^Q_{SEP}) + F_{CG} \tag{4.3}$$

The first two terms indicate the amount of fuel consumed by the separated scheme in order to fulfill the demand, if the cogeneration plant is unable to supply all heat and power required. The fuel consumed by the cogeneration plant can be expressed either as the ratio of the heat produced to the thermal efficiency, or as the ratio of the power produced to the electric efficiency. By introducing thermal and electric efficiencies for cogeneration and separate plants, and two heat-to-power ratios, one for the cogeneration plant λ_{CG}, and one for the demand side λ_D, Eqn. 4.3 becomes

$$\begin{aligned} F_{TOTAL} &= \frac{P_D - P_{CG}}{\eta^P_{SEP}} + \frac{Q_D - Q_{CG}}{\eta^Q_{SEP}} + \frac{Q_{CG}}{\eta^Q_{CG}} \\ &= \frac{1}{\eta^P_{SEP}} \cdot \left(\frac{1}{\lambda_D} Q_D - \frac{1}{\lambda_{CG}} Q_{CG} \right) + \frac{Q_D - Q_{CG}}{\eta^Q_{SEP}} + \frac{Q_{CG}}{\eta^Q_{CG}} \end{aligned} \tag{4.4}$$

If the fraction of heat generated by a CHP plant with respect to the heat required is designated as y, then the expression for F_{TOTAL} can be written as

$$F_{TOTAL} = Q_D \cdot \left(\frac{1}{\eta^P_{SEP}} \cdot \left(\frac{1}{\lambda_D} - \frac{y}{\lambda_{CG}} \right) + \frac{1-y}{\eta^Q_{SEP}} + \frac{y}{\eta^Q_{CG}} \right) \tag{4.5}$$

If the cogeneration plant supplies all heat required, i.e. $y = 1$, Eqn. 4.5 is reduced to

$$F_{TOTAL} = Q_D \cdot \left(\frac{1}{\eta^P_{SEP}} \cdot \left(\frac{1}{\lambda_D} - \frac{1}{\lambda_{CG}} \right) + \frac{1}{\eta^Q_{CG}} \right) \tag{4.6}$$

For the specific case of a heat pump, the term $1/\lambda_{CG}$ is omitted, because the heat pump does not produce any power.

The total fuel consumption can be compared with the reference case, where all required heat and power are generated separately. An expression of the relative fuel consumption factor can be defined as:

$$\Phi = \frac{F_{TOTAL}}{F_{REF}} = \frac{\frac{1}{\eta^P_{SEP}} \cdot \left(\frac{1}{\lambda_D} - \frac{1}{\lambda_{CG}} \right) + \frac{1}{\eta^Q_{CG}}}{\frac{1}{\eta^P_{REF}\lambda_D} + \frac{1}{\eta^Q_{REF}}} \tag{4.7}$$

Whereas the relative fuel consumption factor can assess any combination of combined and separate production of heat and power (i.e. when $\lambda_{CG} \neq \lambda_D$), it is also of interest to compare a cogeneration scheme with a reference separate production scheme in the case when both supply the same amount of heat. Thus, the difference in fuel consumption is the fuel savings of the CHP plant:

$$\Delta F = (F^Q_{SEP} + F^P_{SEP}) - F_{CG} \tag{4.8}$$

The first term is a sum of the amount of fuel consumed by the reference boiler plant F^Q_{SEP}, and the fuel that would be consumed by a utility to generate power equal to that of the CHP plant, F^P_{SEP}. The second term is the fuel consumed by the cogeneration plant, F^Q_{CG}. Introducing efficiencies, the fuel savings becomes:

$$\Delta F = F^Q_{SEP} + \frac{\eta^P_{CG}}{\eta^P_{SEP}} F_{CG} - F_{CG} = Q_D \cdot \left(\frac{1}{\eta^Q_{SEP}} + \frac{1}{\eta^Q_{CG}} \cdot \left(\frac{\eta^P_{CG}}{\eta^P_{SEP}} - 1 \right) \right) \tag{4.9}$$

Since ΔF is an absolute value, the fuel energy saving ratio (FESR) is defined as the ratio of savings to the separate scheme's fuel consumption (Horlock, 1997):

$$FESR = \frac{\Delta F}{F^Q_{SEP} + F^P_{SEP}}$$

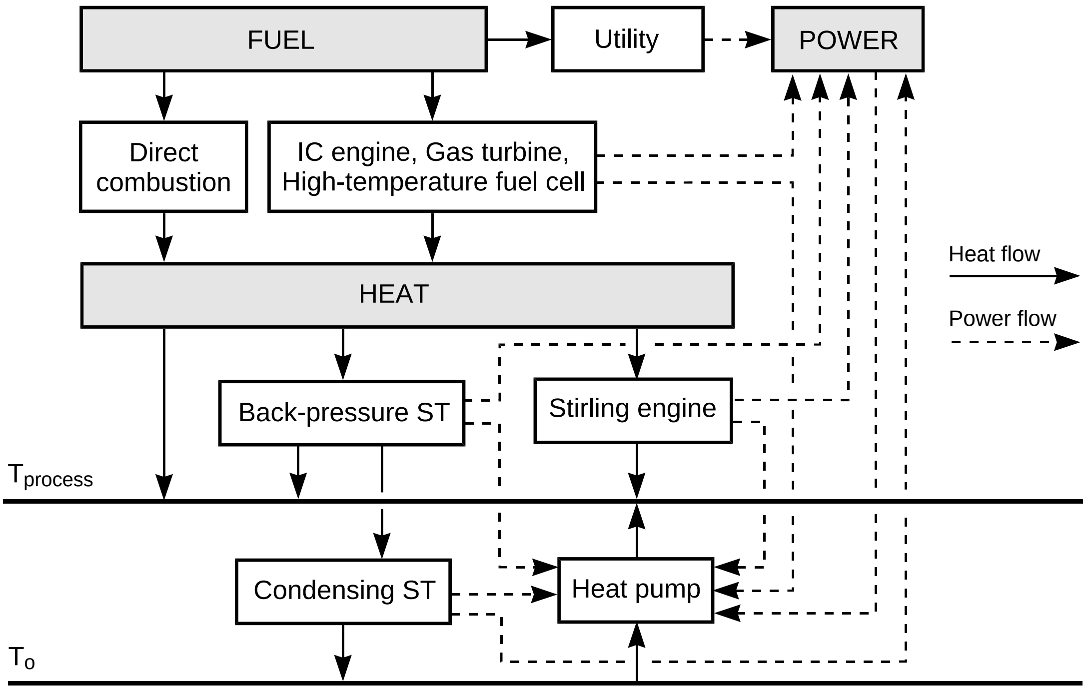


Fig. 4.2. Possible cogeneration combinations.

$$FESR = \left(\frac{1}{\eta^Q_{SEP}} + \frac{1}{\eta^Q_{CG}} \cdot \left(\frac{\eta^P_{CG}}{\eta^P_{SEP}} - 1\right)\right) \bigg/ \left(\frac{1}{\eta^Q_{SEP}} + \frac{\eta^P_{CG}}{\eta^Q_{CG} \cdot \eta^P_{SEP}}\right)$$
$$= 1 - \frac{\eta^Q_{SEP} \cdot \eta^P_{SEP}}{\eta^Q_{CG} \cdot \eta^P_{SEP} + \eta^Q_{SEP} \cdot \eta^P_{CG}} \qquad (4.10)$$

In the case, when heat and power demands are “matched”, i.e. $F_{TOTAL} = F_{CG}$, the relation between FESR and Φ is simply:

$$FESR = \frac{F_{REF} - F_{CG}}{F_{REF}} = 1 - \frac{F_{CG}}{F_{REF}} = 1 - \Phi \qquad (4.11)$$

A critical condition, when fuel savings is zero, can be obtained from Eqn. 4.10:

$$\frac{\eta^P_{CG}}{\eta^P_{SEP}} + \frac{\eta^Q_{CG}}{\eta^Q_{SEP}} = 1 \qquad (4.12)$$

Considering efficiencies of the separate production as constant, the critical cogeneration efficiency can be derived (Korobitsyn and Hirs, 1995), and a zero-saving line can be plotted in the coordinates of the thermal and electric efficiencies of the cogeneration plant (Wiemer, 1997).

The last decade has been marked by an increasing number of highly efficient combined-cycle power plants being put into operation; hence it seems necessary to evaluate the influence of an improvement in separate electric production (utility) efficiency on the fuel savings of a CHP plant.

Table 4.1. Performance specifications of the LM6000 gas turbine.

Parameters:		**Exhaust gas composition:**	
Electric power	38.4 MWe	O_2	14.29%
Exhaust flow	125.1 kg/s	CO_2	2.97%
Exhaust temperature	462°C	N_2	76.08%
Simple-cycle efficiency	37.88%	H_2O	6.66%

4.3 Configurations

Modern power technology offers many possibilities to realize the cogeneration concept via different routes: from fuel as a source to heat as a sink, by direct combustion or by multi-level cascading (Fig. 4.2). Of these options, a number of typical schemes were chosen for an analysis as listed below:

CB	Conventional boiler [10 bar], the reference case.
CB-ST	Conventional boiler [80 bar] with back-pressure steam turbine.
GT-B	Gas turbine with single pressure HRSG [10 bar].
GT-B-ST	Gas turbine with single pressure HRSG [80 bar] and back-pressure steam turbine.
GT-B2-ST	Gas turbine with dual pressure HRSG [10/80 bar] and back-pressure steam turbine.

Each of these delivers 40 t/h of superheated process steam [10 bar, 203°C]. The steam turbine has an isentropic efficiency of 80% and expands steam from 80 bar/430 °C to 10 bar/203 °C. Thermal efficiency of the reference conventional boiler was set to 95%. A heat recovery steam generator features the approach and pinch temperature of 10 K. All systems comprise a deaerator [1.2 bar]. It was assumed neither boiler blow-down, nor deaerator vent flows, and that the return flow constituted 100% of the process flow. The flow diagrams of the cases are presented in Figures 4.3 and 4.4. In addition to these, three modifications with a duct burner were also considered:

GT-BF	the same as case GT-B, with supplementary firing
GT-BF-ST	the same as case GT-B-ST, with supplementary firing
GT-B2F-ST	the same as case GT-B2-ST, with supplementary firing

Calculations were made using Dutch natural gas as a fuel with a lower heating value of 38 MJ/kg. As a process simulator, the heat-balance package Gate Cycle was used (Enter Software, 1995). The gas turbine simulation was based on the performance data of the General Electric LM6000 engine (Table 4.1).

Also, two heat pump systems were considered:

EHP	Electrically-driven heat pump system.
HAHP	Heat-activated heat pump system.

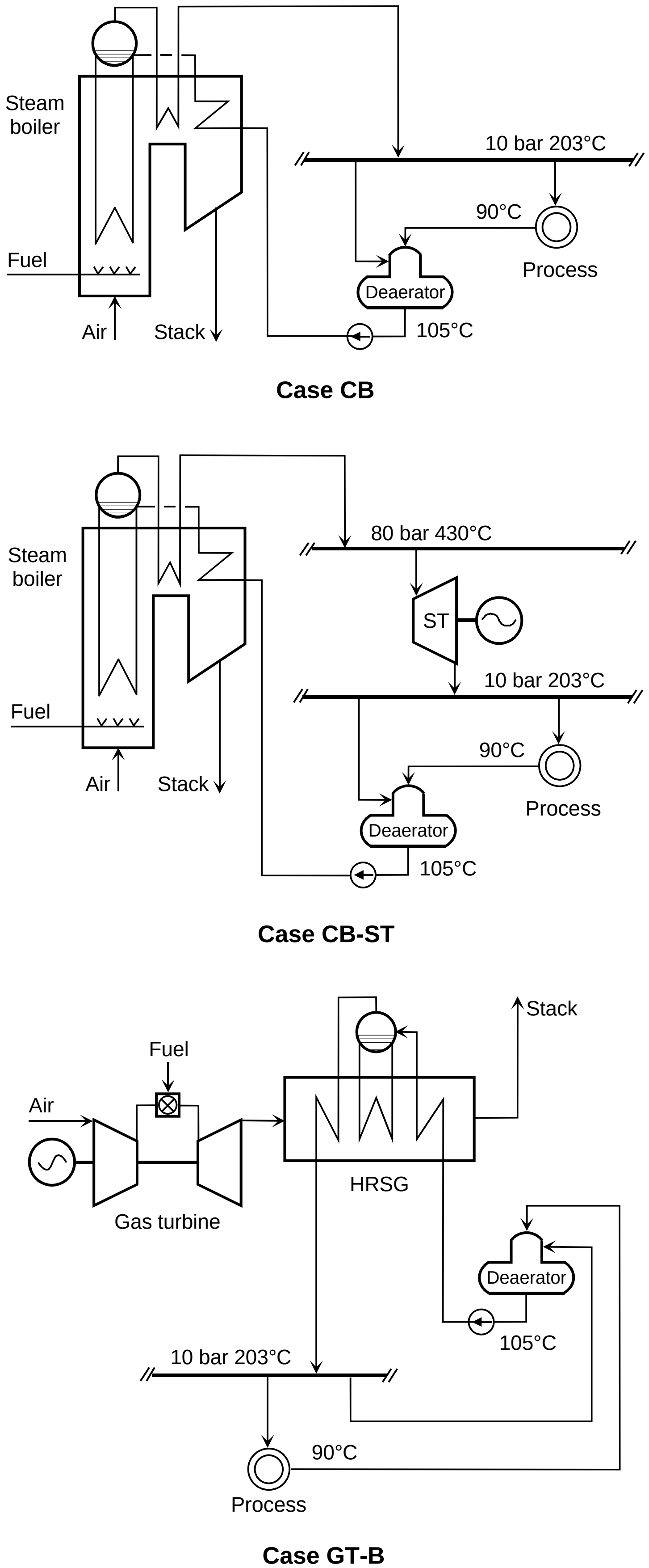


Fig. 4.3. Flow diagrams of Cases CB, CB-ST, GT-B.

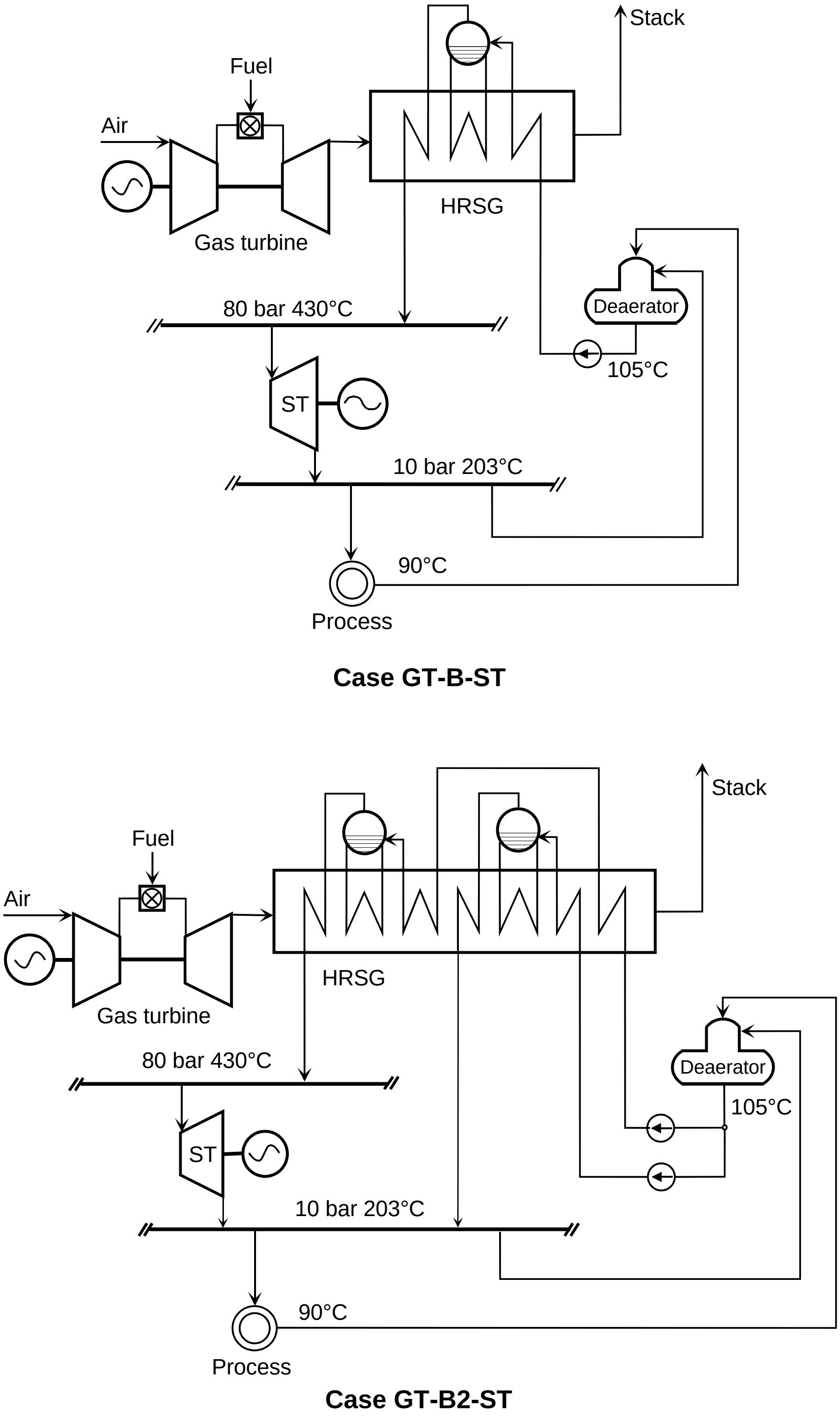


Fig. 4.4. Flow diagrams of Cases GT-B-ST and GT-B2-ST.

Both heat pump systems were based on the same vapor compression cycle (Fig. 4.5). COP of the cycle is a function of the Carnot factor, temperature differences in the heat acceptor and rejector, and the process demand and return temperatures:

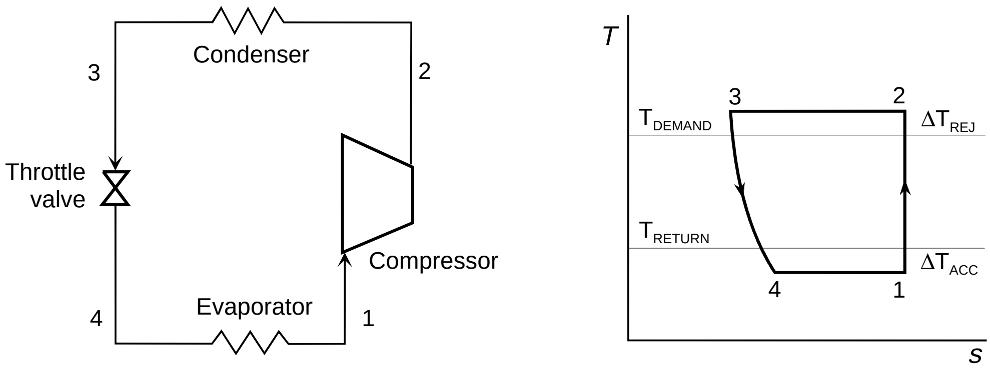


Fig. 4.5. Schematic and T-s diagram of the vapor compression heat pump.

$$COP_{HP} = 1 + f_{CARNOT} \cdot \left(\frac{T_{RETURN} - \Delta T_{ACC}}{(T_D + \Delta T_{REJ}) - (T_{RETURN} - \Delta T_{ACC})} \right) \tag{4.13}$$

By taking 0.6 as the Carnot factor, which is typical for such a cycle (Walker et al., 1994) and ΔT of 10 K, a COP of 2.5 is obtained for the heat pump cycle operating within the process temperature range (90 °C and 203 °C). Regarding system COPs, the following factors were taken into account. For the electric heat pump: electric efficiency of the separate power production (utility efficiency) and efficiency of the electric driver. For the heat-activated heat pump: efficiency of the driver or the absorption unit, and utilization of its waste heat, which adds to system COP.

Therefore, assuming an electric driver efficiency of 0.95, and utility efficiency levels of 0.4, 0.5 and 0.6, the overall COP of the electric heat pump system at these levels becomes 0.95, 1.19, and 1.42, respectively. For the heat-activated heat pump, which is driven by a prime mover (such as a gas engine) at an efficiency of 0.3, and includes the utilization of waste heat, a system COP of 1.25 is obtained. A similar COP value is representable for the absorption heat pump system.

4.4 Performance analysis

The analysis was based on a constant steam demand of 40 t/h. When a simulation model had a different steam production, fuel consumption and generated power were linearly scaled to fulfill the steam demand. Then, all options were compared with the conventional boiler as a reference. This analysis was also applied to a case of doubling steam production by means of supplementary firing (HRSG options). In this case, the flow values were also scaled down to the constant steam supply of 40 t/h. A summary of results is given in Table 4.2. The table lists the following parameters: heat and power production, and fuel consumption (in thermal and exergetic megawatts); the stack temperature; thermal and electric efficiencies; heat-to-power ratio; thermal and electric efficiencies based on exergy values; the fuel energy saving ratio and the relative fuel consumption factor. The latter two parameters were calculated for three levels of utility efficiency (0.4, 0.5 and 0.6).

Table. 4.2. Main results of the simulations.

			CB	CB-ST	GT-B	GT-BF	GT-B-ST	GT-BF-ST	GT-B2-ST	GT-B2F-ST	EHP	HAHP
Heat		MW.th	27.360	27.360	27.360	27.360	27.360	27.360	27.360	27.360	27.360	27.360
		MW.ex	9.026	9.026	9.026	9.026	9.026	9.026	9.026	9.026	9.026	9.026
Power		MW.el	-	4.225	24.280	12.100	42.700	23.500	30.750	17.700	-	-
Fuel		MW.th	28.717	33.269	64.143	45.450	101.588	64.975	73.111	52.188	23.030*	21.888
		MW.ex	29.808	34.533	66.580	47.177	105.448	67.444	75.889	54.171	23.905*	22.720
Stack		°C	116	121	148	141	235	208	155	144	-	-
η^Q			0.953	0.822	0.427	0.602	0.269	0.421	0.374	0.524	1.188*	1.250
η^P			-	0.127	0.379	0.266	0.420	0.362	0.421	0.339	-	-
λ_{CG}			-	6.476	1.127	2.261	0.641	1.164	0.890	1.546	-	-
FUE			0.953	0.949	0.805	0.868	0.690	0.783	0.795	0.863	1.188*	1.250
η^Q_{EX}			0.303	0.261	0.136	0.191	0.086	0.134	0.119	0.167	0.378*	0.397
η^P_{EX}			-	0.122	0.365	0.256	0.405	0.348	0.405	0.327	-	-
η_{EX}			0.303	0.384	0.500	0.448	0.491	0.482	0.524	0.493	0.378*	0.397
FESR	0.4		-	0.153	0.283	0.229	0.250	0.257	0.308	0.285	-	-
at η^P_{SEP}:	0.5		-	0.105	0.170	0.141	0.110	0.142	0.190	0.186	-	-
	0.6		-	0.070	0.073	0.070	- 0.017	0.043	0.086	0.104	-	-
Φ (λ_D=5)	0.4		-	0.858	0.404	0.681	0.201	0.469	0.234	0.510	1.002	0.839
at η^P_{SEP}:	0.5		-	0.902	0.669	0.812	0.684	0.729	0.569	0.699	0.857	0.828
	0.6		-	0.934	0.867	0.909	1.045	0.923	0.819	0.841	0.748	0.820
Φ (λ_D=0.5)	0.4		-	0.964	0.847	0.918	0.795	0.864	0.804	0.874	1.001	0.959
at η^P_{SEP}:	0.5		-	0.972	0.905	0.946	0.909	0.922	0.876	0.914	0.959	0.951
	0.6		-	0.979	0.958	0.971	1.014	0.976	0.943	0.950	0.921	0.943

Notes:

*) Based on a value of utility efficiency η^P_{SEP} of 0.5.
Values of the fuel energy saving ratio FESR and the relative fuel consumption factor Φ are given for three levels of utility efficiency (0.4, 0.5 and 0.6).

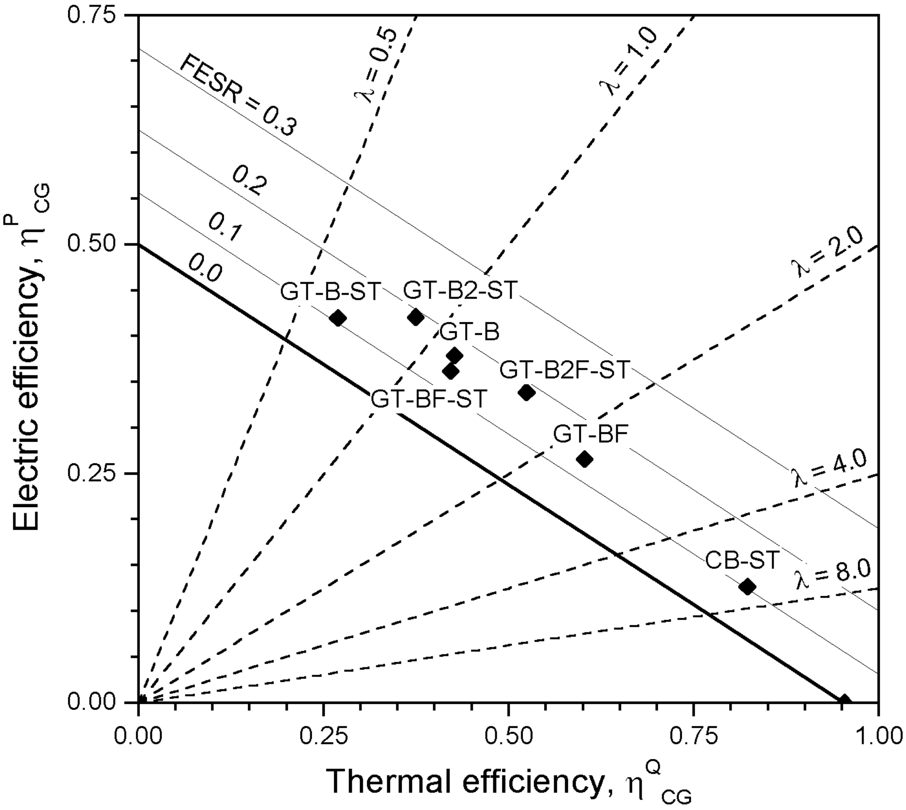
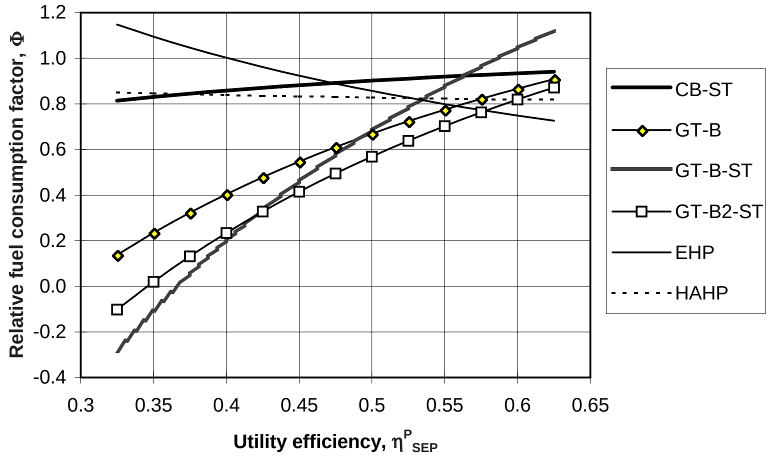


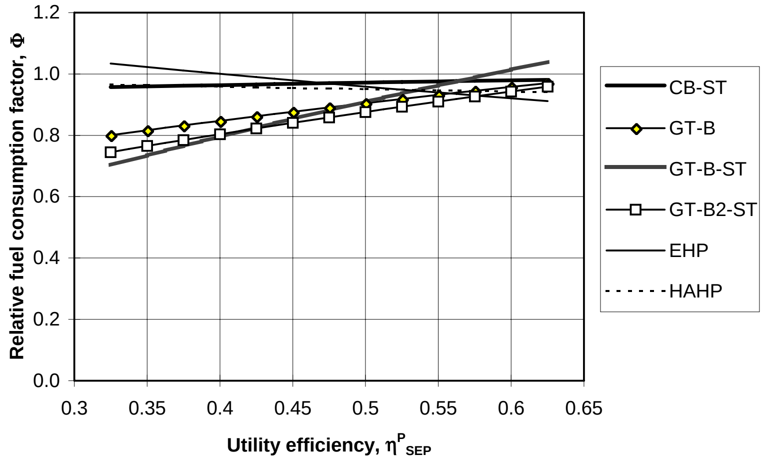
Fig. 4.6. Fuel energy saving ratio of the cogeneration options.

It can be seen from the table that case HAHP uses the least, and case GT-B-ST the most fuel to fulfill the heat demand. At the same time, case GT-B-ST generates the most power with $\lambda_{CG} = 0.64$. The options based on the conventional boiler and heat pumps show the highest fuel utilization efficiency, but when evaluated in exergetic terms, appeared to have the lowest values. At a utility efficiency of 0.5, all cogeneration options obtain FESR values between 0.1 and 0.2 (Fig. 4.6). The options with the highest FESR are the gas turbine-based plants with a double-pressure HRSG and a steam turbine, GT-B2-ST and its modification with supplementary firing GT-B2F-ST. When the cases are examined under different levels of utility efficiency η^P_{SEP}, some sensitivity of the options with a high heat-to-power ratio is revealed. In Fig. 4.6 the line of zero FESR value becomes steeper, and affects these cases most. It is especially apparent in case GT-B-ST: while being quite beneficial at $\eta^P_{SEP} = 0.4$, its FESR value becomes negative at $\eta^P_{SEP} = 0.6$ (Table 4.2). Having the lowest heat-to-pressure ratio, caused by exclusively generating high-pressure steam in HRSG for the steam turbine, the case should compete with utility in electric efficiency, but a value of 0.42 is not sufficient for this condition. For case CB-ST with the highest λ_{CG}, the improvement in utility efficiency is much less noticeable.

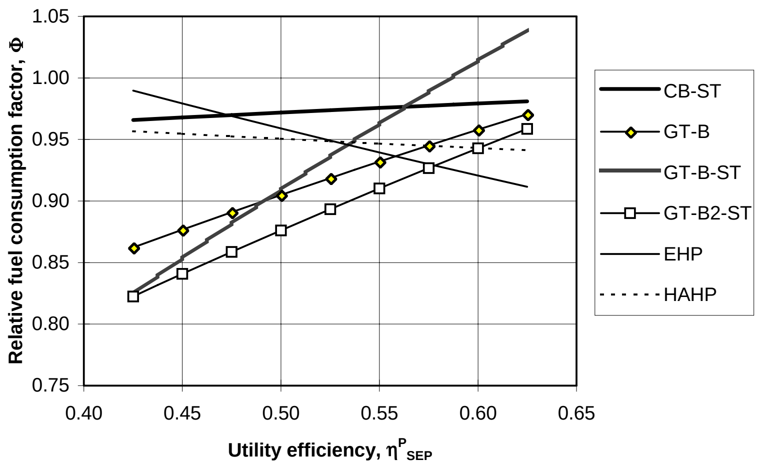
Regarding FESR, no great differences were observed between unfired and firing cases. In the configuration of a gas turbine with a single-pressure HRSG and a steam turbine, case GT-B-ST, the fired option shows a higher ratio of fuel savings, and sup-



a)



b)



c)

Fig. 4.7. Influence of utility efficiency on the relative fuel consumption for the demand heat-to-power ratio: (a) $\lambda_D = 5$; (b) and (c) $\lambda_D = 0.5$.

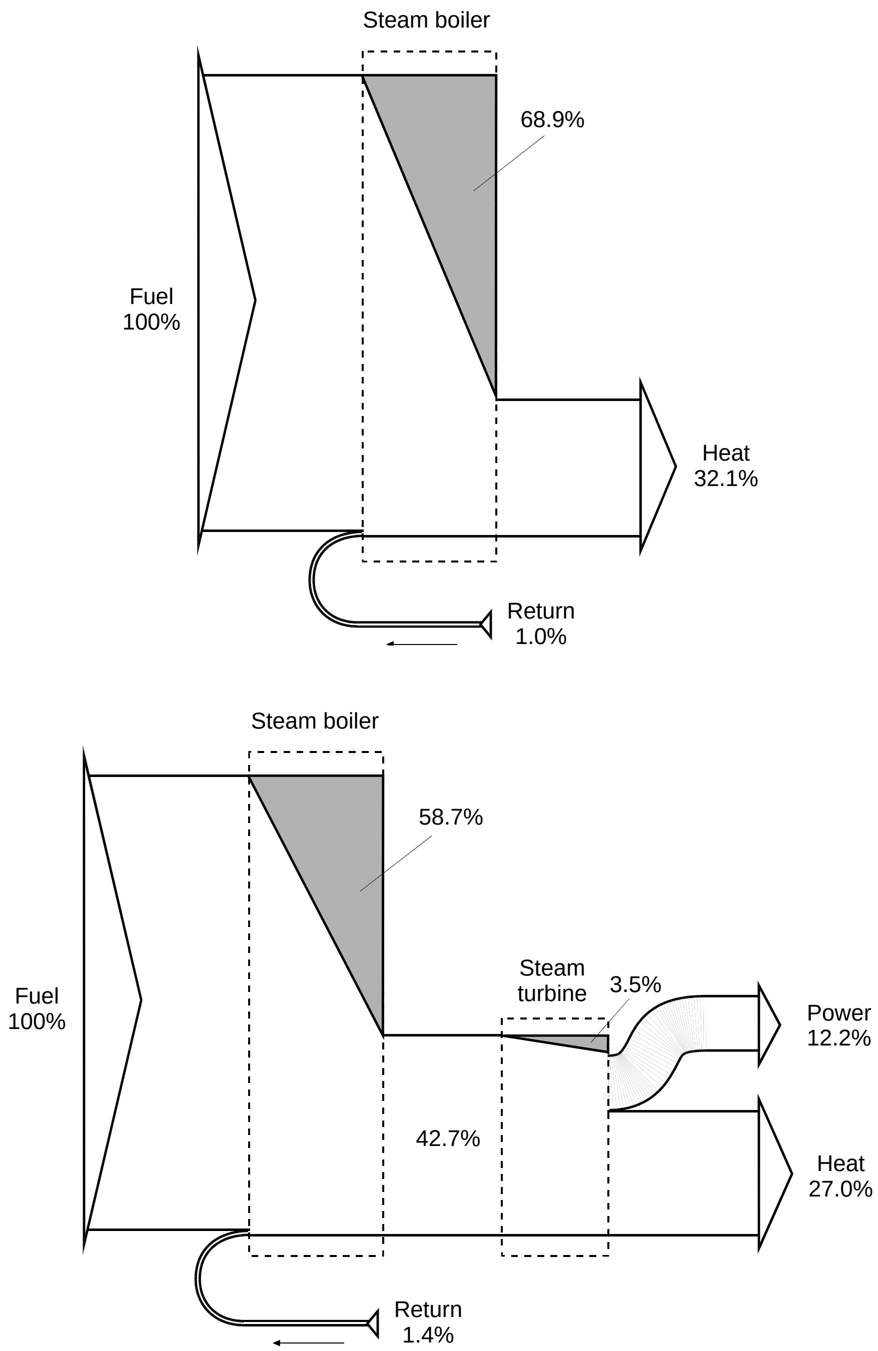


Fig. 4.8. Grassmann diagrams of cases CB and CB-ST.

plementary firing has some positive effect in reducing the stack temperature from 235 to 208 °C (case GT-BF-ST).

The influence of utility efficiency on the relative fuel consumption is presented in Figure 4.7 for two conditions. The first one is the situation when only local process needs are satisfied. Usually, the heat-to-power ratio λ_D is about 4 to 5 (Wiemer, 1997). At a typical value of 5 (Fig. 4.7a), a clear advantage of gas turbine-based plants are noted, especially at lower utility efficiency values. Their performance is so good that they consume only 20 to 40% fuel of the reference scheme, and at $\eta^P_{SEP} = 0.35$, the

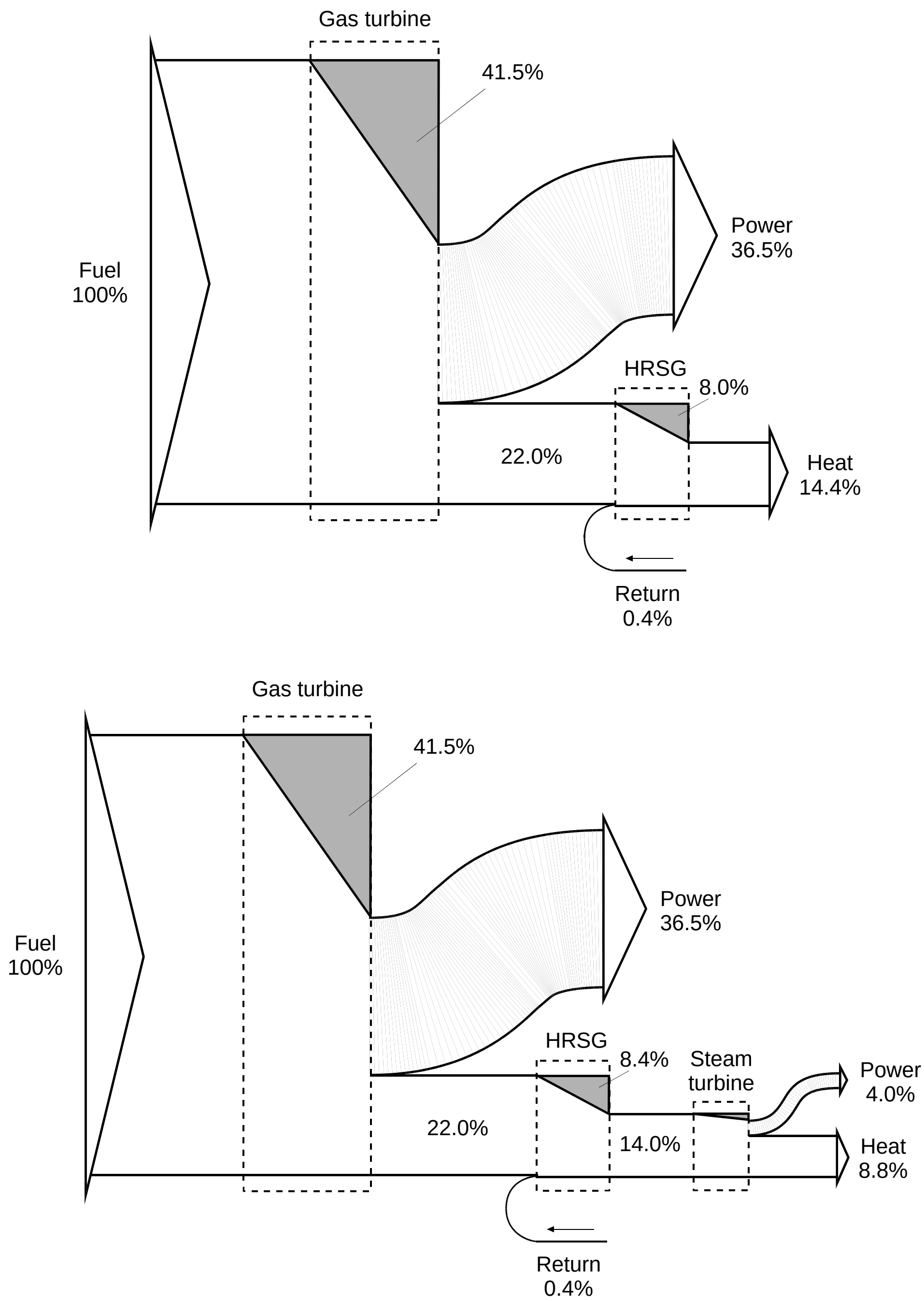


Fig. 4.9. Grassmann diagrams of cases GT-B and GT-B-ST.

consumption factor even becomes negative, because excessive power production in these cases is performed at electric efficiencies higher than that of the utility.

The heat-activated heat pump has a value of factor Φ that is independent from utility efficiency, and it remains steady at 82-84% of the fuel consumption of the reference. But this is not the case for the electric heat pump: it becomes advantageous only when η^{P}_{SEP} exceeds a value of 0.4, and above 0.6 EHP turns out to be the most efficient of the other options. The conventional boiler with a steam turbine stands out above the reference scheme, but its advantage is rather minor: it consumes just 86% to 93% of the reference.

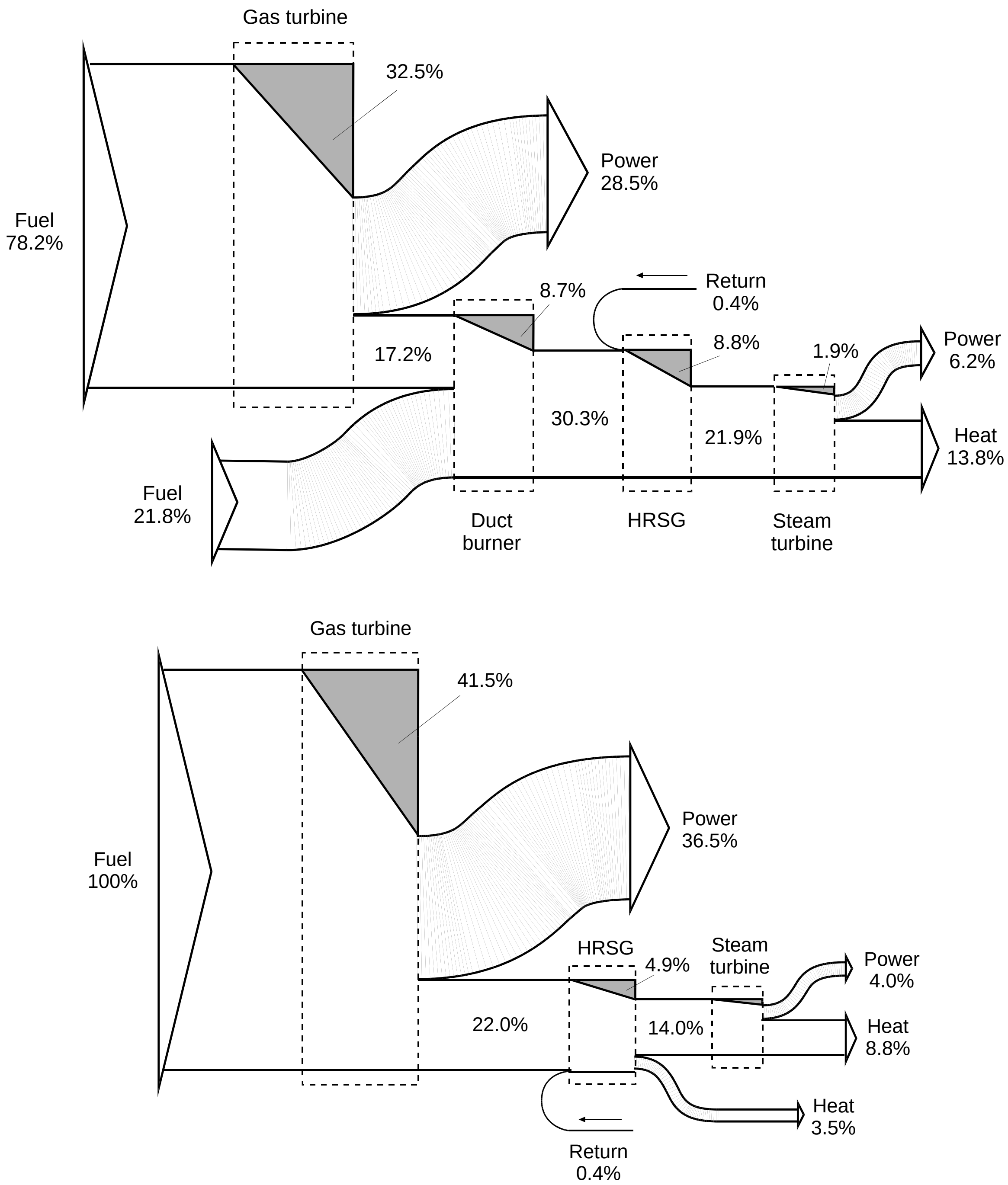


Fig. 4.10. Grassmann diagrams of cases GT-BF-ST and GT-B2-ST.

As has already been noted before, case GT-B-ST loses its advantage at a utility efficiency of 0.553.

When more power than heat is required, i.e. λ_D is, for example, 0.5, the CHP plants' advantage is no longer so profound. A look at Fig. 4.7b shows a small difference among them and a comparable value of the relative fuel consumption. Still, the gas turbine plants offer better performance (80% to 90% of the reference's fuel consumption), intersecting other options' lines at η^{P}_{SEP} of about 0.55-0.6, as an enlarged view shows (Fig. 4.7c). Similar trends are observed in Figures 4.7a and 4.7c.

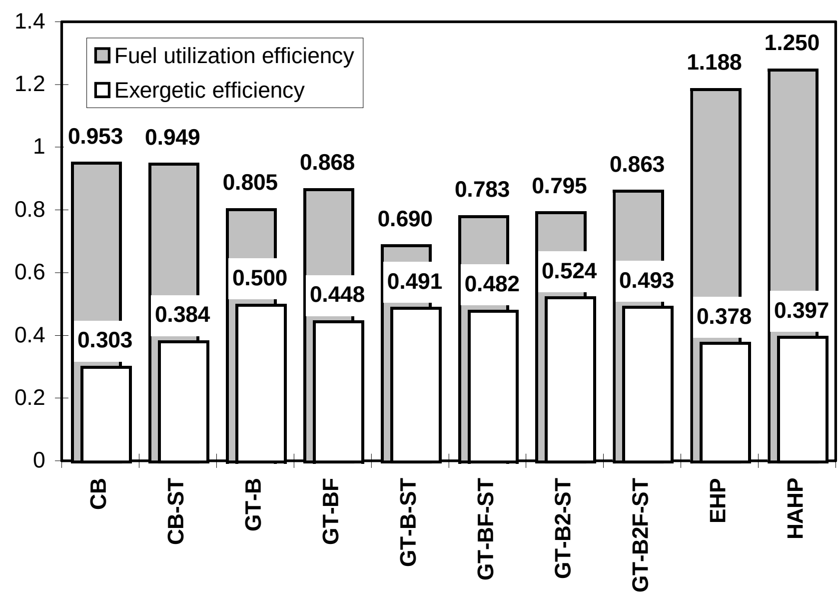


Fig. 4.11. Comparison of fuel utilization efficiency and exergy efficiency for the options considered

4.5 Exergy analysis

Exergy analysis, applied to a cogeneration plant as a whole, was also used to analyze performance of separate components of a power plant. The plant was divided into black boxes, such as: a boiler, a gas turbine, a duct burner, HRSG, and a steam turbine.

The following assumptions were made:

(1) only chemical exergy was used for fuel,

(2) only physical exergy was used for flue gas and water/steam flows.

Exergy flows through the power plants are presented in Grassmann diagrams (Figures 4.8 to 4.10). Substantial exergy losses can be seen to occur in the conventional boiler. Depending on the steam pressure, the losses vary from 57% to 67%. In a gas turbine the losses amount to 41.5%. Introducing supplementary firing deteriorates exergy efficiency in all gas turbine plants.

Regarding the performance of HRSG, an equal exergy ratio for HRSG is noted in options GT-B and GT-B-ST, whereas option GT-B-ST generates steam of a higher exergy value. That is explained by the lower steam production of case GT-B-ST. Although the mass flow of high-pressure steam in case GT-B-ST accounts for 63% of the flow in case GT-B, the higher exergetic quality of this steam makes it comparable with the low-pressure steam flow in GT-B configuration.

The final chart (Fig. 4.11) presents a comparison of overall exergy efficiency versus fuel utilization efficiency (the First Law efficiency). It confirms the evident advantage of the gas turbine plant in general, and, of the unfired double-pressure HRSG scheme, in particular. The efficiencies of the gas turbine alternatives lie in the range of 50% with two extremes: option GT-BF (single-pressure HRSG with supplementary firing and without steam turbine), which has an efficiency of 45%, and option GT-B2-ST (double-pressure unfired HRSG and steam turbine) with an efficiency of 52.4%.

The heat pump systems proved to have a performance comparable to a simple cogeneration scheme using a conventional boiler with a steam turbine. Their exergetic efficiencies lie between 38% and 40%. These figures would be higher at a smaller temperature difference between the process and return flows.

A remarkable difference can be seen between energy and exergy efficiencies, and even an opposite trend for most of the options should be noted: while exergy efficiency rises, energy efficiency decreases. This suggests the general use of the exergy criterion for assessing cogeneration plants.

4.6 Conclusions

Different methodologies demonstrated the apparent benefits of the gas turbine plant featuring the dual-pressure unfired HRSG concept in comparison with the other options considered. This scheme shows the best fuel saving ratio, and has one of the lowest relative fuel consumption factor at different levels of utility efficiency together with the best exergetic performance value.

A low heat-to-power ratio of a cogeneration plant necessitates its operation at high electric efficiency in order to provide fuel savings as utility efficiency grows. A conventional boiler-based CHP installation with a high heat-to-power ratio is much less sensitive to an improvement of utility efficiency. However, in terms of technology development only cogeneration plants with exergetic efficiency close to that of utility should be used. This calls for the same programme of repowering low-efficiency cogeneration plants, as was done in the 1970–1980s for direct-fired utility plants.

The heat-activated heat pump demonstrates a better performance than the reference scheme, consuming about 83% fuel of the reference, when the demand heat-to-power ratio is 0.5, while at $\lambda_D = 5$ the relative consumption increases to 95%. The electric heat pump system is directly dependent on utility efficiency and can outperform the other options only when the utility reaches an efficiency value of 0.6.

Supplementary firing at gas turbine-based plants was found not attractive in exergy terms: the unfired schemes prove to be better in all cases.

It was shown that the fuel utilization efficiency based on the First Law failed as a means for comparing different CHP configurations. The high FUE value of the conventional boiler or heat pump system reflects only the quantitative, and not the qualitative side of the process, and only the exergy criterion can demonstrate the imperfections of these configurations.

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Chapter 5
Externally-Fired Combined Cycle

Abstract

The most advanced systems for the generation of electricity from coal are the Integrated coal Gasification Combined Cycle (IGCC) and the Pressurized Fluidized Bed Combustion (PFBC) system. They both represent state-of-the-art technology and provide electric power at 41-46% LHV efficiency. However, the high level of complexity leads to lower reliability and higher equipment and maintenance costs. A simpler system, based on a coal-fired steam boiler integrated with a gas turbine, has been proposed. In such a plant, known as the externally fired combined cycle (EFCC), compressed air is heated indirectly in a heat exchanger located in the coal furnace. The hot air is then expanded in a turbine, and the exhaust is passed on to the coal furnace as preheated combustion air. Steam produced in the boiler generates additional power in a steam turbine. The exclusion of coal combustion products from the gas turbine avoids the expense of hot gas clean-up and corrosion of turbine blades by coal ash.

In order to obtain a high level of efficiency the air entering the gas turbine should have a temperature close to the turbine inlet temperature, which can be as high as 1300° C for modern turbines. Such temperatures can be accomplished by the use of ceramic heat exchangers, or alternatively, by supplementary firing of a premium fuel, such as natural gas, in the air flow.

To analyze the potential of an externally-fired gas turbine system with supplementary firing, a computer model was set up. Using realistic assumptions, the optimal parameters were found, and the influence of supplementary firing was calculated. It was shown that for an EFCC, supplementary firing provides a transition solution. To compete with other power plants in the future, a high-temperature ceramic heat exchanger would be required. Finally, a basic exergy analysis was carried out to assess the EFCC concept and to determine exergy losses within the systems under consideration. The overall exergy efficiency was found to be comparable with those of the IGCC or PFBC systems.

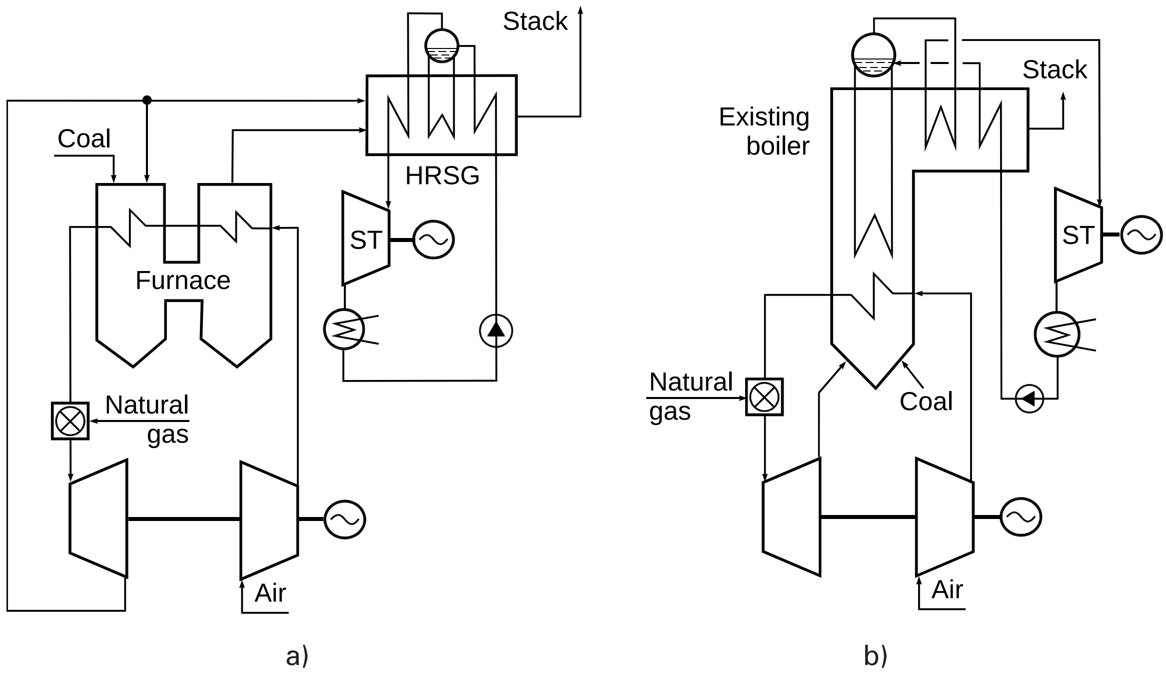

Fig. 5.1. Schematic diagram of an externally-fired combined cycle plant: (a) with a specially-designed air heater; (b) in repowering configuration.

5.1 Introduction

Coal represents about 70% of the world's proven fossil fuel resources and provides 31% of the world's electricity (Ramezan et al., 1996). Electric power from coal is predominantly generated in direct coal-fired power plants, where hot products of coal combustion exchange heat with steam in a boiler, and the steam produces work in a steam turbine. Due to thermodynamic and metallurgic constraints, the efficiency of such plants is rather low. Modern coal-fired power plants obtain an efficiency of 38-40% (LHV) operating at 250-300 bar and 530-560 °C. Higher efficiencies can be achieved by further increasing steam parameters, but even for supercritical boilers operating at pressures around 350 bar and temperatures of 700 °C efficiency is expected to be not higher than 47% (LHV). These figures contrast with the efficiency of natural gas fired combined cycle plants that at relatively moderate parameters convert 45-52% of the fuel's lower heating value into electric power, while today's more advanced plants can achieve even 57% efficiency (Farmer, 1996).

An intrinsic handicap of the direct-fired boiler is a large temperature difference between the combustion gases and the working medium, which is more than 1200 K for a typical plant. This work potential is better utilized in a gas turbine cycle that covers a temperature range from 1400 °C to 400 °C. Since coal as a solid fuel is not quite suitable for a gas turbine, its chemical energy can be delivered to the turbine in an indirect way. There are several technologies known today that provide an effective conversion of coal into electricity by employing the gas turbine cycle. These are the Integrated Coal Gasification Combined Cycle (IGCC), the Pressurized Fluidized Bed Combustion (PFBC), and the Externally-Fired Combined Cycle (EFCC) plants.

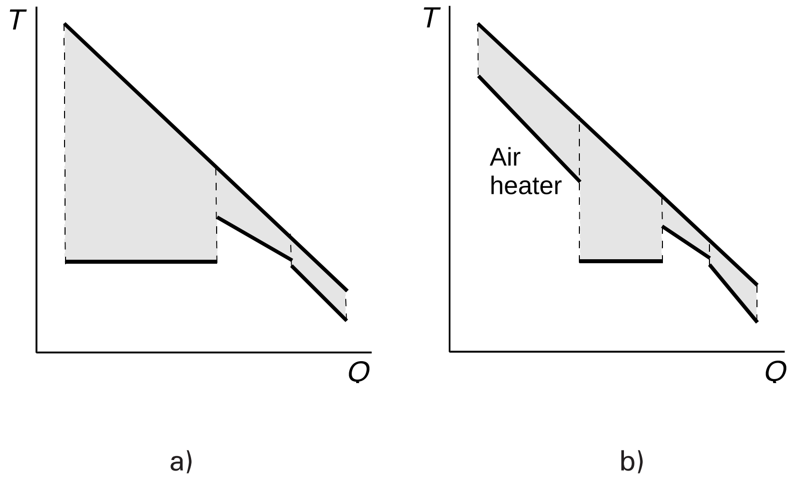

Fig. 5.2. Flue gas heat utilization in:
(a) a steam boiler; (b) an externally-fired cycle

In an IGCC plant coal is gasified, cleaned-up and fed to the gas turbine. Then, exhaust heat from the gas turbine is recovered in a steam boiler. Coal gasification proceeds in a fixed bed, fluidized bed, or in an entrained flow system. Modern coal gasification power plants show an LHV efficiency of 42 to 45% (Maude, 1993; Stambler, 1996). However, an IGCC plant contains a number of critical components for synthesis gas cooling and clean-up, and it is has a rather high level of complexity and capital costs.

Combustion of coal in a fluidized bed under pressure integrated with a gas turbine is another advanced coal conversion technology. The gas turbine is driven by compressed air which is heated in the fluidized bed. Nevertheless, most of the power is generated by the steam turbine, which accounts for up to 80% of the plant's total output. Today's PFBC plants operate with an efficiency of 37-38% (LHV), and the next generation plants are expected to reach a level of 40-45% (Almhem and Lofe, 1996). Just as the IGCC technology, pressurized bed combustion is also characterized by its complexity and high capital costs.

Externally-fired combined cycle makes use of a gas turbine as well, but avoids coal combustion gases entering the turbomachinery. Heat addition in the gas turbine cycle occurs indirectly in a high temperature heat exchanger, which is situated in the coal furnace (Fig. 5.1a). After expansion, air can be partly redirected to the furnace to supply preheated air for combustion, and/or passed to a heat recovery steam generator. This concept is also feasible for repowering of existing steam plants (Klara et al., 1996). In this case, instead of a specially-designed high temperature air furnace, air is heated in the existing furnace (Fig. 5.1b). However, an implementation of the repowering concept may require major reconstruction of the steam boiler.

A comparison with a Rankine cycle shows an advantage of the externally-fired cycle in heat utilization, as shown in Fig. 5.2. Although air has a lower specific heat than steam, it does not require high pressures at high temperatures. Heat exchange between air and the flue gases occurs in the gaseous state, so the average temperature difference in the heat exchangers is smaller than in the case of steam.

Table 5.1. Overview of EFCC plants in literature

Authors	Year	Heat exchanger				SF	Gas turbine			Recycle	Steam turbine				Plant efficiency	
		ε %	ΔT °C	Δp/p %	T air °C	T air °C	p ratio	η comp %	η turb %	%	p in bar	T in °C	p cond mbar	η turb %	η HHV %	η LHV %
LaHaye et al.	1989	80		4.5	1150	-	10	88	88	100					46	
Foster-Pegg[1]	1990		111		788	-	9.1	86/80	88	0	45	482		75	33	
Parson et al.[2]	1991	90		2	1260	-	13.5	89.1	91.3	100	41	482			48.3	
De Ruyck et al.[3]	1991		100		800	860	7.7	80	85	100						36.3
Huang et al.	1992				1260	-	16	90	90	100	30	482	70	85		53
Vandervort et al.[4]	1993	85		3	1260	-	30			100	60	471	34		39.8	
Huang et al.[5]	1994	90			1260	-	12	88	90	100						48.7
Seery et al.[6]	1995				927	1371	40			40					47	
Ruth[7]	1995				980	1285	12			40					48.5	
Klara et al.[8]	1995			4.5	1000	1288	15				180	579	50	84	47.5	
Consonni et al.	1996		100	4.5	1350	-	15	89.5	90.8	100	239	558	50			49
Mathieu et al.	1996				1300	-	14			100	140	540	50	85		52
Wiechers[9]	1997	90		5	980	1165	15	85.6	88	49	130	520	40			46.7

Notes:
1. atmospheric fluidized bed, 15% steam injection
2. in HAT-cycle with 16% water injection efficiency is 52.2%
3. HAT-cycle
4. LM6000 gas turbine, 1% steam injection
5. HAT-cycle with 20% water injection
6. FT4000 gas turbine, 35% natural gas
7. 35% natural gas
8. 35% natural gas
9. 24% natural gas

Heat exchanger: hot side ΔT and average pressure drop are given.
SF: a hyphen means that no supplementary firing is provided.
Recycle: fraction of gas turbine outlet flow sent to the furnace.

Air heating occurs in the most critical component of the system, the high temperature heat exchanger. With the exception for the heat exchanger, all other equipment is of proven technology. Depending on the turbine inlet temperature and other parameters, an EFCC plant can obtain an efficiency of 40-49% (LHV).

5.2 Background

An externally-fired gas turbine cycle was proposed long ago as a possible realization of the Joule-Brayton cycle, but much interest has been regained in the last decade. In the United States this concept has been developed by a group of companies, including Hague International, the developer of the high-temperature ceramic heat exchanger. The group is headed by the Morgantown Energy Technology Center (LaHaye and Zabolotny, 1989; Parsons and Bechtel, 1991; Vandervort et al., 1993).

Another group, under the management of the Pittsburgh Energy Technology Center, is working with industries to develop a similar power cycle called the High Performance Power System (HIPPS). The air heater is being designed by the Foster Wheeler and United Technologies teams (Ruth, 1995; Seery et al., 1995).

In Europe, work on EFCC is being carried out in Italy by a consortium of Ansaldo, ENEL, and research institutions, with funding from the European Union (Consonni et al., 1996a). The Swedish National Board for Industrial and Technical Development NUTEK is promoting an externally-fired evaporative gas turbine cycle, in particular, using biomass fuel (Eidensten et al., 1994; Yan et al., 1996). The Netherlands Agency for Energy and Environment Novem has included EFCC in its New Energy Conversion Technologies program for technical and economic evaluation (Korobitsyn, 1996).

In literature, the cycle is described in a number of publications, a summary of which is given in Table 5.1. The authors, using comparable assumptions, came out with similar performance figures. For example, cycles with turbine inlet temperatures of 1200-1300 °C obtained an LHV efficiency of 48-52%. The advantages of air humidification were outlined in the works by Parsons and Bechtel (1991), De Ruyck et al. (1991), Huang and Naumowicz (1994), etc.

Huang and Naumowicz (1992) concluded that a high efficiency of the cycle is determined by a high turbine inlet temperature, while the cycle pressure ratio does not considerably influence the efficiency. A pressure ratio of 12-15 was chosen in the most studies. As for the turbine inlet temperature, it is limited by the heat exchanger materials. Nickel-based superalloys allow operation at 800-825 °C. More advanced oxide dispersion alloys withstand temperatures around 950 °C (Klara, 1995). A ceramic heat exchanger made from silicon carbide composites is capable to operate at even higher levels. Temperatures up to 1075 °C were achieved in the experiments reported by Solomon et al. (1996), and the level of 1370 °C and higher is anticipated (Vandervort, 1993; Seery et al., 1995). To improve the corrosion resistance of silicon carbide ceramic oxide coatings can be used (Van Roode, 1993).

However, development of the ceramic heat exchanger still requires some engineering effort: such problems as relatively large dimensions of the exchanger and the ducting, thermal expansion of the ceramic components at start-up and stop conditions, connections and seals, slag behaviour on ceramic surface and others, should be taken into account. The current price per square meter of ceramic surface, which is 2.5 or several times higher than that of a metallic heat exchanger (Consonni and Macchi, 1996), may be prohibitive. The use of supplementary firing in order to raise the air temperature to the gas turbine inlet conditions can be considered as an interim solution. This possibility is discussed by Ruth (1995), Klara et al. (1996), and others, and elaborated in this chapter.

5.3 Plant configurations

At a given turbine inlet temperature, the amount of fuel consumed in the supplementary burner is determined by the temperature level in the air heater. A number of configurations were defined to estimate the performance of EFCC for different types of the air heater: (1) a metallic one made from nickel-based alloys operating at 800°C; (2) a metallic one made from oxide dispersion alloys operating at 980°C; and (3) a ceramic (silicium carbide) heater at 1165°C, a part of which can be made of metal.

They were compared with other power cycles. The configurations were denoted as:

PC	pulverized coal fired power plant
EFCC-800	EFCC, air heater temperature of 800°C, supplementary firing to 1165°C
EFCC-980	air heater temperature of 980°C, supplementary firing to 1165°C
EFCC-1165	air heater temperature of 1165°C, no firing
GTCC	natural gas fired combined cycle plant
IGCC	coal gasification combined cycle plant

The options with a gas turbine were based on the specifications derived from those of a Siemens V94.2 engine:

Compressor isentropic efficiency	88.0%
Turbine isentropic efficiency	85.6%
Pressure ratio	13.7
Turbine inlet temperature	1165 °C
Combustion chamber pressure drop	0.27 bar
Combustor efficiency	99%

The steam section of all configurations consisted of a single-pressure boiler that provided superheated steam of 550 °C and 140 bar to a condensing steam turbine. The turbine had an isentropic efficiency of 88%, and its condensing pressure was set at 0.05 bar.

The heat exchanger featured 90% effectiveness and a pressure drop of 5%. Supplementary firing was provided by burning methane (the lower heating value of 50

Table 5.2. Summary of results

	PC	EF-800	EF-980	EF-1165	GTCC	IGCC
ST fraction, %	100.0	38.2	39.3	39.9	33.2	49.8
GT fraction, %	0.0	61.8	60.7	60.1	66.8	50.2
Coal fraction, %	100.0	51.9	75.4	100.0	0.0	100.0
Methane fraction, %	0.0	48.1	24.6	0.0	100.0	0.0
LHV efficiency, %	34.84	47.75	46.70	45.58	50.15	45.82
Surface, m^2/kW	223	315	357	425	252	
- boiler	223	270	276	281	252	
- air heater	0	45	80	143	0	

MJ/kg). Part of the gas turbine outlet flow that is passed to the coal furnace was controlled so that the flue gas temperature at the steam boiler inlet remained at 630 °C. Depending on the coal fraction in the total fuel input, a weighted 2.5% penalty was subtracted from the gross plant efficiency due to the flue gas desulphurization (FGD).

For the coal gasification plant, the Shell technology based on an entrained flow gasifier was chosen. The gasifier operated at 1500 °C and 30 bar, and the pressure losses across the gasifier of 5 bar were assumed.

5.4 Discussion

Simulation results show a considerable performance gain of EFCC plants over the pulverized coal plant, meaning about 30% lower fuel consumption (Table 5.2). At the same time, external firing options cannot reach the level of a natural gas fired combined cycle due to a higher internal consumption rate for coal preparation, the flue gas

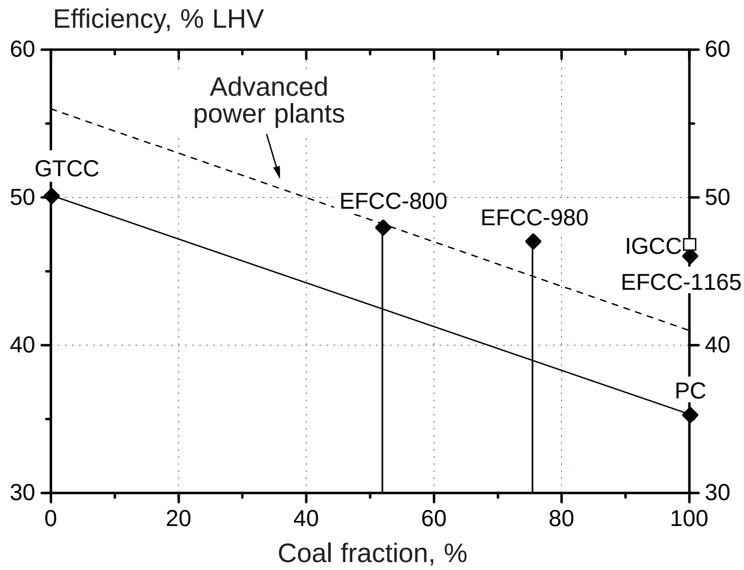

Fig. 5.3. Power plants efficiency versus coal fraction in the total fuel input.

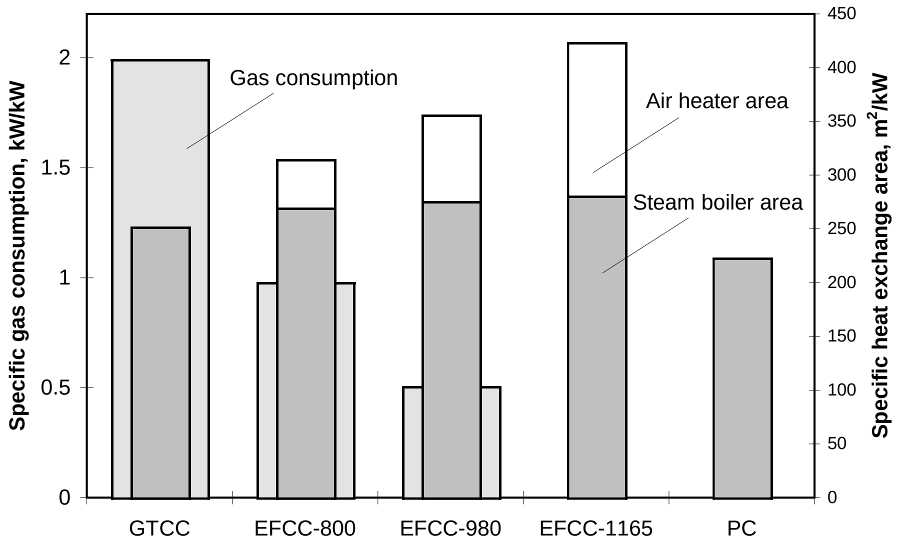

Fig. 5.4. Specific methane consumption and specific surface area of the steam boiler and the air heater per 1 kW of the net power output.

desulphurization, losses in the air heater, etc. Nevertheless, EFCC efficiencies were found comparable with that of the IGCC plant.

Bearing in mind that a EFCC plant's ultimate goal is to use 100% coal, schemes can be evaluated whether the use of natural gas in the EFCC is justified in comparison with two separate plants, a natural gas-fired combined cycle and a direct-fired plant. Such judgement can be done by using a mean efficiency relative to a coal fraction in the total fuel input. As shown in Fig. 5.3, externally-fired options have an advantage above the mean efficiency line between PC and GTCC. It ranges from 5.6% points in case EFCC-800 to 10.7% points in case EFCC-1165. When compared with advanced plants, depicted by a dashed line on the chart, the advantage becomes much smaller, and in case EFCC-800 even goes marginal. This means that the effect of supplementary firing in EFCC plants is rather limited in a long term, and only the development of a high-temperature heat exchanger for a 'coal-only' EFCC can provide its competitiveness with respect to other power cycles, such as IGCC.

As more coal is used in the externally-fired cycle, a small drop in efficiency can be noted. This is caused by the reduction of the natural gas consumption, and therefore, a lower flow across the expander. Another reason is a higher FGD penalty levied on efficiency by the increased coal consumption.

Regarding the surface of heat exchangers, the boiler surface area was found of comparable size for all plants, while the area of the air heater varies considerably (Fig. 5.4). In the 'coal-only' EFCC, it constitutes 34% of the total surface versus 14% in the case with maximum supplementary firing (EFCC-800). On the same chart, a specific natural gas consumption in kW(thermal) per kW(electric) is plotted. It allows a comparison between the gas consumption and the surface area. For example, 0.5 kW of natural gas consumed by EFCC-980 is equivalent to 63 square meters of ceramic surface in case EFCC-1165. An economic analysis should evalu-

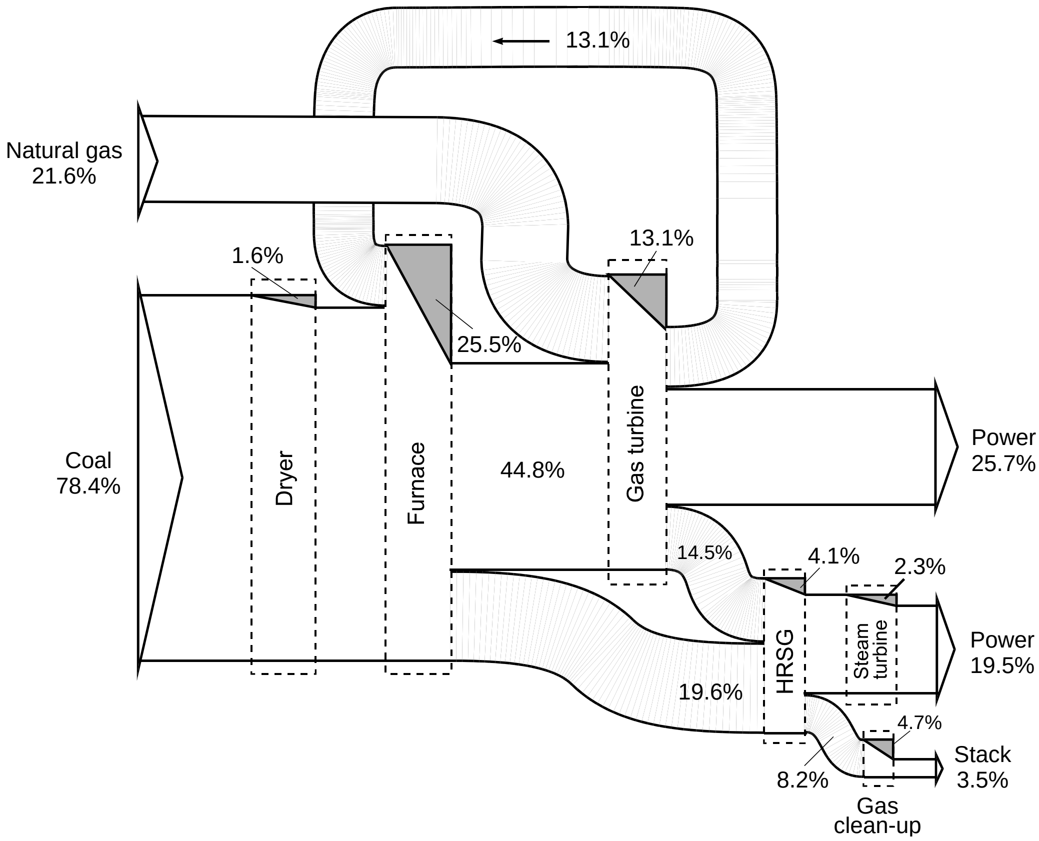

Fig. 5.5. Grassmann diagram of an EFCC power plant.

ate an economic optimum, as soon as the price of alloys and ceramic surfaces will be available.

To maintain the boiler inlet temperature at 630 °C in all EFCC configurations, the fraction of the turbine outlet flow sent to the furnace was set to 0.335, 0.494, and 0.66 for EFCC-800, EFCC-960, and EFCC-1165, respectively. The difference is due to a different air heater's duty. An air heater with a larger duty requires more preheated air coming from the gas turbine outlet.

5.5 Exergy analysis

To assess systems exergetically, case EFCC-980 was compared with an IGCC plant, assuming that they both are based on commercially available hardware. The results of the analysis are presented in the Grassmann diagrams (Figures 5.5 and 5.6). The analysis showed an equivalent efficiency of the plants, but a different distribution of exergy losses. As expected, the largest losses were found in the combustion process. The total losses in the coal furnace and the gas turbine for case EFCC-980 amount to 38.6%. This figure agrees with those reported by Chen (1997) and Eidensten and Yan (1997) for similar systems, 35.3% and 41.9%, respectively. This value is also comparable with the losses in the gasifier and the gas turbine of the IGCC plant that make up 36.5%. They are the greatest losses within the systems, and can be diminished by

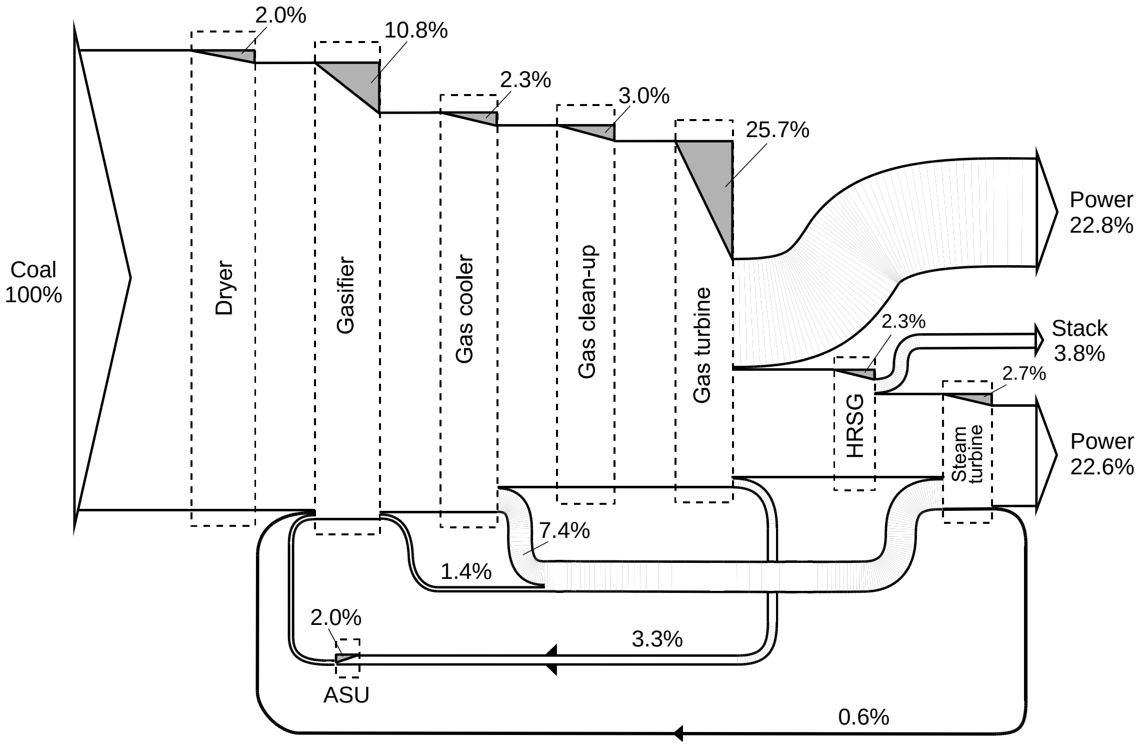

Fig. 5.6 Grassmann diagram of an IGCC power plant.

choosing the most efficient equipment for these key components. Other measures, such as recuperation or exhaust gas recycling, may also be considered.

Exergetic efficiency of heat recovery can be improved by choosing more pressure levels in the HRSG. A high-temperature clean-up system will reduce losses related to removal of aggressive compounds and particles.

5.6 Conclusions

Externally-fired combined cycle plant, except for the air heater, based on currently available hardware and can operate at efficiencies similar to those of advanced coal plants such as IGCC.

Supplementary firing in the externally-fired combined cycle power plants provides a short-term solution, as materials for the high temperature heat exchanger are being developed. The efficiency of a plant with supplementary firing is remarkably higher than that of a pulverized coal plant. In a long term, however, the use of natural gas for firing would not be justified, since it can be better utilized in high efficient gas-fired combined-cycle plants.

At the same time, supplementary firing allows a flexible control over the power output, which is rather limited in a ‘coal-only’ option.

The ‘coal-only’ plant necessitates the use of a costly ceramic-based heat exchanger, which surface area can be half of that of the steam boiler. Possible problems caused by the use of ceramics include limitation on the maximum pressure, manufacturing of ceramic connections and seals, slag deposition on the ceramic surface.

An economic analysis is needed to find an optimum between an expensive heat exchanger surface and the consumption of a premium fuel for additional firing.

In exergetic terms, EFCCs were proved to be comparable with IGCC plants. Using the results of exergy analysis, some improvements can be considered in order to reduce irreversibilities within a system.

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Chapter 6
Integration of a Gas Turbine and a Municipal Solid Waste Incinerator

Abstract

Modern municipal waste incinerators not only considerably reduce waste volume, but also produce an appreciable amount of heat which can be recovered in a steam boiler. The aggressive nature of the flue gases does not allow the temperature of steam in the boiler to rise above 400 °C. An increase in steam temperature can be achieved by external superheating in a heat recovery steam generator positioned behind a gas turbine, so that steam of a higher energy content becomes available for electricity production, or process needs. This can be accomplished in a number of ways. In one case, steam generated at a waste-to-energy plant is superheated in a combined-cycle plant that operates in parallel. In the other case, the exhaust from a gas turbine plant is sent through a superheater section to the waste incinerator's boiler providing preheated combustion air. Performance of these configurations together with two modified schemes was analyzed in terms of efficiency, natural gas consumption and boiler surface area. An exergy analysis of the cases was carried out. The results showed that the integrated options can effect a substantial increase in efficiency. The hot windbox configuration was found the most effective solution, offering a smaller boiler surface area along with a moderate rate of natural gas consumption.

6.1 Introduction

Of waste processing technologies such as landfill, composting, or recycling, incineration remains the most effective volume reducing technology. It also can manage disposal of various wastes, such as combustible solids, semi-solids, sludge, liquid wastes, and gases. The postcombustion systems of an incineration plant control undesirable airborne emissions. In addition, the heat released from combustion can be recovered in a boiler to supply steam to a steam turbine, or for process needs. In this way Switzerland, Luxembourg and Denmark recover energy from over 70% of their waste

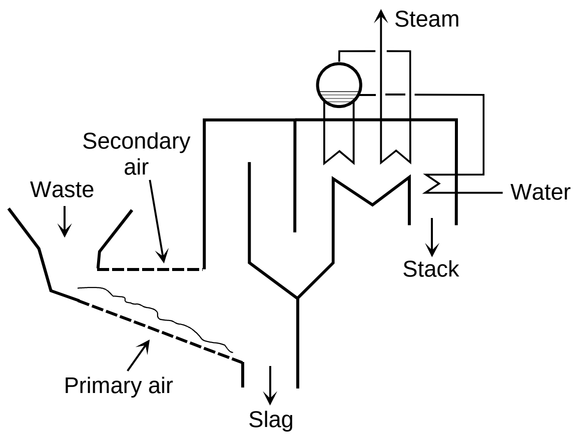

Fig. 6.1. Simplified solid waste incinerator schematic (Reference case).

(Wolpert, 1994). An incineration plant can process as much as 2500 tons of refuse per day (Tillman et al., 1989). Heat recovery and power generation have become a common practice at incineration plants, and many waste-to-energy (WTE) plants have been built around the world (Isles, 1993; Pirson and Braker, 1994; Ramsgaard-Nielsen, 1994; Anonymous, 1996). A general overview of the new thermal treatment technologies can be found in Pfeiffer and Van Egmond (1995).

The composition of the municipal solid waste (MSW) varies considerably and depends on many factors, such as location, local policy, origin of the waste, etc. Typical waste consists of: paper and paperboard 20-40% by mass, yard and food wastes 20-35%, wood 2-6%, plastic 6-10%, textile 2-6%, glass 3-8%, metals 3-10%, and inert matter up to 10% (Tillman et al., 1989, Pirson and Braker, 1994). The heating value of the waste varies accordingly from 6 to 14 MJ/kg. The magnitude of the heating value affects the ratio between primary and secondary air in the incinerator. For waste with a low heating value, the primary air constitutes 80% of the total air flow, while for waste of high and medium heating value, the percentage lies between 50% and 70%.

In order to avoid slag deposits, erosion and corrosion in the boiler, and to maintain a sufficient residence time for after-burning, several arrangements in the boiler design should be provided. The evaporator's tubing is to be located in the second pass, the superheater's tube bundles in the second or the third pass, where flue gas temperature drops to an acceptable level. The surface temperature of the tubes can be adjusted with the use of a de-superheater.

The flue gas, passing through the boiler, is cooled from 1000 °C in the radiation section to 600°-800 °C at the entrance of the convection area. After passing the evaporator's bundles the gas flows through the superheater and economizer (Fig. 6.1). The aforementioned precautions result in a relatively low temperature of the superheated steam of around 400 °C. On the other hand, temperature of the flue gas in the stack should not drop below 200 °C due to the risk of condensation of aggressive compounds. These limits constrain the high efficiency of the MSW incineration boiler. Some improvement of the cycle can be achieved by raising the temperature of the steam externally or by preheating the combustion air, or both.

Table. 6.1. Composition of the municipal solid waste.

Mass%		**Molar%**	
Yard and food waste	30.0	Carbon	31.24
Paper and paperboard	29.0	Hydrogen	4.28
Plastic	12.0	Oxygen	17.83
Glass	3.0	Nitrogen	0.96
Metal	3.0	Sulfur	0.17
Textile	6.0	Phosphorus	0.10
Combustibles	12.0	Chlorine	0.95
Non-combustibles	5.0	Water	26.90
		Ash	17.57

6.2 Gas turbine integration

The high temperature of a gas turbine's exhaust makes it well-suited to integration with an MSW incinerator. Exhaust heat can be recovered in the waste heat boiler, where further superheating of steam coming from the incinerator takes place. Utilization of gas turbine exhaust heat as preheated combustion air is another possible integration scheme.

Plant configurations, where the heat recovery steam generator (HRSG) operates in parallel with the incineration boiler, are described in the literature: the lower temperature steam from the incinerator can be passed either to the intermediate pressure section of a steam turbine (Ramsgaard-Nielsen, 1994), or be further superheated in the HRSG (Andersson, 1996; Van Wijk, 1994). In the latter configuration, the integrated plant was reported to have a 2.2% higher efficiency than the mean efficiency of two separate plants. However, in the both integrated configurations, the gas turbine plays a dominant role, consuming up to 80% of the fuel input.

Another approach is to use of the gas turbine exhaust as the inlet flow for the incinerator's furnace. This concept is often used for steam plant repowering. An improvement in the total plant efficiency from 41.3% to 45.9% together with a 28% power increase was reported by Pijpker and Keppel (1986).

The gas turbine exhaust contains 14-16 vol% oxygen, compared to 21 vol% in the air. Thus, in order to provide the same amount of oxygen to the boiler, a 30% larger flow from the gas turbine is required. At the same time, exhaust heat reduces heat demand in the boiler, and less oxygen is needed. Davidse and Roukema (1984) estimated that the gas turbine flow should be about 106% of the normal air flow for the same heat load in the boiler's furnace. Considering the combustion of waste, the exhaust flow should have a pressure high enough to pass air through the waste layer on the grates. This can be accomplished either by the use of air blowers or by expansion to a pressure above the atmospheric level.

Such a hot windbox configuration is dependent on the gas turbine performance. A gas turbine trip can affect the steam boiler reliability, therefore, some precautions should be made. For example, air fans can be kept in a stand-by position, or operated in parallel with the gas turbine. When the gas turbine flow is too large for a given boiler,

Table. 6.2. Plant specifications.

Waste incineration boiler:		**HRSG:**	
HP steam	40 bar, 400 °C	Approach temperature	15 °C
LP steam	3.7 bar, saturated	Pinch temperature	25 °C
Condenser pressure	0.1 bar	Heat transfer coefficients:	
Deaerator pressure	3.5 bar	· economizer	65 W/m^{2}K
Minimum stack temperature	200 °C	· evaporator	55 W/m^{2}K
Combustion efficiency	0.96	· superheater	35 W/m^{2}K
ST isentropic efficiency	0.85	Superheater temperature	520 °C
		Minimum stack temperature	80 °C
Gas turbine:			
Net power output	38 MW		
Efficiency (LHV)	0.319		
Exhaust temperature	545 °C		
Exhaust flow	139.4 kg/s		

part of the flow can be bypassed to a stack, or to the convection section of the boiler.

The performance of the hot windbox scheme can be further improved, if external superheating is applied. If the steam from the incineration boiler is passed to a superheater located in the gas turbine exhaust duct, where it is not exposed to the corrosive gases, steam temperatures higher than 400 °C can safely be achieved. The superheater has a simpler design and a much smaller surface area than a heat recovery boiler, while increasing the steam temperature to the same level as that in the HRSG.

6.3 Configurations

To estimate the effect of gas turbine integration in different configurations, integrated plants were compared with a conventional MSW incinerator. The amount of waste being processed by all plants was assumed to be 230 000 tons of solid waste per year, or 33 tons per hour. This amount of refuse is equal to a fuel input of 92 MWth, if the lower heating value of the waste averages 10 MJ/kg. The composition of the waste is presented in Table 6.1. The specifications of the reference incineration plant, the gas turbine plant based on a General Electric Frame 6 turbine, and the heat recovery steam generator are given in Table 6.2. The simulation models based on the gas turbine were scaled to have the same fuel (waste) input for comparison analysis.

Case 1 represents a combination of a gas turbine plant with a heat recovery steam generator and a waste incinerator. The steam coming from the incineration plant is passed to the HRSG, where it is superheated to 520 °C, and then sent to the steam turbine (Fig. 6.2).

Since the exhaust from the HRSG has a temperature of 80-100 °C, this heat can be utilized in the incineration boiler as secondary air. This modification of the previous case is referred to as Case 2. In Case 3, the gas turbine exhaust heat is partially recovered in a superheater, following which it is passed to the MSW incinerator (Fig. 6.3).

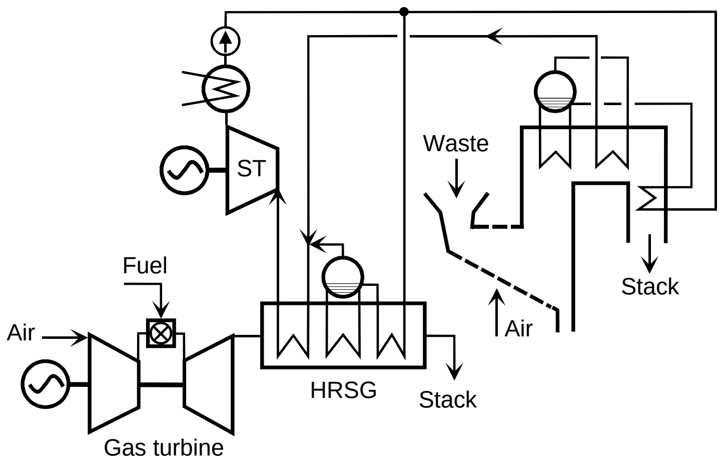

Fig. 6.2. HRSG-Incineration boiler parallel configuration (Case 1).

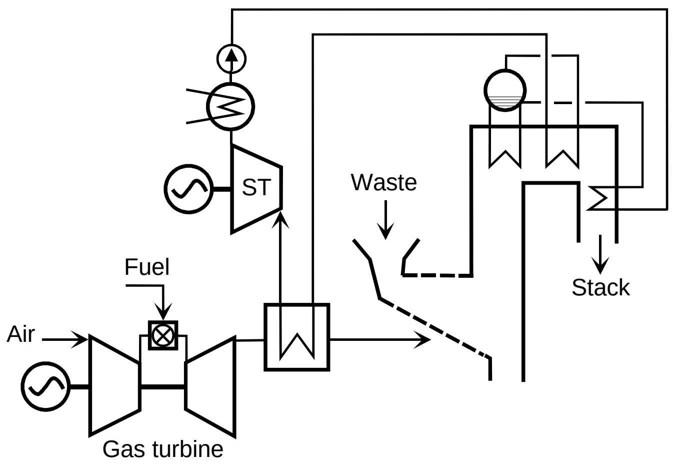

Fig. 6.3. The hot windbox configuration with superheating (Case 3).

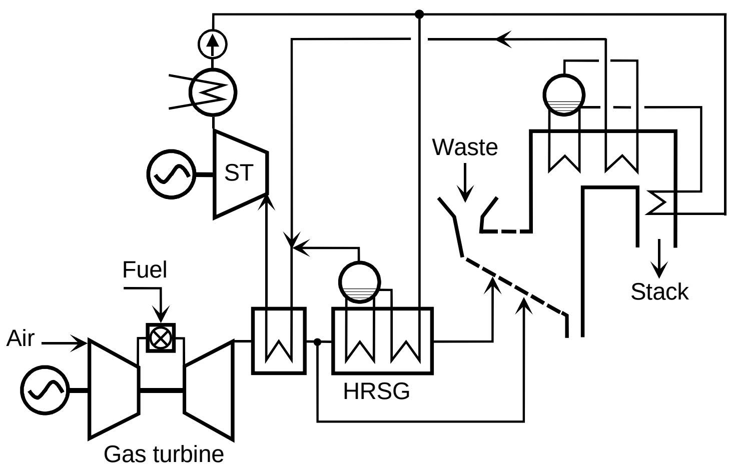

Fig. 6.4. HRSG-Incineration boiler parallel configuration with an exhaust bypass (Case 4).

Table 6.3. Summary of results (scaled).

	Reference	Case 1	Case 2	Case 3	Case 4
Fuel input, MWth					
· incineration boiler	92.00	92.00	92.00	92.00	92.00
· gas turbine	0.00	100.14	101.39	48.44	104.73
Total	92.00	192.14	193.39	140.44	196.73
MSW share, %	100.00	47.88	47.57	65.51	46.76
Natural gas share, %	0.00	52.12	52.43	34.49	53.24
Power output, MWe					
· steam turbine	22.92	46.46	46.92	36.89	47.79
· gas turbine	0.00	31.96	32.37	15.48	33.44
Total	22.92	78.42	79.29	52.36	81.23
Efficiency, %					
· based on total input	24.91	40.81	41.00	37.28	41.29
· based on MSW	24.91	28.64	28.88	29.54	29.10
Specific surface area,					
m^2/MWe	294	710	686	340	518

In this configuration a large steam flow from the incineration boiler does not allow the steam temperature in the superheater to rise above 486 °C. Thus, another scheme with an HRSG and an exhaust bypass is considered. The schematic of this case is given in Fig. 6.4. Here, a superheat of 520 °C can be achieved by controlling the amount of the flue gas that is directed to the furnace via the bypass.

In all integrated configurations the steam pressure in the incineration boiler was raised from 40 bar to 80-100 bar in order to take full advantage of superheating.

The cases were simulated using GateCycle heat balance software package (Enter Software, 1995). The simulations were made under ISO conditions (15 °C, relative humidity 60%, sea level). Pressure losses in heat exchangers and through the waste combustion grates, blow-down, and deaerator vent flows were not taken into account. Internal consumption, which constitutes 12-15% of the total power generated, was omitted as well. Thus, only gross efficiency is given in the analysis.

6.4 Performance analysis

The results of the analysis showed that integration of an MSW incinerator with a gas turbine plant leads to a significant increase in efficiency. The gross plant efficiency (based on LHV) improved from 24.9% in the reference case up to 37.3% in Case 3 and 41.3% in Case 4. As shown in Table 6.3 this figure is due to the gas turbine contribution, which consumes to 53.2% of the total fuel input. The use of the HRSG exhaust as the secondary combustion air in Case 2 had a little effect on the total efficiency due to its rather low temperature: the difference between Case 1 and Case 2 is

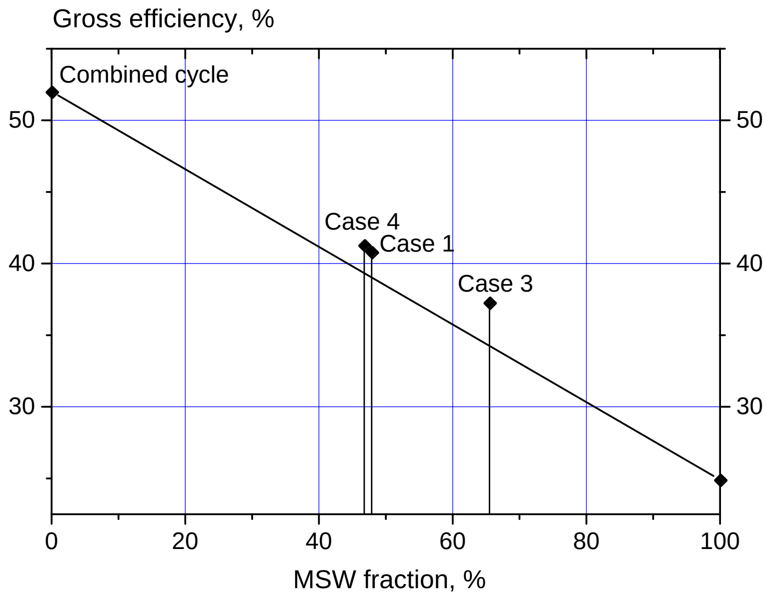

Fig. 6.5. The MSW fraction in the total fuel input versus gross efficiency.

only 0.2% points. Case 4, where an exhaust bypass is employed in order to rise the superheated steam temperature, has the best overall efficiency of 41.3%, however at the expense of the highest natural gas consumption rate.

Since the prime objective of a waste incineration plant is to process waste, the natural gas share should be limited, while maintaining a high efficiency. On this basis Case 3 represents a good match between the MSW fraction in the total fuel input and the plant efficiency. The waste share in this case is 65%, nonetheless, the plant efficiency is as high as 37.3%.

Excluding the gas turbine part, waste-based efficiency can be defined as the ratio between the power generated by the MSW combustion and the MSW fuel input. To determine the first term the gas turbine share should be subtracted from the total power production. This share can be found by multiplying the gas turbine fuel input by a combined-cycle efficiency, so that the expression for the MSW-based efficiency becomes:

$$\eta_{MSW} = \frac{P_{TOTAL} - F_{NG} \cdot \eta_{CC}}{F_{MSW}} \tag{6.1}$$

When 52% is taken as the combined-cycle efficiency, the calculated MSW-based efficiency varies between 28.6% for Case 1 and 29.5% for Case 3. In the best case, Case 3, this means an increase of 4.6% points in comparison to the reference.

The effect of integration can be illustrated, if the gross efficiencies of the cases are plotted against the MSW fraction in the total fuel input together with the mean efficiency of separate plants (Fig. 6.5). The latter represents a line between two extremes: a combined cycle plant that operates on 100% natural gas and has an efficiency of 52%, and a waste incineration plant (the reference case) that processes 100% waste at

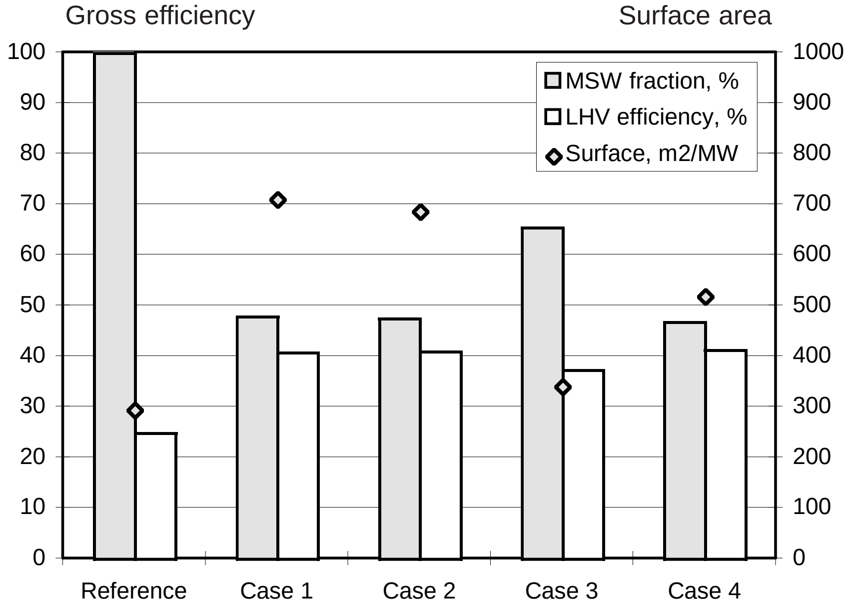

Fig. 6.6. Comparison of the MSW fraction in the total fuel input, the gross plant efficiency (LHV), and the specific boiler surface area.

24.9% efficiency. The benefits of integration are especially noticeable in Case 3: while Case 1 and Case 4 stand out above the line by 2.2 and 2.3 percentage points respectively, Case 3 shows a 3.6% advantage. Case 2 is not shown due to a minor distinction from Case 1. The values of the integration advantage are, of course, different, at other levels of the combined-cycle plant efficiency.

Considering the total boiler surface area, all integrated schemes require a larger surface than the conventional incineration plant. In Table 6.3 the surface area is expressed in specific units, m^2/MW. As is seen from the table, the schemes, where the incineration plant operates in parallel with an HRSG, require twice as much area as the reference plant. In Case 3, which has only a superheater, the specific surface area is just 15% larger than that of the reference. The hybrid scheme, Case 4, exceeds the reference by 76%.

A comparison of the MSW fraction in the fuel input, the specific boiler area, and the total plant efficiency is illustrated in Fig. 6.6. The chart indicates that the high efficiency of Cases 1 through 4 are obtained at the expense of the gas turbine plant. This results in a lower MSW fraction in the fuel input and a larger boiler surface area. Case 3 appears to be the most advantageous among the options considered.

6.5 Exergy analysis

In addition to the conventional enthalpy-based analysis, exergy analysis was performed in order to assess the systems thermodynamically. The flow parameters (mass flow, temperature, pressure, enthalpy, chemical composition) from the GateCycle models were post-processed to obtain exergy values of the flows.

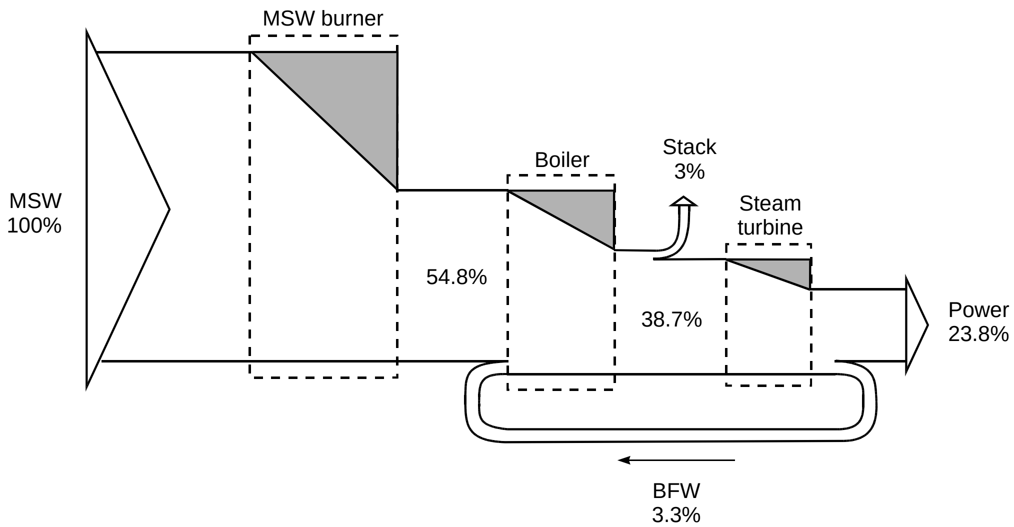

Fig. 6.7. Grassmann diagram of the conventional waste incinerator (Reference case).

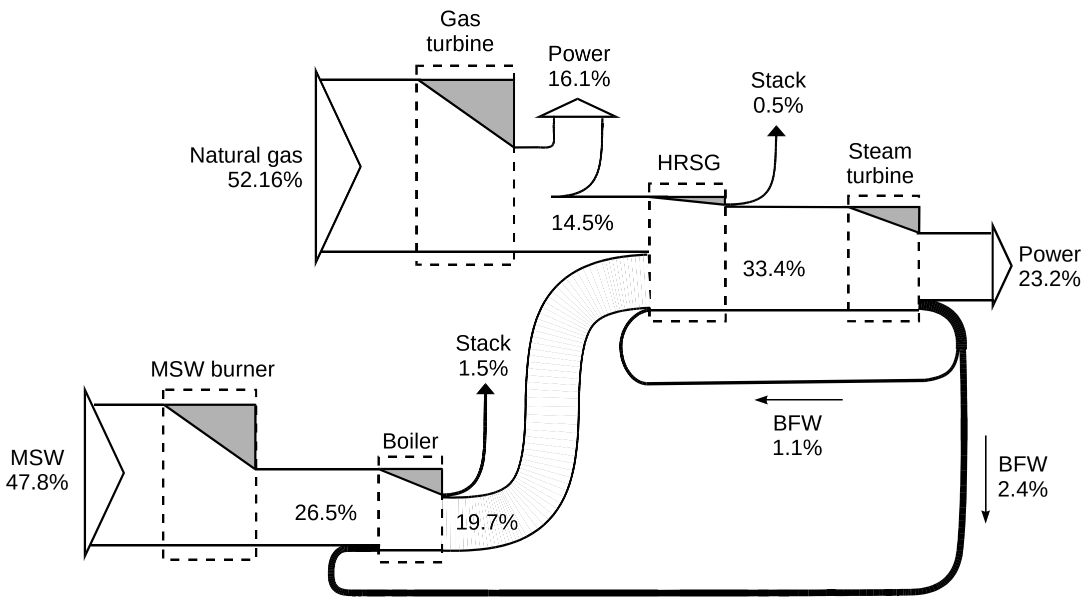

Fig. 6.8. Grassmann diagram of the parallel configuration (Case 1).

The results of exergy analysis are presented in Fig. 6.7-6.9 as Grassmann diagrams. The exergy values were calculated as a percentage of the total fuel input. The analysis showed that the largest loss of 45% occurs in the burner because of the highly irreversible nature of the combustion process (Fig. 6.7). The loss in the burner can be diminished to some extent by preheating the combustion air and by recycling the flue gas (Harvey et al., 1995). The use of the gas turbine exhaust as inlet air provides both measures. As seen in Fig. 6.9, this leads to an improvement of burner exergetic efficiency from a value of 54.8% to 64.2%.

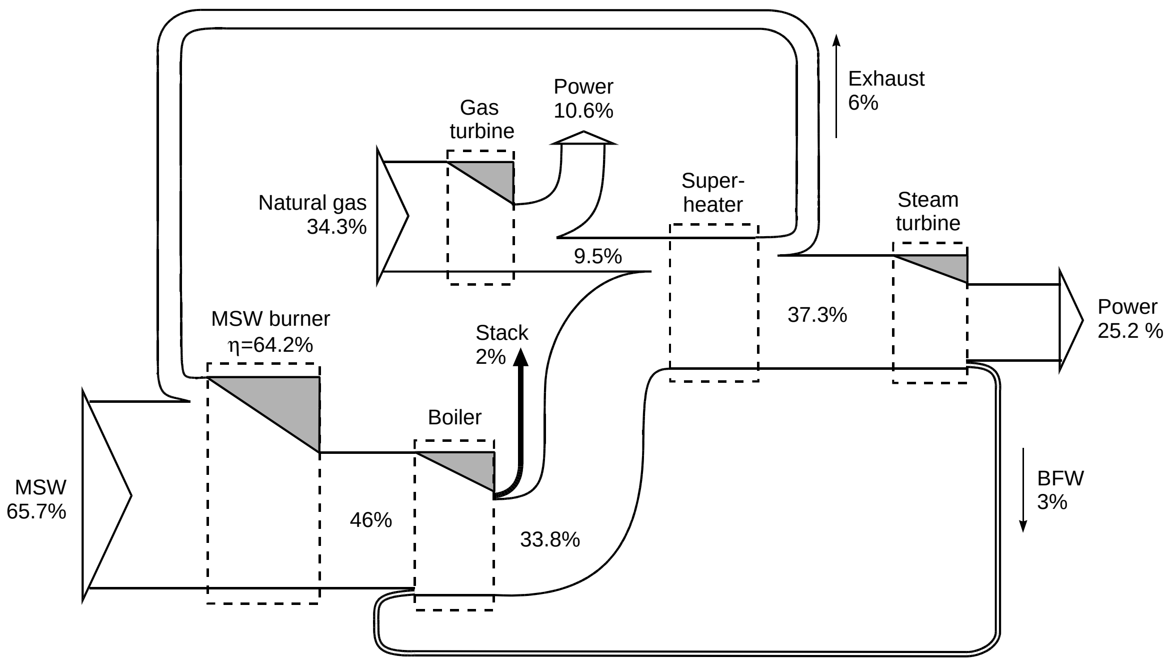

Fig. 6.9. Grassmann diagram of the hot windbox configuration with external superheating (Case 3).

Another major loss was found in the boiler, owing to a large temperature difference between the combustion gases and the working medium. In the reference case about 13% of the total fuel exergy is destroyed in the boiler. This loss cannot be avoided within the incineration boiler, so external superheating should be applied in order to increase the exergy of the steam. If steam is superheated in an HRSG, such as in Case 1, the exergy efficiency of the plant grows from 23.8% to 39.3% (Fig. 6.8).

6.6 Conclusions

Integration of a gas turbine and an MSW incinerator results in a considerable rise in efficiency. For the configurations studied, integration added 12-15% to gross plant efficiency. This is especially noticeable in the schemes where the incineration boiler operates in parallel with the HRSG. This improvement is caused by a more effective gas turbine plant; however, a significant amount of natural gas is needed.

When compared with the mean efficiency of separate plants with respect to the MSW fraction in the total fuel input, integrated schemes showed an advantage in efficiency ranging between 2.2 and 3.6 percentage points.

Among the options studied, the hot windbox configuration with external superheating offers high efficiency along with lower natural gas consumption. In addition, this scheme has half the boiler surface area than that of other integrated options, which feature a full-scale HRSG together with an incineration boiler. However, this option requires a ducting, and an excessive gas turbine outlet pressure should be provided to pass the exhaust gas through the waste layer on the grates.

The use of the HRSG outlet gases as the secondary combustion air in the incinerator had a negligible effect on efficiency. The scheme with partial exhaust diverting was found quite effective, though its waste fraction in the total fuel input was the lowest of all the cases.

The integrated options made use of heat recovery steam generators that lead to an increased boiler surface area. In the parallel configurations the area can be twice as large as that of the reference. For the windbox option the increase is only 15%.

Exergy analysis showed that the substantial exergy loss in the burner can be reduced by 9.4% points, if the gas turbine exhaust is applied as pre-heated combustion air. The second largest exergy loss that occurs in the incinerator boiler due to the large temperature difference is regarded as unavoidable.

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Chapter 7
Air Bottoming Cycle

Abstract

The heat recovery steam generator and the steam turbine of the conventional combined cycle plant can be replaced by the air bottoming cycle (ABC). In the ABC, the exhaust of an existing, topping gas turbine is sent to a gas-air heat exchanger, which heats the air in the secondary gas turbine cycle. These two turbines make up the dual gas turbine combined cycle (DGTCC) plant, which is characterized by the absence of water and steam equipment, low working pressure and a short start-up time. The plant allows unmanned operation, and can be implemented in regions where water resources are limited. In addition, air which leaves the secondary cycle at a temperature of 200 to 270 °C can be used for process needs. That makes it possible to implement the DGTCC as a cogeneration plant.

To analyze the performance of the dual gas turbine cycle, a number of simulation models, based on existing gas turbines, were made. Energy and exergy analyses of the DGTCC with various topping gas turbines were performed. The analyses showed that performance of the DGTCC is comparable with that of the conventional combined cycle plant with steam bottoming.

7.1 Introduction

The concept of an air turbine with an external heat source is not new: it dates back to the invention of the gas turbine. Nevertheless, interest in the idea has grown in the last decade: the air turbine is a key component in the externally-fired combined cycle (Chapter 7, this thesis) and in the air bottoming cycle. The latter was discussed by Farrell (1988) and by Wicks (1991). One of the implementations is the use of air bottoming with the topping gas turbine cycle (Anonymous, 1991). The exhaust gases from the gas turbine are recuperated in a heat exchanger by heating air in the secondary, air turbine cycle. The heated air expands in the turbine, supplying additional

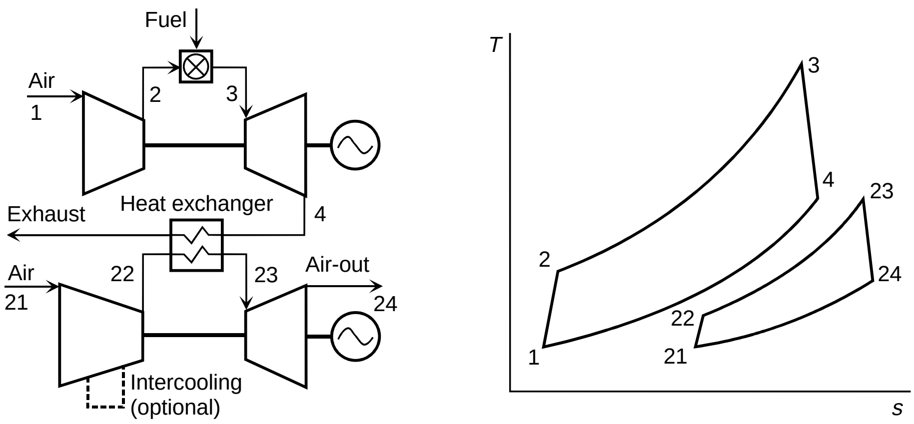

Fig. 7.1. Gas turbine with the air bottoming cycle.

power (Fig. 7.1). Such a configuration is known as the dual gas turbine combined cycle (DGTCC) (Wicks and Wagner, 1993; Weston, 1993).

In a conventional combined cycle plant the exhaust heat of the gas turbine is utilized by the heat recovery steam generator and steam turbine. In addition to the boiler and the steam turbine, the steam bottoming cycle (SBC) incorporates a condenser, a water treatment unit, pumps, piping, etc., resulting in considerable costs. At the same time, an advanced unfired SBC provides only a third of the total plant power. It also has some drawbacks in operation and maintenance: the start-up period of the steam boiler can take some hours, the high pressure equipment requires strict ASME code compliance and licensed operators. Substitution of the SBC by a bottoming cycle which requires no water, nor steam, seems an attractive alternative for power generation in the under 50 MW range.

The use of air bottoming with small topping gas turbines offers an increase in power and efficiency without the complexity of the SBC. The present chapter discusses the configuration and performance of the combined-cycle plants based on small and medium gas turbines.

7.2 Background

Wicks (1991) derived the concept of the air bottoming cycle from the theory of the ideal fuel-burning engine by comparing the engine with the Carnot cycle. While the Carnot cycle is ideal for heat sources and heat sinks of constant temperature, this is not the case, when considering fuel burning engines. In an ideal fuel-burning engine, the combustion products are created, and present a finite size hot reservoir, that releases heat over the entire temperature range from their maximum to ambient temperature. When this is put into a T-s diagram, a triangular area can be drawn, instead of the rectangular area in the case of the Carnot cycle. Thus, the cycle consists of isothermal compression, heat addition, and isentropic expansion.

This three-process cycle may not be practical for a single cycle engine because very high pressure ratio will be required. This concept is more realistic for a heat recover-

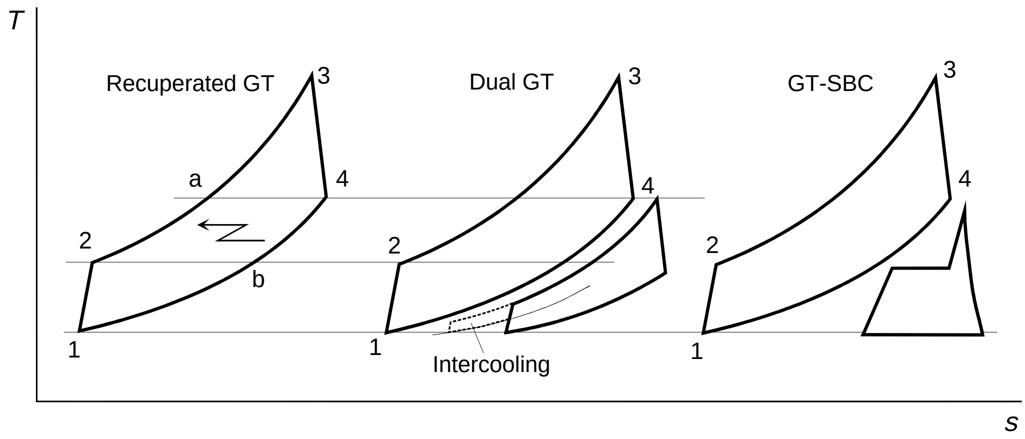

Fig. 7.2. T-s diagram of the recuperated gas turbine cycle, the dual gas turbine combined cycle, and the gas turbine with steam bottoming cycle.

ing bottoming cycle. Since the maximum cycle temperature will be the topping cycle exhaust temperature, rather than the topping cycle firing temperature, the corresponding pressure ratio will not be that high. The gas turbine cycle consisting of an adiabatic compressor and an expander, and a heat exchanger can be applied as an air bottoming cycle. The isothermal compression would require continuous cooling during the compression process. This can be approached with an increasing number of intercoolers, or by the introduction of wet compression (Poletavkin, 1980).

The air bottoming cycle can be compared with both the recuperated gas turbine and the steam bottoming Rankine cycle. As seen on the T-s diagram for the same topping gas turbine (Fig. 7.2), heat recovery in the case of the recuperated cycle is limited by the compressor outlet temperature and cannot proceed below that temperature level (line *4-b*). The dual gas turbine combined cycle has no such limitation, and the use of intercooling can improve the heat recovery even further. In comparison with the steam bottoming cycle, it is clear that the steam boiler has much smaller pinch temperatures than the ABC, although the average ΔT of the former is still comparable to that of the latter, because the heat exchange in the air cycle occurs in a single gas phase.

Along with the utilization of a gas turbine exhaust, other sources of heat can be considered: such as waste heat from a chemical reactor, or an incinerator's furnace. For better ABC performance, the heat temperature, however, should exceed 400 °C.

The air bottoming cycle was proposed to increase efficiency of the simple-cycle gas turbine units operating on Norwegian oil platforms in view of the CO_2 tax (Bolland, 1995). Converting these units into conventional combined-cycle plants was found not feasible due to the considerable weight of the steam bottoming equipment and the boiler feed water requirements. The same considerations inhibit the use of the steam-injected gas turbine (STIG) cycle in off-shore applications (Spector and Patt, 1997), whereas the air cycle offers less weight and an efficiency close to that of the SBC.

The dual gas turbine combined cycle can be implemented as a combined heat and power plant by supplying the outlet air flow, which has a temperature above 200 °C, for heating needs. The hot air may be used either directly as a product, or be seen as a heat transfer medium.

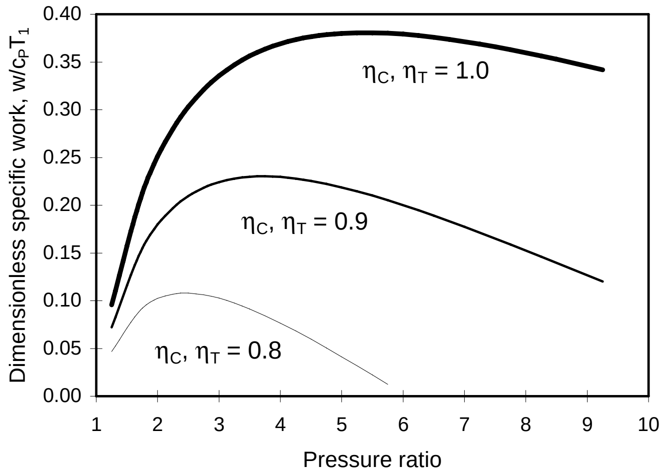

Fig. 7.3. Specific work of the ABC at different levels of turbomachinery efficiency.

7.3 Thermodynamic considerations

A cogeneration plant based on an existing gas turbine engine with the air bottoming was modeled and compared with a conventional gas turbine cogeneration plant. The operational parameters of the ABC were set to typical values of the simulated hardware. Since this is the bottoming cycle, optimization to the maximum power results in the maximum efficiency.

Introducing γ as the ratio of specific heats c_P/c_V, k as $(\gamma-1)/\gamma$, t as the temperature ratio T_{23}/T_{21} (using the designations in Fig. 7.1), the specific work of the ABC per unit air flow can be described by

$$\frac{W}{c_p T_{21}} = t \cdot \eta_T \cdot (1 - r^{-k}) - \frac{1}{\eta_C} \cdot (r^k - 1) = \frac{(t \cdot \eta_T \cdot \eta_C - r^k)(r^k - 1)}{r^k \cdot \eta_C} \qquad (7.1)$$

If a temperature of 480 °C is taken as a typical value of the top cycle temperature (the temperature ratio equals 2.8), then a number of specific-work curves can be plotted for different turbomachinery efficiencies (Fig. 7.3). The chart shows that by improving the efficiency of the compressor and expander from 0.8 to 0.9, twice as much work can be obtained from the cycle. Also, it can be noted that the curve around the optimum point becomes flatter as efficiencies grow.

The top cycle temperature T_{23} is determined by the heat exchanger properties and can be set either by a fixed difference between the hot inlet and cold outlet temperatures ΔT:

$$T_{23} = T_3 - \Delta T, \qquad (7.2)$$

Table 7.1. Performance specifications of the selected gas turbines.

Engine	Power MW	Efficiency % LHV	Pressure ratio -	Exhaust flow kg/s	Exhaust temp. °C
Solar Centaur H	3.78	26.4	9.8	18.4	514
Allison 501-KB5	3.93	28.7	10.1	15.6	549
Allison 571-K	5.91	33.9	12.7	18.7	533
Mitsubishi MF-111A	12.61	30.3	15.0	54.6	547
GE LM2500	21.56	35.4	18.9	68.5	529
ABB GT-10	24.00	33.5	14.0	78.8	541

or by the heat exchanger effectiveness ε

$$\varepsilon = \frac{T_{23} - T_{22}}{T_3 - T_{22}} = \frac{T_{21} \cdot r^k - T_{23} \cdot \eta_C}{T_{21} \cdot r^k - T_3 \cdot \eta_C} \tag{7.3}$$

so that the top cycle temperature can be expressed as:

$$T_{23} = \varepsilon \cdot T_3 + \frac{T_{21} \cdot r^k \cdot (1 - \varepsilon)}{\eta_C} \tag{7.4}$$

The temperature difference in this case is not fixed. For example, at a pressure ratio of 4, a compressor isentropic efficiency of 0.9, and an effectiveness of 0.85, ΔT equals 50 K, and at a pressure ratio of 6 the difference is 40 K.

7.4 Configurations

Because the topping gas turbine outlet temperature determines the performance of the bottoming cycle most, turbines with an exhaust temperature above 500 °C were chosen from the the small and medium power range. Their performance data are presented in Table 7.1.

In the bottoming cycle both compressor and expander had an isentropic efficiency of 90%. Regarding the compressor, schemes were analyzed without intercooling, as well as with one, and with two intercoolers. When intercoolers were applied, water was used as an intercooling medium. The intercooler cooled the air down to 30 °C. The pressure losses at the air side of the heat exchanger were assumed to be 120 mbar, and 20 mbar on the gas side. Heat exchanger effectiveness was set to 85%. Simulations were made with the use of the Gate Cycle code (Enter Software, 1995) at ISO conditions.

The combined heat and power concept with the ABC was compared with a cogeneration plant that incorporates a steam bottoming cycle. The comparison was made on the

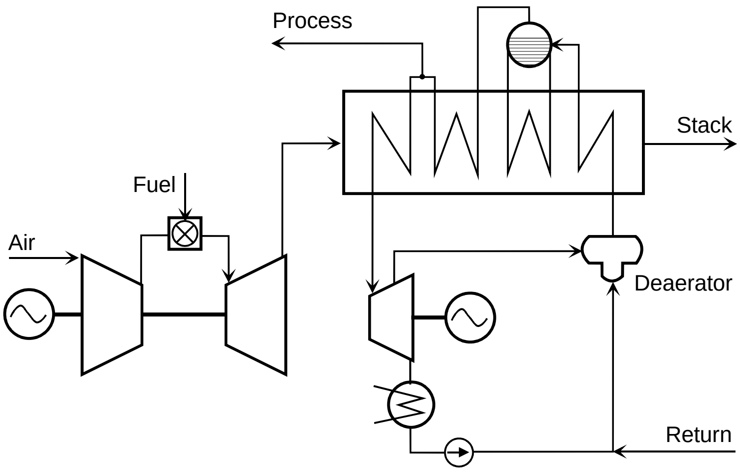

Fig. 5.4. The reference cogeneration plant with the steam bottoming cycle.

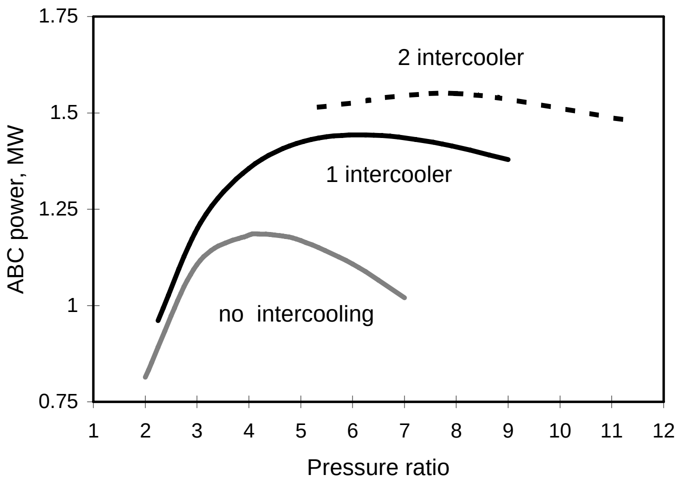

Fig. 7.5. Optimization of the overall pressure ratio in the ABC (Allison 571-K).

basis of an equal amount of exergy delivered to the process at the same temperature. A single-pressure HRSG was chosen as typical for smaller gas turbines (Fig. 7.4). The HRSG is characterized by the pinch temperature of 8 K, and the approach temperature of 5 K. Superheating of 10 K was applied to the process steam to prevent condensation in the distribution grid. In addition, a continuous blow-down was set to 1% of the total flow through the evaporator. In the evaluations, for both ABC and SBC the same topping gas turbines were used. Excessive steam was sent to the condensing steam turbine (an isentropic efficiency of 80%) to generate additional power. The pressure in the condenser was set at 0.05 bar, and the deaerator pressure was maintained at 1.2 bar.

7.5 Performance analysis

Regarding performance of the air bottoming cycle at different pressure ratios, optimum values were found for each topping gas turbine in three bottoming configura-

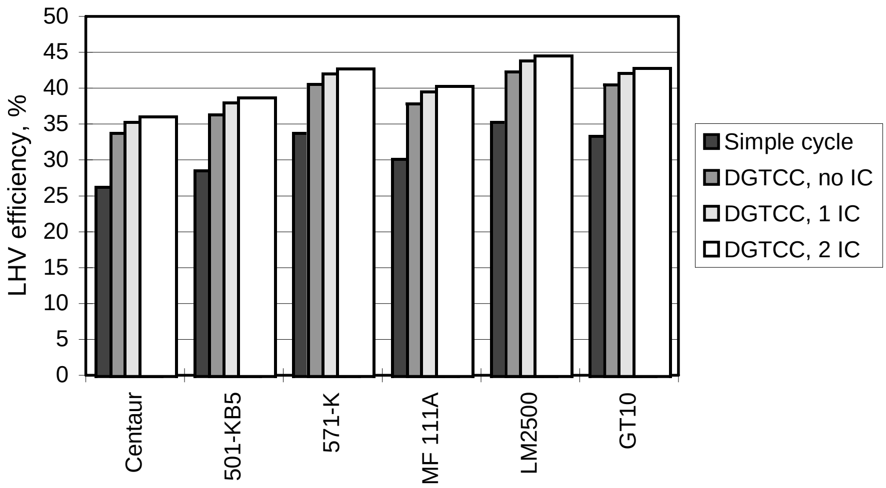

Fig. 7.6. LHV efficiency of the simple-cycle and DGTCC plants.

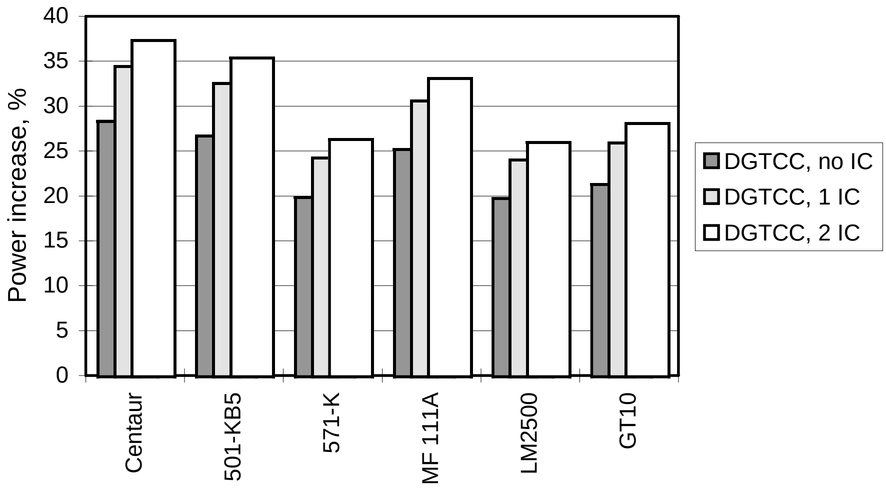

Fig. 7.7. Power increase with respect to the simple-cycle power output.

tions (without intercooling, with one and with two intercoolers). An example for the Allison 571-K is given in Fig. 7.5. The chart indicates a significant effect of intercooling on the ABC performance: an addition of an intercooler increases the power output of the non-intercooled cycle by 22%, and the second intercooler adds 8% more. The optimum pressure of the configurations was 4.2, 6.2, and 6.9, respectively. Comparable figures were obtained for other gas turbines, since they all operate within the similar temperature range and with the same turbomachinery efficiencies. Those also correspond with the figures given by Bolland (1995) for the GE LM2500.

The results of simulation are summarized in Figures 7.6 and 7.7. The application of the air bottoming cycle gives an LHV efficiency gain from 6.8-7.7% points (no intercooling) to 8.3-9.4% points (1 intercooler) and to 9.3-10.2% points (2 intercoolers).

Table 7.2. Comparison of ABC and SBC (Allison 571-K and GE LM 2500).

Gas turbine	System	Total plant power MW	Electric efficiency %	Exergetic efficiency %	Process temperature °C
Allison 571-K	1. Simple cycle	5.9	33.8	32.5	N.A.
	2. ABC, no intercooling	7.0	40.4	47.2	278
	3. ABC, one intercooler	7.4	42.3	46.0	220
	4. ABC, two intercoolers	7.5	43.2	46.3	210
	5. SBC, T_{PROC} as in case 2	7.1	40.9	47.1	273
	6. SBC, T_{PROC} as in case 4	7.3	41.8	44.6	201
GE LM2500	7. Simple cycle	21.6	35.4	34.0	N.A.
	8. ABC, no intercooling	25.6	42.1	49.3	278
	9. ABC, one intercooler	26.9	44.1	48.2	220
	10. ABC, two intercoolers	27.4	45.0	48.4	207
	11. SBC, T_{PROC} as in case 8	26.0	42.6	49.0	267
	12. SBC, T_{PROC} as in case 10	26.5	43.5	46.5	199

These improvements result in overall efficiencies ranging from 36.2% to 44.7%. The best performance was obtained by the Allison 571-K and the GE LM2500. Whereas one intercooler improves the efficiency of a DGTCC by approximately 1.6% points compared to the non-intercooling case, the second one adds only 0.7% points. It is evident that the use of more than two intercoolers is not worthwhile.

The ABC increases a gas turbine plant's power by 20% to 35%, depending on the number of the intercoolers and the gas turbine's simple-cycle efficiency. The increase is especially pronounced in the cases of less-efficient turbines, such as the Solar Centaur H and the Allison 501-KB5 (Fig. 7.7). These turbines produce more waste heat, and although the bottoming cycle helps to recover it, the overall efficiency of these cases remains lower than that of the better engines.

7.6 Cogeneration

Two dual gas turbine combined cycle plants based on the Allison 571-K and the GE LM2500 were compared with conventional cogeneration plants built around the same topping turbines and providing heat of the same exergetic value and temperature. In addition to efficiency of power generation, exergetic efficiency was also calculated as the ratio between the sum of power generated and heat supplied to the fuel input.

In comparing the performance data of ABC and SBC gas turbine plants, a small difference can be seen in the cases without intercooling (Table 7.2). For example, case 2 is characterized by 7 MW of power and 40.4% electric efficiency, which is comparable to the performance data of case 5, where power and efficiency amount to 7.1 MW

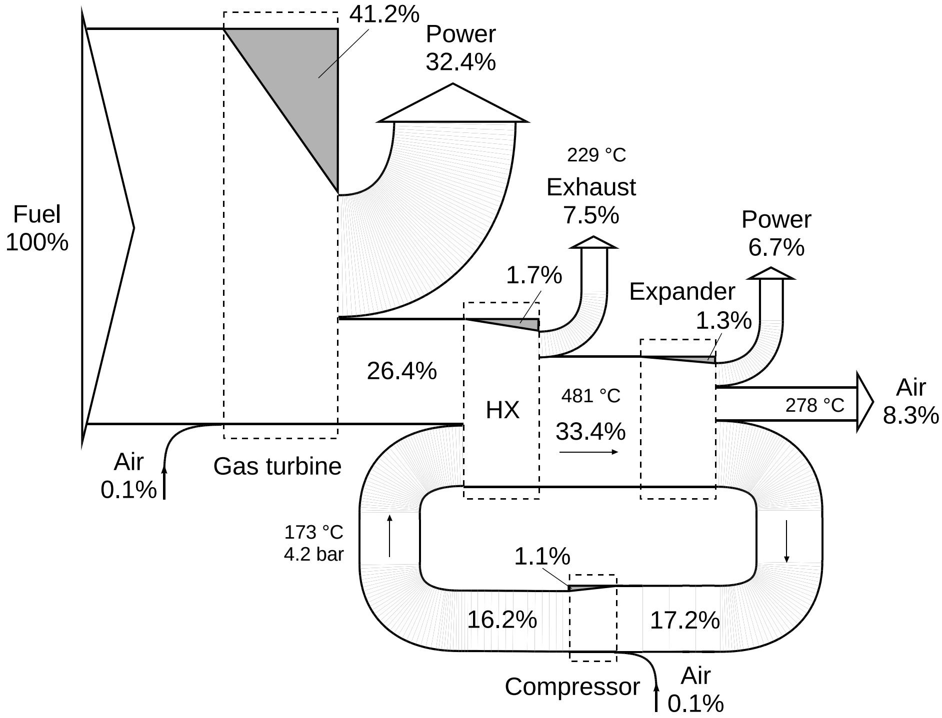

Fig. 7.8. Grassmann diagram of the DGTCC without intercooling based on the Allison 571-K gas turbine.

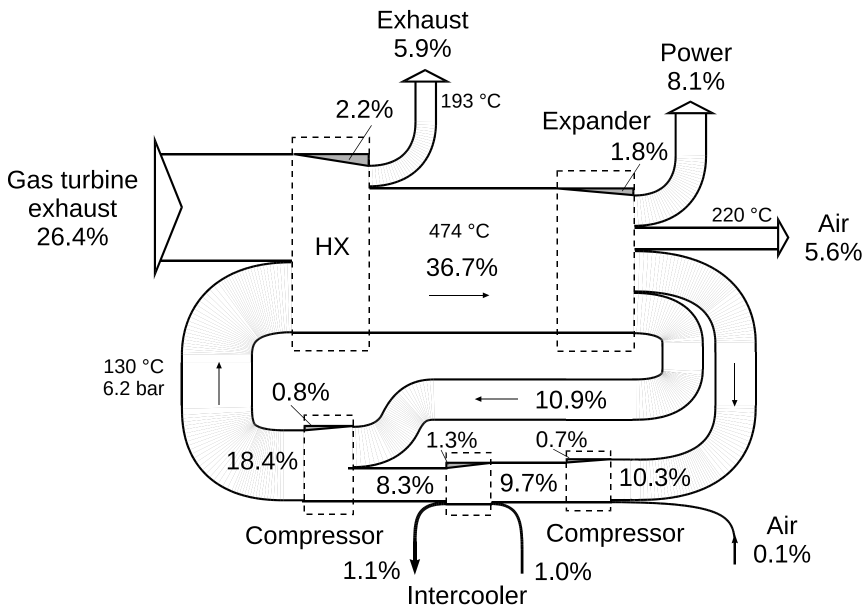

Fig. 7.9. Grassmann diagram of the DGTCC with 1 intercooler based on the Allison 571-K gas turbine, only the bottoming cycle is shown.

and 40.9%, respectively. Similar results were obtained for the LM2500 turbine. When considering ABC with two intercoolers, the difference between ABC and SBC becomes more noticeable, but still lies within 4%.

Regarding exergy, Grassmann diagrams for cases 2 and 3 are given (Figures 7.8 and 7.9). It should be noted that intercooling deteriorates the overall exergetic efficiency by approximately 1.5%, because of a lower heat production rate and some heat loss occurring in the intercoolers (about 0.1% of the fuel input exergy). Intercooling, by decreasing the temperature of the compressor outlet air, allows better utilization of the topping gas turbine exhaust: stack losses drop from 7.5% to 5.9%, and more heat becomes available for the expander (33.4% vs 36.7% in the intercooled case). As a result, power production grows. At a CHP plant intercooling permits a load control, while maintaining a high electric efficiency: when heat production has a priority, intercooling is minimized or completely shut off.

7.7 Applications

Since the DGTCC produces hot air in large quantities, the air can be used to improve the efficiency of a combustion process, e.g. in an atmospheric biomass gasifier or a waste incinerator. This can also be implemented for drying purposes such as silt drying, or in paper and cardboard production. These processes do not require clean air, thus the exhaust from the gas turbine could also be utilized.

It should be noted that the clean, hot air from the ABC is sterile and therefore of special interest for some applications such as food or pharmaceutics industries. The process of drying milk powder, the potato starch production, and other industries, which need a large flow of clean hot air, are listed in Table 7.3.

Among options for air bottoming is its application in a thermal process, where heat of the process is recovered to drive turbomachinery indirectly. This may be necessary, for example, in the case of corrosive combustion products. A chemical plant, a fluidized bed, or a nuclear reactor can be considered as a heat source.

The ABC provides an attractive performance enhancement of simple-cycle gas turbines without the need for a water-steam cycle. This can be useful at a site with scarce water resources or with weight and space constraints. The simple operation of DGTCC allows unmanned operation of the plant at remote sites such as pipelines and off-shore platforms.

Finally, a specific property of air with respect to the gas turbine exhaust gas should be noted. To prevent condensation of corrosive acids in the stack, the combustion products from the gas turbine should not be cooled below the dew point of the acids. Usually 100°C is considered a safe level, although this represents some heat waste. An exhaust flow of heated air, on the contrary, requires no such temperature constraint. The temperature of the heated air can be reduced as low as desired without the occurrence of harmful compounds.

7.8 Conclusions

Implementation of the air bottoming cycle at a gas turbine adds 7 to 10% points to the simple-cycle efficiency, and a rise in the power output of 20-35%. Thus, the total com-

Table 7.3. Process heat temperatures and year load duration of the industries, where the ABC concept can implemented.

Industry, product	**Temperature** °C	**Year load duration** hours
Dairy products	50-220	3000-5000
Bakery	100-200	4000-6000
Starch	170	5000-7000
Coffee-extract	200-250	5000
Tobacco	120	2000
Animal food	400-1000	4000-5000
Paper and cardboard	140	5000
Synthetic resins	140-200	5000-7000
Bulk polymers	50-200	5000-7000
Sludge and silt drying	350-600	3000-5000

bined-cycle efficiency approaches a value of 45% in the case of gas turbines with high simple-cycle efficiency, such as the Allison 571-K and the General Electric LM2500.

The use of intercooling considerably improves the performance of the ABC increasing bottoming cycle power output by 30% in relation to the non-intercooled case. Nevertheless, the introduction of more than two intercoolers does not seem to be justified.

The sensitivity of the ABC performance to the turbomachinery efficiency necessitates the use of the most efficient equipment.

When compared with the steam bottoming cycle, the ABC showed performance values close to and exceeding those of the SBC, while featuring a simpler and robuster design, smaller dimensions, and the absence of a water treatment process.

However, due to the low heat capacity of the media a considerable surface area will be required in the air-gas heat exchanger, and an economic analysis will be needed to compare the air bottoming cycle with other alternative schemes.

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Chapter 8
Partial Oxidation Gas Turbine

Abstract

This chapter addresses the use of the partial oxidation process within a gas turbine cycle. In a partial oxidation reaction fuel is oxidized in a sub-stoichiometric atmosphere at a pressure and a temperature similar to those in the combustion chamber of a conventional turbine. Following the sub-stoichiometric stage, oxidation of the fuel is completed in the final stage. The absence of excess air in the first stages makes it possible to reduce the work required by the compressor and to decrease NO_X formation. To assess the partial oxidation concept, several gas turbine systems were simulated and optimized. The results of the comparison study and exergy analysis of the cycles considered are presented.

8.1 Introduction

Partial oxidation is a reaction in which fuel does not burn completely due to the lack of oxygen, and a gas mixture of carbon monoxide and hydrogen is produced. The temperature of the reaction is determined by the equivalence ratio. Typically, such a reaction proceeds at a temperature of 1300 °C and pressure up to 60 bar. Since no steam is added in this fuel reforming reaction, the ratio H_2/CO is rather low, ranging from 1.6 to 1.8 (Tindall and Crews, 1995). If the temperature is lower, the product may contain some unconverted methane.

The high temperature of the reaction makes it possible to incorporate a PO reactor in a gas turbine cycle. This can be implemented in a two-stage gas turbine cycle, with the PO reactor as a combustion chamber in the first stage. Such a cycle is presented schematically in Fig. 8.1. After compression (process *1-2*), air is fed to the partial oxidation reactor, where a synthesis gas is formed. The gas then is expanded in the high-pressure turbine to an intermediate pressure (process *3-a*), and secondary air is introduced before the final expansion stage to complete oxidation of the fuel (process *b-4*).

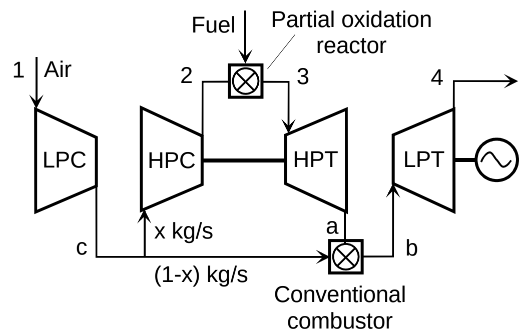

Fig. 8.1. Basic scheme of the POGT cycle.

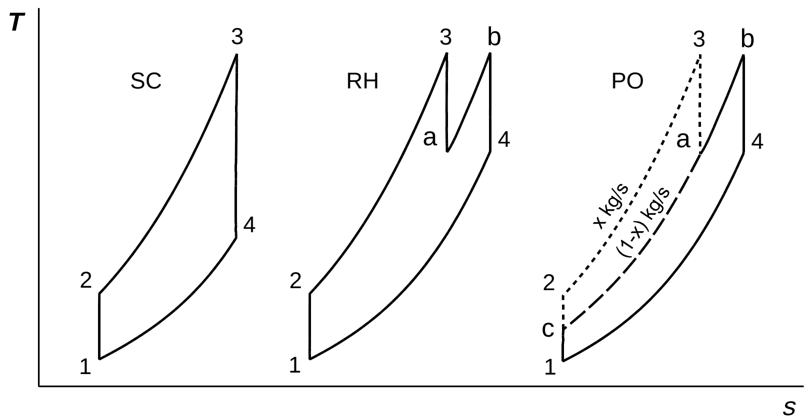

Fig. 8.2. T-s diagrams of the simple, reheat, and PO gas turbine cycles.

The T-s diagram (Fig. 8.2) illustrates the difference between the PO and reheat gas turbine cycles: in the former cycle only a part of the working fluid is brought to the maximum pressure. The flow split in point *c* is mixed again in point *a*. Fraction *x* is determined in the PO reaction by the required turbine inlet temperature T_3. The value of *x* lies between 0.15 and 0.22 for the temperature range of 1200 to 1400 °C.

The partial oxidation gas turbine (POGT) features low NO_X forming due to the sub-stoichiometric atmosphere in the high-pressure stage and better combustion characteristics of the synthesis gas (shorter flame, less air excess) in the second stage. NO_X forming under fuel-rich conditions were reported to be as low as 2.6 ppm (Yamamoto et al., 1997). However, many investigators indicate that soot-forming can take place in the PO process, especially at high equivalence ratio values. Experiments showed that up to 15% of the methane's carbon can be converted into soot in a reaction with an equivalence ratio of 3 (Rabou, 1996). This problem calls for the use of steam injection, which in addition to soot suppression, increases hydrogen content in the synthesis gas and the mass flow across the expander.

8.2 Background

The POGT concept was described in the early 1970s by J. Ribesse (1971) and by V.M. Maslennikov (Christianovich et al, 1975). Ribesse's scheme represents a compressor,

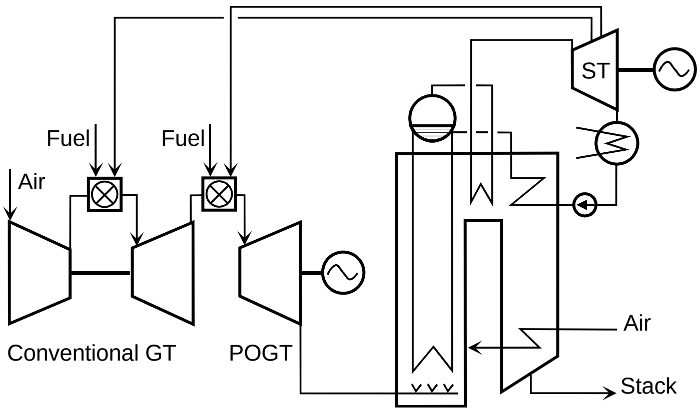

Fig. 8.3. Repowering of an existing steam boiler by the use of the PO gas turbine.

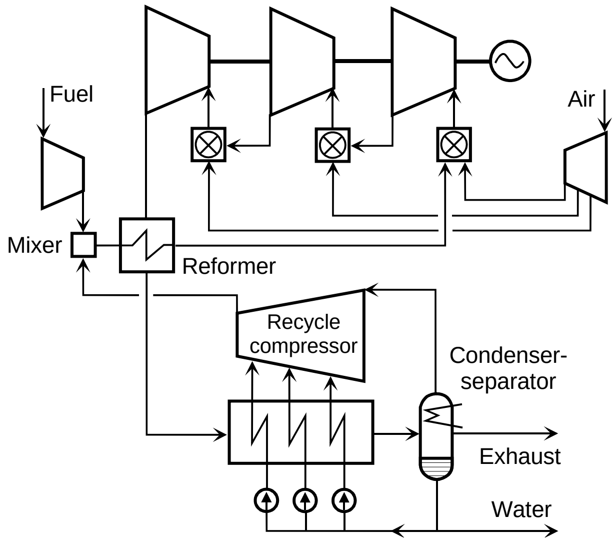

Fig. 8.4. Gas turbine cycle with flue gas recycling, thermochemical recuperation and wet compression.

a PO reactor and an expander. The system generates a synthesis gas which can subsequently be fed into a specific process. The lower heating value of the product ranges from 6 to 9 MJ/kg. Adding this value to the generated power gives an LHV efficiency of 36 to 38%. In his calculations, Ribesse assumed a turbomachinery polytropic efficiency of 85% and a turbine inlet temperature of 807 °C.

Christianovich and Maslennikov (1976) proposed use of the partial oxidation process to reform oil residue into a fuel gas, from which sulphur components in form of H_2S can be removed. After sulphur removal, the fuel gas can be burned in a steam power plant or in a gas turbine. The latter option in the form of a natural gas-fired power plant was analyzed by the same group of researchers (Maslennikov and Shterenberg, 1992, Maslennikov et al., 1997). They found that the so-called incremental efficiency of the POGT unit in the retrofit modifications could be as high as 80% (calculated as a ratio of the incremental power to the incremental fuel consumption). A simplified scheme of a repowering arrangement is shown in Fig.8.3.

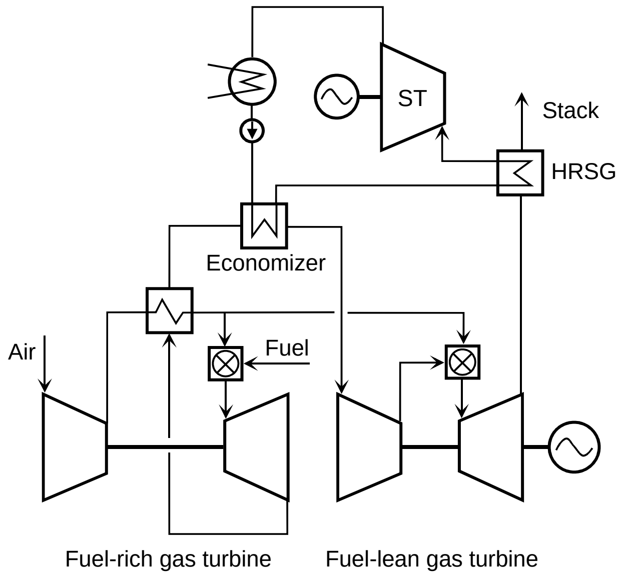

Fig. 8.5. The Chemical gas turbine.

The partial oxidation process was outlined in a patent by Nurse (1991), where it serves to gasify carbonaceous fuels (oil, coal, natural gas, naphtha, etc.) for subsequent use in a gas turbine. The patent also describes means for removing sulphur from the synthesis gas prior to the complete combustion.

The POGT concept was investigated by Westinghouse Electric Corporation and the Institute of Gas Technology in a study within the U.S. Department of Energy's Advanced Turbine Systems Program (Rabovitser et al., 1996). Such factors as the low air excess, low air compressor work, and multi-stage expansion allowed the authors to claim LHV efficiencies up to 68% in a combined-cycle configuration.

Hodrien and Fairbairn (1993) in their survey on advanced cycles for British Gas mentioned the POGT as a highly promising cycle with a potential efficiency above 60%.

Harvey et al. (1995) studied a gas turbine cycle with thermochemical recuperation, recycling of the exhaust gases, partial oxidation and intercooling (Fig. 8.4). The proposed system obtained an LHV efficiency of 65.4%. However, it was concluded that the gain in cycle efficiency due to the improved fuel oxidation process is rather small (1.4% points).

Arai et al. (1995) proposed to employ the partial oxidation reaction in the Chemical Gas Turbine. In this configuration fuel-rich combustion in the first stage allows the use of carbon fiber reinforced carbon composites (C-C composites) as a material for the turbine blades. The composites can withstand higher temperatures than metal materials, but are subject to rapid degradation if exposed to a small amount of oxygen (Arai and Kobayashi, 1997). Recent tests confirmed the material's ability to maintain its integrity even at 1800 °C (Anonymous, 1997).

A basic scheme of this concept is given in Fig. 8.5. In the fuel-rich section, the syngas is expanded to the atmospheric pressure and after cooling in the recuperator and the steam generator, compressed again to the maximum pressure. With a TIT of the first, fuel-rich turbine of 1500 °C, Yamamoto et al. (1997) found the thermal efficiency of the Chemical gas turbine to be as high as 44% in the simple cycle and 62%

in a combined-cycle configuration. The combined-cycle scheme was also modeled by Lior (1995): with a single-pressure HRSG, TIT of the first and the second turbines of 1500 °C and 1300 °C, respectively, and a pressure ratio of 20, the plant was calculated to have 66% LHV efficiency.

A gas turbine system with partial oxidation was selected for an evaluation study within the framework of the New Energy Conversion Programme (NECT) by the Netherlands Agency for Energy and Environment Novem (Korobitsyn, 1996). The POGT concept in different configurations was analyzed by Deen (1996) at the Netherlands Energy Research Foundation ECN. Twelve configurations were simulated and optimized with the use of the Aspen Plus process simulation package (AspenTech, 1996). All models were based on a gas turbine system with turbomachinery polytropic efficiency of 88%, a TIT of 1230 °C, combustion chamber pressure loss of 2.5%, recuperator ΔT of 40K, and recuperator pressure loss of 5%.

The results of that study showed that the basic POGT in a combined-cycle configuration has a minor advantage above the conventional gas turbine system, which was only 1.1% points. The use of advanced techniques such as recuperation, intercooling, and exhaust gas recycling with thermochemical recuperation resulted in LHV efficiencies up to 63.2%. The use of the exhaust gas recycling in a POGT configuration improved the efficiency by 3.1% points compared to a similar conventional gas turbine cycle. Intercooling had a slight positive effect on overall efficiency.

8.3 Configurations

Another study was carried out at ECN in order to make a definitive judgement on the POGT cycle. Initially it was limited to one partial oxidation stage and one conventional stage without a bottoming cycle. One of the arguments in favour of the POGT cycle was that the sub-stoichiometric combustion reduces air excess, and therefore, required compressor work. However, it also diminishes the amount of work obtained in the expander, and no efficiency rise can be expected. Moreover, the PO stage delivers a rather small fraction of work compared to the total cycle power, and hence, its influence on overall performance is quite limited.

Regarding the fraction x of the total mass flow, as evident from Fig. 8.2, at $x = 1$, the PO cycles becomes a reheat cycle, and at $x = 0$, it transforms into a simple cycle. Therefore, such a cycle might be seen as a sort of intermediate cycle without any specific advantages above either the simple or reheat cycle. This was shown first for the cycles based on ideal conditions (isentropic compression and expansion, air as the working fluid with constant specific heats, no pressure losses), and then, for the cycles based on real efficiencies, real gas properties and pressure losses.

The equations of efficiency and specific work for the ideal simple, reheat, and PO cycles use the following notations: t as the cycle temperature ratio, γ as the specific heats ratio, m as the pressure ratio exponent, and r as the pressure ratio:

$$t = \frac{T_3}{T_1}, \quad \gamma = \frac{c_p}{c_v}, \quad m = \frac{\gamma - 1}{\gamma} \tag{8.1}$$

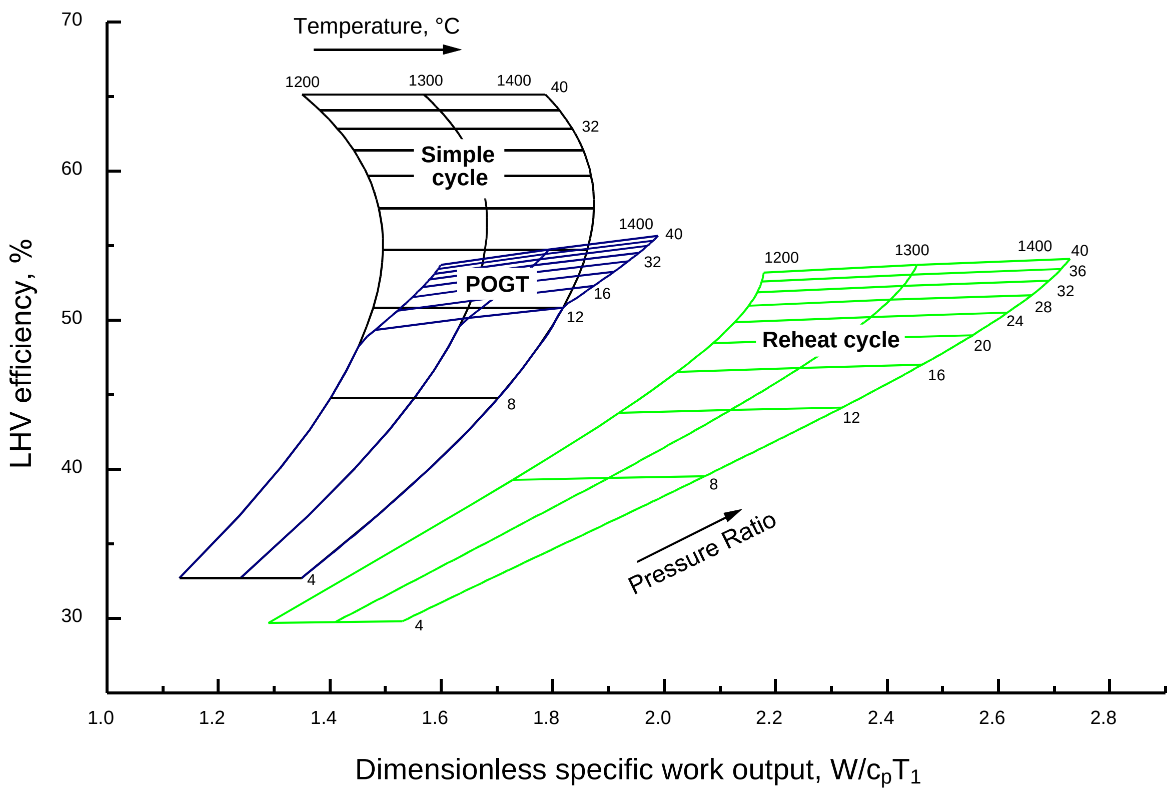

Fig. 8.6. Performance map of the simple, reheat, and PO cycles, calculated as ideal cycles.

The specific work output is given in dimensionless form as the ratio of the net work output to the product of the air heat capacity c_p and the air inlet temperature T_1.

Efficiency and specific work of the simple cycle can be described as:

$$\eta = 1 - \frac{1}{r^m} \tag{8.2}$$

$$\frac{W}{c_p T_1} = t - r^m + 1 - \frac{t}{r^m} \tag{8.3}$$

For the reheat cycle:

$$\eta = \frac{2t - r^m - \frac{2t}{\sqrt{r^m}} + 1}{2t - r^m - \frac{t}{\sqrt{r^m}}} \tag{8.4}$$

$$\frac{W}{c_p T_1} = 2t - r^m + 1 - \frac{2t}{\sqrt{r^m}} \tag{8.5}$$

And for the partial oxidation cycle:

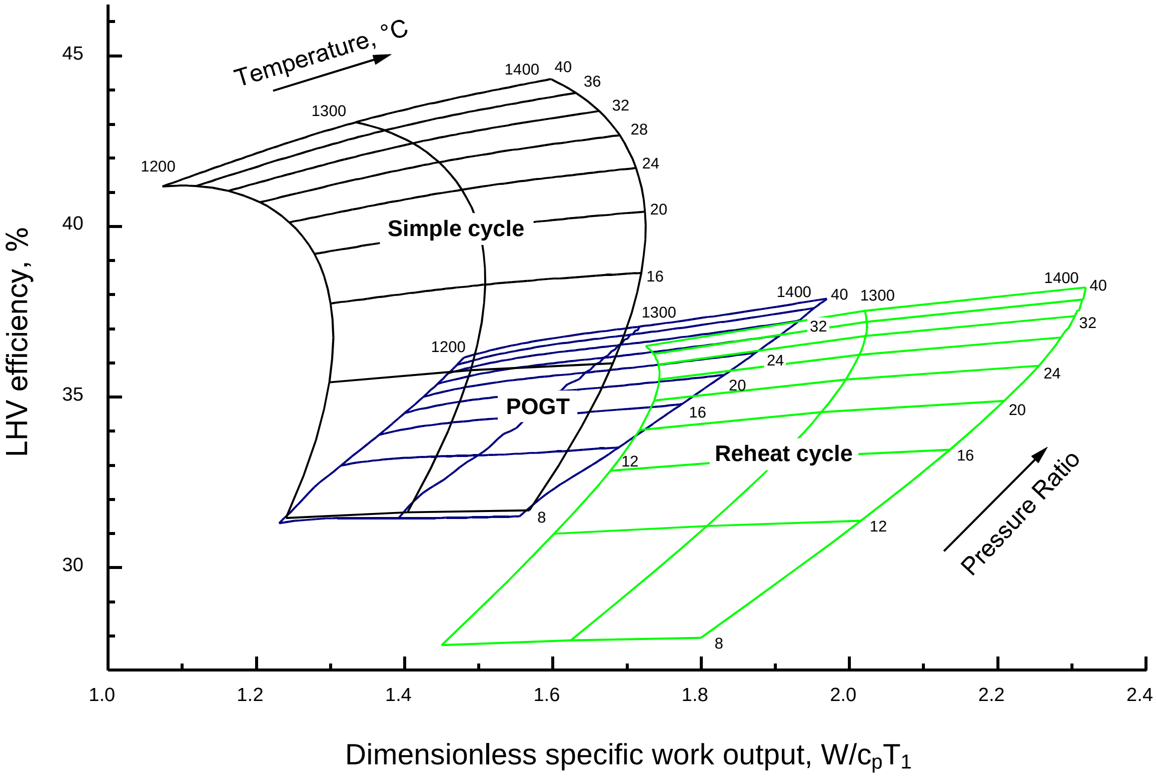

Fig. 8.7. Performance map of the simple, reheat, and PO cycles, calculated as real cycles.

$$\eta = \frac{t\cdot\left(1-\left(\frac{P_1}{P_c}\right)^m\right)+xt\cdot\left(1-\left(\frac{P_c}{P_2}\right)^m\right)-x\cdot\left(\frac{P_2}{P_1}\right)^m+(x-1)\cdot\left(\frac{P_c}{P_1}\right)^m+1}{t\cdot\left(1-\left(\frac{P_c}{P_2}\right)^m\right)+(1-x)\cdot\left(t\cdot\left(\frac{P_c}{P_2}\right)^k-\left(\frac{P_c}{P_1}\right)^m\right)+xt\cdot\left(1-\left(\frac{1}{t}\right)\cdot\left(\frac{P_2}{P_1}\right)^m\right)} \tag{8.6}$$

$$\frac{W}{c_p T_1} = t\cdot\left(1-\left(\frac{P_1}{P_c}\right)^m\right)+xt\cdot\left(1-\left(\frac{P_c}{P_2}\right)^m\right)-x\cdot\left(\frac{P_2}{P_1}\right)^m+(x-1)\cdot\left(\frac{P_c}{P_1}\right)^m+1 \tag{8.7}$$

The performance map based on these equations is displayed in Fig. 8.6. The intermediate pressure in the reheat and PO cycles was optimized for the maximum work output. In the latter cycle, the parameter x was set at 0.2. The chart indicates that at lower pressures POGT cycle resembles the simple cycle, and only at pressures above 12 bar it produces more work. Nevertheless, the PO cycle cannot reach the values of specific work obtained in the reheat cycle. Efficiency of the PO cycle is, however, slightly higher than that of the reheat turbine.

A more realistic approach to assess performance of the simple, reheat and PO cycles with and without recuperation was carried out with the use of the Aspen Plus simulation package. The cycles were modeled with the following assumptions:

Table 8.1. Main results of performance and exergy analysis for the cases with a TIT of 1400 °C.

Case		without recuperation SC	RH	PO	with recuperation SCR	RHR	POR	POX
Pressure	bar	18	6.5/40	9.5/40	13	9.8/40	13.7/40	40
Specific work	kJ/kg	501	678	579	469	626	529	983
LHV efficiency	%	39.40	37.88	37.30	48.69	42.66	47.98	14.04
Exhaust temperature	°C	655	887	798	438	694	489	483
Exergy:								
Power	%	37.96	36.49	35.93	46.91	41.10	46.22	13.53
Exhaust	%	27.46	32.59	31.40	18.50	26.61	19.68	72.10
Total	%	65.42	69.08	67.33	65.41	67.71	65.90	85.63
Efficiency in CC	%	56.91	58.98	57.60	59.68	59.46	59.80	N.A.
Combustion loss	%	28.37	24.98	27.86	26.22	24.45	25.57	12.54

Turbine inlet temperatures	1200, 1300, 1400 °C
Maximum pressure	40 bar
Polytropic efficiency, compressor	90%
Polytropic efficiency, expander	88%
Mechanical efficiency, generator	98%
Pressure loss in the combustion chamber	2%
Minimal temperature difference in the recuperator	40K
Pressure loss in the recuperator	5%

For the sake of simplicity, turbine cooling was not modelled. Methane was considered as a fuel. The ambient conditions were at 1.013 bar, 25 °C, 60% relative humidity. It was assumed that the methane would be delivered to the plant at required pressure, since the pressure in the gas pipelines at an industrial consumer's site can be up to 60 bar.

The simulations were made using the Aspen Plus package with the Lee-Kesler-Plöcker properties set. Since a polytropic expander is not included in the standard Aspen Plus unit operation models set, a user-defined model was set up (Kers, 1997).

8.4 Performance analysis

The curves of efficiency and specific work of the cycles without recuperation are presented in Fig. 8.7. The simple cycle (SC) shows a considerable advantage in efficiency over the other two cycles, 44% against 38%, but produces much less in terms of specific work. The difference between the reheat (RH) and PO cycles is not such pronounced regarding efficiency, but quite noticeable in specific work. At maximum, the value of specific work equals 2.3 (or 678 kJ/kg in absolute units) in the reheat cycle, whereas for the PO cycle it is about 1.9 (579 kJ/kg).

The chart shows that the performance of the POGT cycle improves almost linearly with a rise in the pressure ratio and the turbine inlet temperature, while for both the SC and RH cycles their maximum values of specific work can be seen within the given range of pressures and temperatures. The chart supports the assumption that the

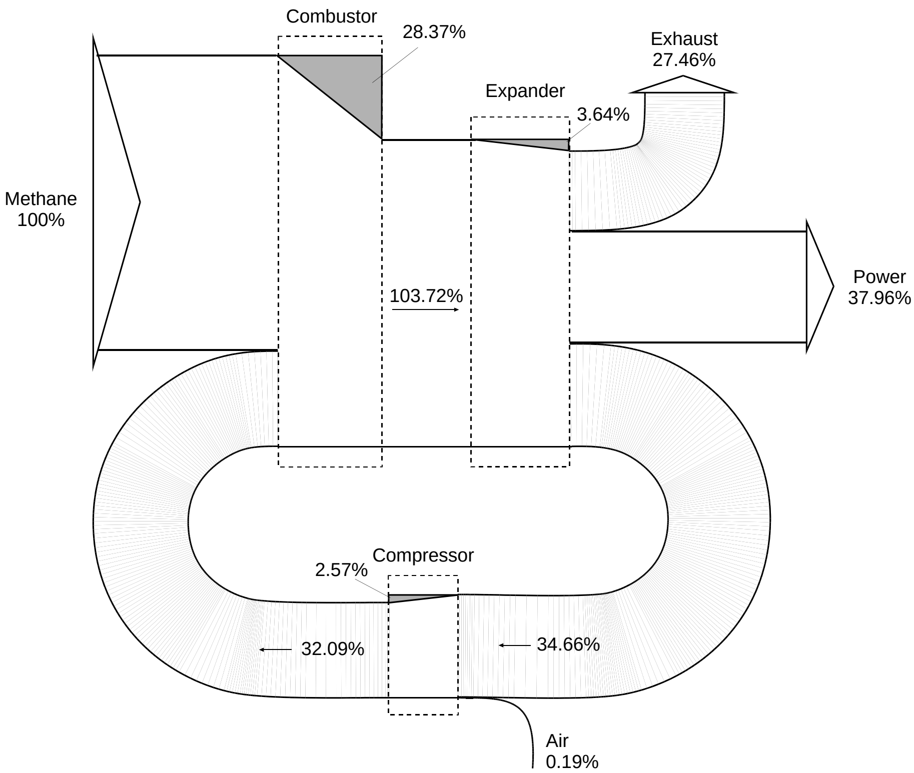

Fig. 8.8. Grassmann diagram of the simple cycle.

character of the PO cycle lies in between the SC and RH cycles: at lower pressures it is closer to the simple cycle, and at higher pressures it approaches the reheat cycle.

Table 8.1 gives some performance values for the cycles with a TIT of 1400 °C at maximum specific work conditions, which correspond to the power maximization principle (Bejan, 1988) and are the most beneficial for combined cycle operation (Horlock, 1995). At these conditions, the simple cycle still has the highest LHV efficiency with a 1.5-2.1% points advantage. The high exhaust temperature enables the use of a recuperator. The schemes with recuperation (SCR, RHR, and POR) were also optimized for the maximum work output. In the case of the PO turbine, the recuperation was applied only to the low pressure flow (line *c-a* in Fig. 8.2). As seen from the table, recuperation adds from 9.4% (in SCR) to 10.8% points (in POR) to the LHV efficiency, while the level of exhaust temperature is still suitable for utilization in a steam bottoming cycle. The improvement in LHV efficiency of the reheat cycle is lower than in other cases; this is caused by high pressure in the compressor, and thus a limited temperature range for recuperation. In this cycle the compressor outlet temperature at 40 bar is about 640 °C. In the comparable POGT cycle, heat exchange occurs at lower pressure, and hence within a wider temperature range.

8.5 Exergy analysis

An exergy analysis was performed using a FORTRAN subroutine for the Aspen Plus simulator (Folke, 1997). This subroutine calculates values of physical, chemical and

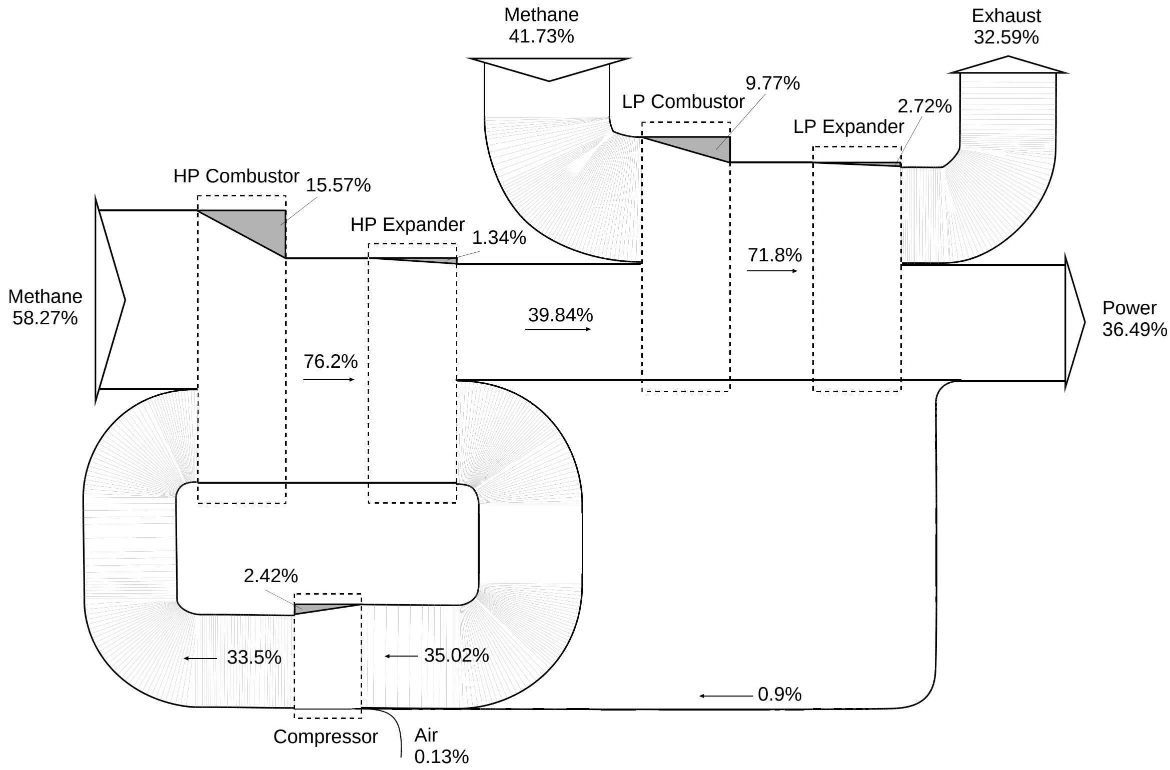

Fig. 8.9. Grassmann diagram of the reheat cycle.

mixing exergy, and includes them in the Aspen Plus output. The original subroutine was modified, and the reference chemical exergy values were substituted by those from Szargut (1988).

Results of the exergy analysis are presented in the form of Grassmann diagrams (Figures 8.8-8.10). The combustion process, which proceeds in a highly irreversible manner, is responsible for the greatest exergy losses within a gas turbine plant. The use of staged combustion reduced the combustion losses from 28.4% in the simple cycle to 25% in the reheat configuration. This effect is due to the high temperature of the gases before entering the LP combustion chamber. Since in the PO cycle most of the combustion air is not preheated, a minor improvement was observed: the combustion losses amount to 27.9%.

Regarding the performance of the cycles in a combined-cycle modification, exergy efficiency figures for different types of steam bottoming cycles calculated by Bolland (1990) were used. The efficiencies varied from 65% for a dual-pressure HRSG to 71% for triple-pressure supercritical reheat cycles. To estimate the performance of the cycles in the combined cycle, the exergy value of the exhaust was multiplied by the bottoming cycle efficiency. A value of 69%, which is typical for an unfired triple-pressure reheat cycle, gives the following combined-cycle efficiencies: for the simple-cycle gas turbine 56.9%, for the reheat gas turbine 59.0%, and for the POGT 57.6%. The cases with recuperation show little difference between them, the efficiency is about 59.6%.

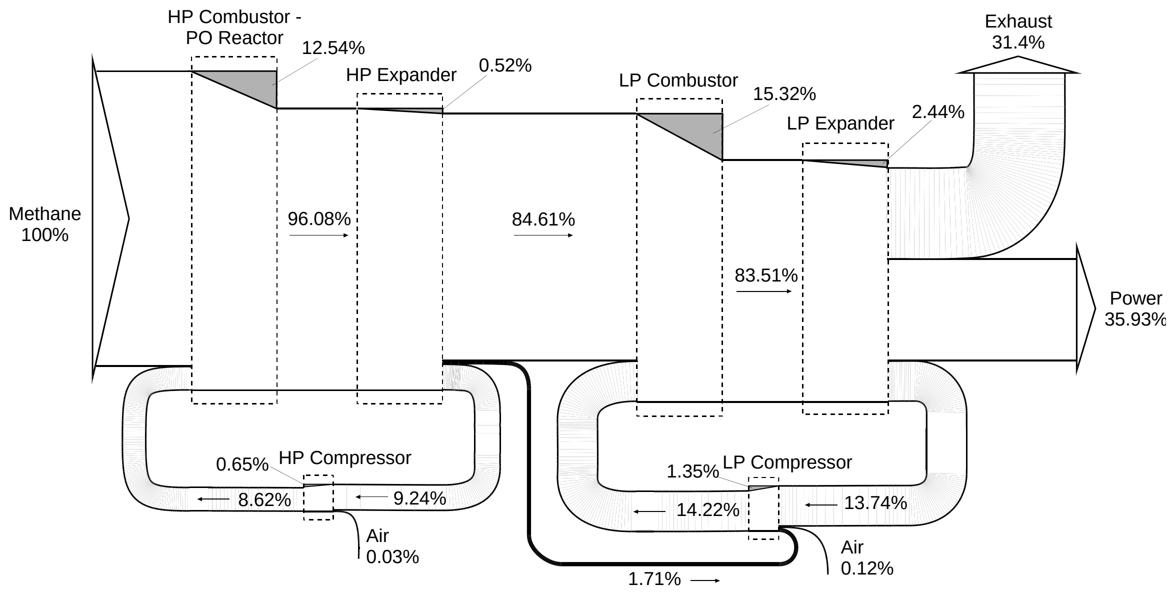

Fig. 8.10. Grassmann diagram of the POGT cycle.

As seen in Fig. 8.10, the PO stage itself has an exergetic efficiency of about 85%, so the stage can be implemented as a stand-alone apparatus by allowing expansion to the atmospheric pressure. The exergetic efficiency of such a unit (case POX in Table 8.1) equals 85.6 %, which is made up of synthesis gas exergy (72.1%) and power (13.5%). The high value of the specific work is due to the sub-stoichiometric combustion (smaller air flow).

If the efficiency of the POX unit is defined as the ratio between the power production to the difference in chemical exergy between the natural and synthesis gases:

$$\eta = \frac{P + E_{EXHAUST}^{PHYS} \cdot f_{CC}}{E_{FUEL}^{CHEM} - E_{EXHAUST}^{CHEM}} \tag{8.8}$$

then the POX unit will obtain an efficiency of 52.3%. Here, the same factor of combined-cycle efficiency is applied to the physical exergy of the outgoing gas (if the physical exergy of the synthesis gas is disregarded, the efficiency is 43.8%). The synthesis gas mixture can be either utilized in a direct-fired boiler, or supplied to a syngas consumer.

Another possible implementation of the PO concept in a gas turbine is a replacement of the low-pressure conventional combustor by a high-temperature fuel cell, which will reduce the exergy losses; an increase in overall efficiency by several percentage points can be projected. The positive effect of integration of a pressurized solid oxide fuel cell in a conventional gas turbine has been indicated, for example, by Parker and Bevc (1996).

8.6 Conclusions

A gas turbine system with partial oxidation show similar performance as conventional gas turbine systems within a feasible range of pressures and temperatures. Of simple, reheat and POGT cycles, the reheat gas turbine has the highest value of the specific work.

The low level of air excess does not improve the gas turbine efficiency, but can be expected to diminish NO_X forming. The reduced atmosphere in the PO stage permits the use of C-C composites with higher turbine inlet temperatures, and therefore, higher efficiencies of the gas turbine cycle.

The high exergetic efficiency of the PO stage indicates the possibility of its use for co-production of synthesis gas and power. The synthesis gas can be utilized in steam-plant repowering schemes, or in chemical plants. Also, the implementation of the POGT as a subsystem in a fuel cell-gas turbine combined cycle plant is possible.

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Appendix 1
Nomenclature

Symbols

c	heat capacity
E	exergy
F	fuel consumption
f	factor
H	enthalpy
m	pressure ratio exponent, $(\gamma-1)/\gamma$
p	pressure
P	power production
Q	heat
S	entropy
t	temperature ratio
T	temperature
x	fraction of flow
γ	ratio of specific heat capacities, c_p/c_v
Δ	difference
ε	effectiveness of a heat exchanger
λ	heat-to-power ratio
η	efficiency

Superscripts

CHEM	chemical exergy
P	related to power
PHYS	physical exergy
Q	related to heat

Subscripts

ACC	heat pump acceptor
C	compressor
CC	combined cycle
CG	cogeneration
D	process demand
el	electric
EX	exergetic
FU	fuel utilization
HP	heat pump
REF	reference
REJ	heat pump rejector
SEP	separate production of heat and power
T	turbine
th	thermal

Abbreviations

AFC	Alkaline Fuel Cell
ASME	The American Society of Mechanical Engineers
CAGT	Collaborative Advanced Gas Turbine programme
CC	combined cycle
CFC	chlorofluorocarbons
CHP	combined production of heat and power
COP	coefficient of performance
CRGT	chemically-recuperated gas turbine
CRISTIG	chemically-recuperated intercooled steam-injected gas turbine
DCSS	Distillation Condensation Subsystem (in the Kalina cycle)
DOE	US Department of Energy
DRIASI	dual-recuperated intercooled aftercooled steam-injected cycle
ECN	Netherlands Energy Research Foundation
EFCC	externally-fired combined cycle
EHP	electric heat pump
EPRI	Electric Power Research Institute
FGD	flue gas desulphurization
FESR	fuel energy savings ratio
FUE	fuel utilization efficiency
GT	gas turbine
HAHP	heat-activated heat pump
HAT	humid air turbine
HHV	higher heating value
HP	high pressure
HRSG	heat recovery steam generator

IGCC	Integrated Coal Gasification Combined Cycle
IP	intermediate pressure
LHV	lower heating value
LP	low pressure
MCFC	Molten Carbonate Fuel Cell
MSW	municipal solid waste
NECT	New Energy Conversion Technologies program
Novem	Netherlands Agency for Energy and Environment
PAFC	Phosphoric Acid Fuel Cell
PC	pulverized coal fired power plant
PFBC	Pressurized Fluidized Bed Combustion
POGT	partial oxidation gas turbine
RH	reheat gas turbine cycle
SBC	steam bottoming cycle
SC	simple gas turbine cycle
SF	supplementary firing
SH	superheater
SOFC	Solid Oxide Fuel Cell
SPFC	Solid Polymer Fuel Cell
ST	steam turbine
STIG	steam-injected gas turbine
TIT	gas turbine inlet temperature
WTE	waste-to-energy plant

About the author

After graduating from the Moscow Institute of Engineering and Management with specialization in power engineering, Mikhail Korobitsyn joined Lenenergo, the utility of the North-Western Russia. He started his career as a maintenance engineer at a 300 MW cogeneration plant in Leningrad (St. Petersburg). Between 1992 and 1994 he worked at Comprimo Engineers and Contractors BV in Amsterdam, where he was involved in several cogeneration projects. In 1994 he started a PhD project at the Chair of Energy Technology of the Department of Mechanical Engineering, University of Twente. Since 1997 he is a research scientist at the business unit Fuels, Conversion and Environment of the Netherlands Energy Research Foundation ECN.