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Survey of modern power plants driven by diesel and gas engines

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ABSTRACT

This paper surveys the latest technology of power plants driven by reciprocating internal combustion (IC) engines, from information collected from publications made mainly during the 1990's.

Diesel and gas engines are considered competitive prime movers in power production due mainly to their high full- and part-load brake thermal efficiency, ability to burn different fuels, short construction time and fast start-ups.

The market for engine power plants has grown rapidly, with estimated total orders for reciprocating engines of 1 MW output and more reaching the 5000 unit level, (10 GW), between June 1995 and May 1996. Industrialized countries much prefer combined heat and power (CHP) production.

Intense interest has been shown in recent years in alternative gas fuels; natural gas appears to be the most promising, but liquid petroleum gas, gas from sewage disposal plants, landfill gas and other biogases, as well as wood gas have also been recognized as other alternatives. Liquid alternatives such as fusel and pyrolysis oil have also been mentioned, in addition to information on coal burning engines.

The percentage of gas engines used has increased and different ones are being developed, based on either the traditional spark ignition (SI), dual-fuel technology or the more recent high pressure gas injection system.

In cold climates, energy production is largely based on CHP plants. Waste heat is utilized for local, regional or district heating or for industrial uses like drying, heating, cooling etc. Even radiative and convective heat from gen-set surfaces are employed, and boilers are used with exhaust outlet temperatures of below dew point. Combined cycle schemes, including turbo compound systems and steam turbines, are also incorporated into engine power plants in order to increase output and efficiency.

Two-stroke, low-speed diesel engine plants show the highest electric efficiencies, with combined cycle plants reaching up to 54%, while gas engine plants achieved between 35% and 47%. The total efficiency of a CHP plant depends on its heat recovery system, recording at its highest rating 98% efficiency.

Exhaust emissions of IC engine power plants must be reduced both by internal and post-combustion methods. The lean-burn SI gas engines seem better with regard to engine-out emissions, while other gas and oil-driven engines with higher oxides of nitrogen emissions are worse. The paper deals only with post-combustion exhaust cleaning systems, reporting on the development of selective catalytic processes (SCR) and three-way catalysts. Data was also collected on combined oxi-cat and SCR reactors and NO reduction concepts that utilize other media than ammonia or urea, as well as more advanced post-combustion methods.

PREFACE

The gas and diesel engine power plant market has grown rapidly, particularly during the 1990s. Information about state-of-the-art and development trends of plants is required for both plant marketing and research and development activities. The technology and efficiency of internal combustion (IC) engine power stations may well have to be dealt with more comprehensively in Finnish Universities and Polytechnics in the future.

This literature review of IC engine power plants has been carried out for VTT Energy as a technology transfer commission of the Turku Polytechnic. I would like to thank Mr. Seppo Viinikainen, Research Manager at VTT, Dr. Nils-Olof Nylund, Group Manager, and Mr. Matti Kytö, Senior Research Scientist, for their confidence, for an interesting subject, and for the opportunity to publish this text.

Mr. Veikko Välimaa, Head of the Mechanical Engineering Department of the Turku Polytechnic, has given me much support and made it possible for me to do this work, for which I am most grateful. Mr. Reijo Halme, Vice-President of the Polytechnic, provided financial support for the final revision of this paper. My sincere thanks to him as well.

I hope, this paper would serve as a survey of recent technology in the field. I intend to use this paper as course material in the IC engine technology area within my teaching.

Turku, April 1997

The Author

CONTENTS

ABBREVIATIONS

BHKW	(engine driven) combined heat and power plant (in German: Blockheiz kraftwerk)
BMEP	brake mean effective pressure
BTE	brake thermal efficiency
CC	combined cycle
CHP	combined heat and power
DCC	diesel combined technology
DCCC	diesel/coal combined cycle
DF	dual-fuel
DFO	diesel fuel oil
GCV	gross calorific value
GD	gas diesel
GI	gas injection
GT	gas turbine
HFO	heavy fuel oil
HPHW	high pressure hot water
IC	internal combustion
LCV	low calorific value
LHV	lower heating value
MDE	MAN Dezentrale Energiesysteme GmbH
MG	mining gas
MHKW	engine driven combined heat and power plant (in German: Motorenheizkraft-werk)
NMHC	non-methane hydrocarbons
OC	oxidation catalyst
SC	single cycle
SCR	selective catalytic reduction
SFOC	specific fuel oil consumption
SI	spark-ignited, spark ignition
SLOC	specific lubrication oil consumption
SNR	selective non-catalytic reduction
TCS	turbo compound system
TDC	top dead center
TWC	three-way catalyst

1 INTRODUCTION

1.1 Chances of diesel and gas engines

Diesel and gas engines are considered competitive prime movers in power production due to their several advantages. The brake thermal efficiency (BTE) of modern reciprocating internal combustion (IC) engines is high, the largest diesel engines reaching more than 50%. Only large combined gas and steam turbine plants are able to achieve still higher fuel conversion efficiencies. Furthermore, diesel engines are economic even at part loads, the BTE remaining at a high level in the load range of 40-110%. A great increase in fuel conversion efficiency can be obtained by connecting a waste heat utilization system to the plant, if there is need for space heating or process steam in the vicinity of the plant.

Additionally, power plants based on reciprocating engines can begin power production in a rather short time compared to large steam power plants. They can be built step by step following the most probable increase in power demand. High non-productive investments can thus be avoided. With several engines, the energy production can be more economically adapted to varying energy demand than with one major unit [Wiese 1996]. The diesel and gas engine power stations also suit regional power production well without a need for large networks for electricity transmission.

Modern reciprocating engines can burn different fuel alternatives. In addition to traditional oil products - diesel and heavy fuel oil - natural and liquid petroleum gas can be used. More advanced alternatives are other gases, alcohols, and vegetable oils. Until now, one of the main disadvantages of the engine power plants has been that solid fuels can not be utilized.

O'Keefe [1995] states that several new developments, and even concepts, along with wider appreciation of the inherent advantages of the engine, have given the reciprocating IC engine a new lease of life. The author sees advanced small-scale gas turbines (GTs) as formidable competitors, but adds that fast startup is a plus for engines, as well as the diesel's ability to burn heavy fuel oil more effectively, with less impact on the prime mover.

When gas is used as fuel, the competition becomes harder. Pure gases cause no problems in gas turbine use. Just small-scale gas turbines are seen as the main competitors for gas diesel engines [Westergren 1989]. The higher efficiency, particularly even at part load, and the higher portion of electricity in the combined heat and power (CHP) plants still remain the important advantages of reciprocating IC engines. One additional factor is that the efficiency of the engines is not as sensitive to ambient conditions as that of the gas turbines [Wiese 1996].

Prussnat [1996] concludes the comparison of the engine and gas turbine driven CHP plants as follows:

* Gas engines are suitable when thermal energy is primarily required in the form of warm water. In steam or hot water production, gas turbines are preferred.

* Gas turbines are not very suitable for daily starts and stops and are thus well adapted for base load operation. In contrast, gas engines are very suitable even for daily production breaks.

* The initial costs of gas engine plants are higher, as commonly are the maintenance costs. However, the electric efficiency of gas engines is clearly higher.

* High prices of electricity speak for the engines, low prices for the gas turbines.

Not all the experiences from the engine driven plants are, however, so positive. A recent paper [Kaivola 1996] compares experiences collected from a company's different small power plants. Economy, high availability, and ease of remote control are mentioned as advantages of a 45 MW_e/60 MW_t natural gas driven gas turbine CHP plant in Finland. The company has not as good operational experiences from its 6 MW_e/6 MW_t diesel plant based on a natural gas driven gas diesel (GD). The only positive point mentioned is standard components. As disadvantages of both the plants, poor durability of materials, and emissions are listed. The availability of the GD plant has also been unsatisfactory, and there have been oil leaks in the plant.

1.2 Markets for diesel and gas engine power plants

The gas and diesel engine power plant market has grown strongly, particularly during the '90s. Fig. 1. shows how the orders for reciprocating engine generators have developed [Wadman 1996]. Strong and increasing activity was recorded for the fifth year in a row in orders for power generation systems of one megawatt output and above. Additionally, a major engine manufacturer in the high-volume 1 to 2 MW category was unable to participate this year, so it can be estimated that total orders for reciprocating engines were in reality at the 5000 unit level.

A significant increase in orders for standby power equipment was observed, which is consistent with the growth reported in the smaller 1 to 2 MW output category [Wadman 1996]. North America and Western Europe remain major markets in terms of total units. Compared to last year, the Far East showed a big increase in orders, whereas Southeast Asia showed a significant decrease, along with Central America and the Caribbean.

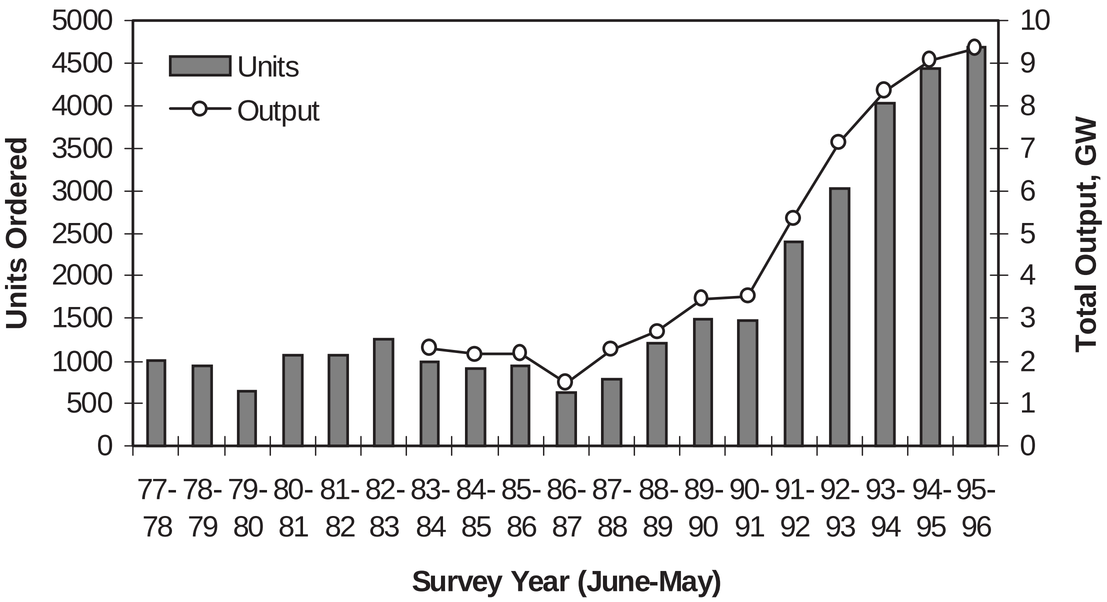

Figure 1. Diesel, gas and dual-fuel engine order trends in power generation market. Reprinted from [Wadman 1996].

In industrialized countries, combined heat and power (CHP) production is strongly preferred. In Germany, e.g., there were 1577 engine driven CHP plants in use in 1994, having a total capacity of about 872 MW_e [Rumpel 1996]. Pischinger [1995] reports that in 1993, the engine driven plants formed a proportion of ca. 3% of the total installed public electric capacity in the so-called old states of Germany (previous West-Germany). In the Netherlands, the share was 16% at the same time. In both countries, the number of this kind of power plants was increasing very fast.

Even in the new German states (previous East-Germany), the number of engine driven CHP plants has strongly increased [List 1996]. At the end of 1994, there were 88 such plants with an installed output of around 150 MW. The average size, about 1.35 MW, was thus clearly higher than in the western states, since the power plants have usually been built to serve as district heating plants. About 90% of the eastern CHP plants use natural gas as fuel, whereas in the old states, the share of this fuel is ca. 50% [Pischinger 1995].

Fig. 2 shows how the number of reciprocating IC engine driven CHP plants has grown in Germany [Rumpel 1996]. In 1994, 24 of these power stations had an output of more than 5 MW, whereas there were 169 plants in an output range of 1-5 MW. In Germany, engine driven CHP plants are seen as suitable, not only for local authorities, but also for regional energy companies [Juptner 1996]. In [Wiese 1996], there is a rather thorough feasibility study of different CHP plants planned in Germany. Power plants driven with diesel and gas engines and gas turbines are compared, as well as combined gas and steam turbine plants. The gas engine driven CHP plant proves to be the most profitable up to an installed output of 15 MW under these German conditions.

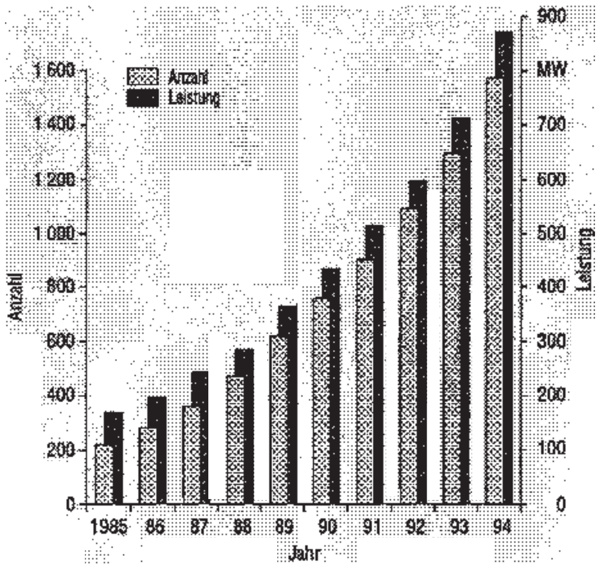

Figure 2. Number and output of engine driven CHP plants in Germany [Rumpel 1996].

1.3 Marine electric systems based on diesel and gas engines

The marine auxiliary generating unit survey and the newly separated diesel-electric marine propulsion data indicate an excellent growth pattern as well, Fig. 3 [Wadman 1996]. With a total output of 792 MW in the period from June, 1995, to May, 1996, diesel-electric marine propulsion has become a good market for engines.

A new example of the competitiveness of gas engines is that two of the eight diesel generating sets of the production platform Veslefrikk B have been converted to gas diesel operation [Kvist 1996]. Recently, gas diesel engines have also been installed in a major ship [Wärtsilä... 1995]. The four 5.92 MW gas diesel engines on the Smedvig tanker will run at 720 r/min and form the basis of a diesel-electric power plant. Wärtsilä claims that gas diesel technology enables efficiencies of more than 45% to be obtained; this is said to be 10% higher than a similar installation powered by gas turbines. (Most probably: 10 percentage points; author's note.)

This marine gas engine installation, as well as the other examples given in [Wärtsilä... 1995], confirm that there can be a growing market for diesel-electric propulsion and maybe even for gas driven engines in the marine sector.

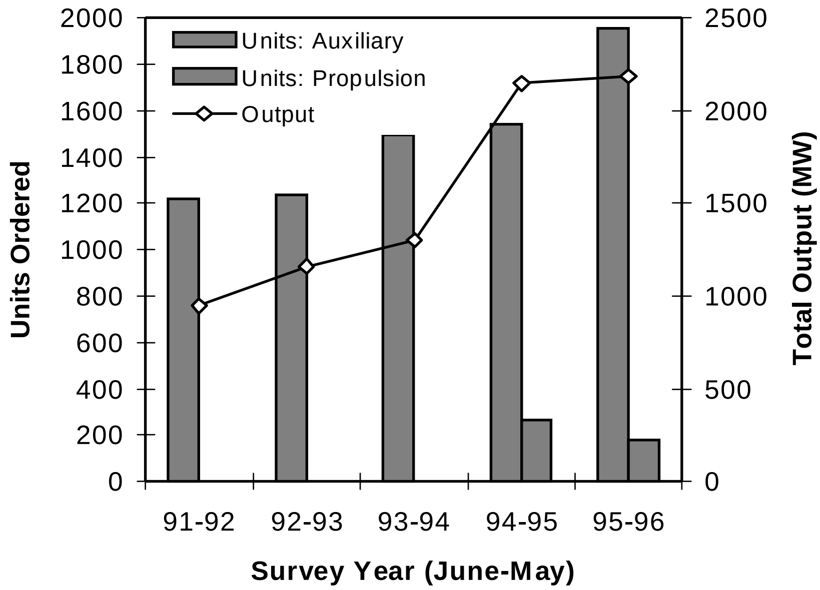

Figure 3. Marine auxiliary and diesel-electric propulsion order trends. Reprinted from [Wadman 1996].

1.4 The aim and structure of the survey

Accordingly, there is growing interest in diesel and gas engine driven power plants and these plants form a potential alternative for the increasing power production sector. The aim of this paper is to survey the latest technology of power plants driven by reciprocating IC engines. Recent articles are reviewed concerning such power generating systems. The main issues are the fuels, engines, and the power plant schemes used in modern, recently built power stations. Efficiencies obtained and to be reached with different concepts are also dealt with. Emissions regulations and emissions control technologies are reported, as well. The review has been limited to papers published in the late '80s and during the '90s.

Finally, the reader should remember that "the many large-scale, innovative ideas and developments under way should not overshadow the thousands of small engine/generator units producing efficient and reliable power today" [O'Keefe 1995].

2 FUELS

2.1 Fuel oils

Continually, most diesel power plants operate with diesel (DFO) or heavy fuel oil (HFO). DFO is very common in an output category of 1 to 2 MW, whereas engines of above 5 MW usually operate with HFO. For instance, the two-stroke engines of the Coloane Power Station in Macau are driven with HFO and are able to burn residual fuels up to 450 cSt at 50°C [Cordeiro 1996]. It is rather common that the power plant engines are able to burn fuel with a viscosity of 700 cSt [e.g., Mullins 1995]. O'Keefe [1995] cautions, however, that a surge in demand for HFO could easily lift prices above the recent lows. Better acceptance by engine owners and advances in petroleum process technology could cause such surge.

Another issue is to keep maintenance costs under control. In a Canadian diesel power station, fuel costs have been reduced by using HFOs rather than distillate [Diesel... 1996]. Use of 100 cSt viscosity fuel produces significant savings, and further decrease in fuel costs will be achieved by using heavier IF-180 fuel. The user would progress to 380 cSt fuel, if he only could be confident of keeping maintenance costs under control, however.

On the contrary, use of gas oil as the main fuel instead of HFO is being investigated in the diesel engine driven CHP plant at the University of Nottingham in the U.K. [Pickering 1994]. There may be an argument for burning this in preference to HFO, due to reduced parasitic losses (more heat recovery) and significantly less maintenance on the engine, centrifuges etc., and probably simpler, more automatic operation. During the planning phase of the plant, the HFO driven diesel engine was predicted to be the most cost effective, practical and energy efficient. At that time, British Gas was not interested in giving competitive gas contract prices, which ruled out a dual-fuel engine. The University also had already an extensive HFO storage facilities.

2.2 Alternative fuels

Different fuel alternatives have, indeed, been studied during recent years in order to enlarge the fuel assortment of the diesel power stations and hence their markets, to reduce the dependence on the oil resources, and to adapt diesel plants to more stringent environmental restrictions.

2.2.1 Gases

Natural gas is evidently the most potential alternative. Diesel, gas and dual-fuel engine orders from June, 1995, to May, 1996, include 423 engines burning natural gas as fuel [Wadman 1996]. Most of them are in an output category of 1 to 2 MW, but some greater natural gas engines were also ordered. In addition, the listed 31 dual-fuel engine orders certainly include engines using natural gas as main fuel.

Prussnat [1996] and Rumpel [1996] list liquid petroleum gas, gas from sewage disposal plant, landfill gas, other biogases, and wood gas as other gaseous alternatives, most of which are being seriously studied.

Wärtsilä Diesel has developed engines having multi-fuel capabilities [Paro 1995]. The original idea was that all floating oil production vessels could generate their own energy from whatever comes out from the well, be it gas or crude oil. Multi-fuel concept has later proved to be suitable for stationary power plants, as well.

For these multi-fuel engines, Wärtsilä selected a pure diesel combustion process where air is compressed and all the fuel energy is injected close to the top dead center, the main part in the form of gas introduced at high pressure [Paro 1995]. A pilot injection of liquid fuel is used as an ignition aid, the pilot amount corresponding to three (3) per cent of the full load energy consumption. The gas diesel process proved to be quite tolerant when it comes to gas composition, which, as far as natural gas is concerned, can vary widely from high methane content to a mixture of methane with heavier gases. A longer lifetime of engine components has been mentioned as an additional advantage of a gas-driven engine [Paro 1995], [Kvist 1996].

Surplus wellhead gas is also exploited at one of the onshore oil-fields in the U.K. [Mullins 1996a]. Three spark-ignited Caterpillar gas engines produce altogether about 3 MW electricity at a speed of 1500 rpm. The gas contains 80% methane, along with propane and other gases. Oil and condensates are removed by means of filters.

The diesel combined heat and power plant MHKW Resse in Gelsenkirchen, Germany, is a real multi-fuel diesel power station. The two engines use mining gas as main fuel, but they can also been driven with natural gas, diesel fuel oil, and with a mixture of these three fuels [Röchling 1995]. The engines are of dual-fuel type, so a small amount of DFO is always needed. In 1993, 87% of the total fuel energy consumed (214 GWh) was mining gas, 7% diesel fuel oil, and 6% natural gas. The engines have been designed for gases having a minimum methane number of 78. The mining gas complies well with this requirement, but the methane number of natural gas can sometimes fall down to 70. A comprehensive control system has thus been designed for the plant in order to be able to run the engines even on natural gas in dual-fuel operation.

Landfill gas is one fuel alternative that has been successfully utilized especially in small-scale regional energy production. One of the largest landfill gas exploiting gas engine plant is the Calvert power station 10 km south of Buckingham in England [Mullins 1994]. Up to 2200 m^3/h of gas are being drawn via 37 gas wells at a pressure of 30 mbar using three pairs of fans. The moisture content of the gas is reduced, and the gas for the power plant is compressed by means of two-stage compressors. The chilled gas is then piped to the engines at a pressure of 6.3 bar. According to O'Keefe [1995], the landfill gas varies considerably in composition, at least in the case reported.

Low calorific value (LCV) gases interest engine manufacturers, as well. Energy... [1996] reports on a chemical company in Austria which exploits waste gas of the production for generating energy by means of a gas engine CHP plant. The gas contains 74 to 78% nitrogen, 16 to 21% hydrogen, 3.5 to 5.5% CO_2, and 0.5 to 1.5% CO. The lower heating value ranges from 0.5 to 0.6 kWh/m^3, being thus only about 1/20 of that of natural gas. The gas is free of charge, thus shortening the amortization time of the plant.

2.2.2 Liquid alternatives

A very special alternative fuel is fusel oil. A German engine-driven CHP plant burns a fuel mixture consisting of 30% fusel oil and 70% diesel fuel [Drunk... 1996]. The aim is to increase the fusel oil portion to 40%. Fusel oil is generated during rectification of raw alcohol. It contains higher alcohols and is toxic. In addition, there is a lot of water in fusel oil.

One interesting alternative fuel even for diesel power applications is pyrolysis oil. The economic prospects for pyrolysis oil look promising in the long term, but near term applications are difficult to find due to the present low prices of fossil fuels [Solantausta 1994]. The production of heat and power in a stationary diesel engine at sawmills is seen as a possible niche market for a plant burning pyrolysis oil as fuel.

The pyrolysis fuel oils has some unique properties: high water and oxygen content makes them difficult to ignite; their possible instability will lead to an increase in fuel viscosity; the acidity could affect materials and maintenance intervals, level of emissions, and need for special catalytic cleaning [Solantausta 1994].

Preliminary tests indicate, however, that the pyrolysis oil burns readily once ignited [Solantausta 1994]. Nevertheless, a pilot fuel engine is suggested to be used to overcome the poor ignition properties of the oil. The authors assume that emissions should not be significantly greater than for reference fuels when the oil is burned in medium-speed, pilot-fuel engines. They emphasize that much further research is needed, however.

2.2.3 Coal

At times, tests and development work have been made to burn coal in internal combustion (IC) engines [O'Keefe 1995]. The work has taken impetus from oil shortages or increase in oil price relative to coal. One new effort is that of an American consortium, the study of which began in 1985 and continues. The work aims at a plant of 10-100 MW output. The first fuel tested was a 50/50 coal-water slurry of 12-micron bituminous coal with 2% ash and 36% volatiles. Medium-speed trunk-piston V-engines have been used, with a moderate brake mean effective pressure of ca. 14 bar and a speed of 400 rpm.

For the cylinder-wall zone, injectors and exhaust valves, ceramic coatings are under test. In trials, a new chrome-carbide coating is reported effective. Encouraging combustion results are also reported from four-day tests run on a mine coal, with 2.7% ash and 44% volatiles [O'Keefe 1995].

3 ENGINES

In base load diesel power plants, both low-speed two-stroke and medium-speed four-stroke diesel engines have been traditionally used. In recent years, the percentage of different gas burning engines has increased. Effort has been put into further development of traditional spark-ignition (SI) or dual-fuel (DF) gas engines, and new concepts have been designed, called gas diesels (GD).

Smaller high-speed engines are also widely used for power generation. For example, the Belgian Railway Authority was installing a 6.0 MVA peak-shaving diesel power station at its Brussels-Schaarbeek feeding point [Belgian... 1994]. The plant comprises four gen-sets powered by 12-cylinder turbocharged Dorman engines. This kind of engine approaches heavy-duty truck engines with their air-to-air intercoolers and unit injectors. Smaller high-speed engines are popular in landfill gas operations, as well, because of their relatively low cost [Mullins 1994]. Rather small engines are very widely used even in local energy production in Germany [Pischinger 1995, Rumpel 1996, Röchling 1995].

The papers referred to do not give accurate information on what the market share is of different types of power plant engines. Chellini [1995a] gives, however, one interesting detail. GEC Alsthom, France, has manufactured more than 320 generators rated over 10 MVA for diesel engines. Of these, 23 are driven by two-stroke diesels and 300 by medium-speed, four-stroke engines.

Another indicator is given in Fig. 4, which presents operating speed ranges of the engines with 1 MW output and above, ordered for power generation systems during a period from

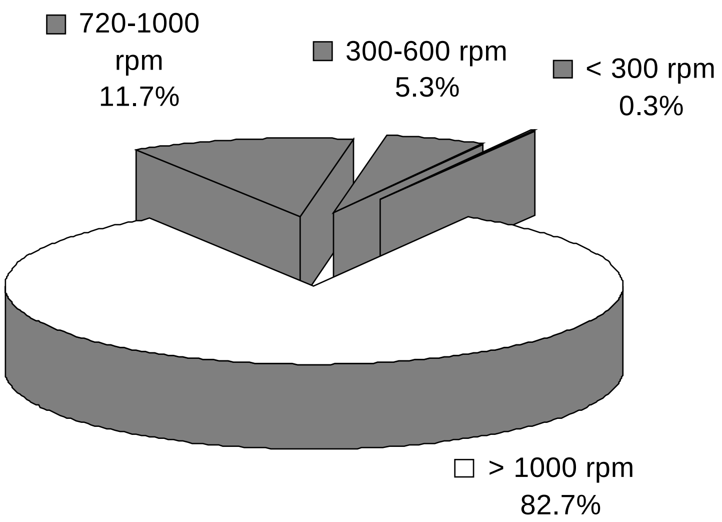

Figure 4. Operating speed ranges of diesel, gas and dual-fuel power generation engine orders in the period from June, 1995 to May, 1996. Reprinted from [Wadman 1996].

June 1, 1995 through May 31, 1996 [Wadman 1996]. One can see that a major part of the engines were high-speed ones, and few two-stroke slow-speed engines were ordered. Four of the engines with a speed of under 300 rpm were in an output category of 30 MW and above, and eight in a category of 10 to 15 MW.

3.1 Oil and liquid mixture burning engines

3.1.1 Low-speed engines

Two new diesel units of the Macau power plant are said to be the largest stationary two-stroke low-speed diesels built to date [Cordeiro 1996]. These Mitsui MAN B&W model 12K90MC-S engines have a capacity of more than 50 MW each. They entered commercial operation in May 1995 and February 1996, respectively. The power plant as a whole includes six two-stroke engines of MC design from MAN B&W. The diesel engine characteristics are given in Table 1. The two-stroke units have been built in three phases, two of them starting commercial operation in 1987 and 1988, two in 1991 and 1992, and the latest in 1995 and 1996.

Table 1. Diesel engine characteristics of the Coloane diesel power station in Macau [Cordeiro 1996].

Engine model	Output	Speed	No. of cyl.	Bore	Stroke
	MW	r/min		mm	mm
9K80MC-S	24	100	9	800	2300
12K80MC-S	37	100	12	800	2300
12K90MC-S	50	103.4	12	900	2300

The authors report that an economic evaluation and the good results achieved with the first extension in the mid-1980s led the company to repeat the low-speed solution in the second extension phase. A reduction in heavy fuel oil consumption in addition to other factors is said to compensate for the larger initial costs of the low-speed diesel units. Adjusted for inflation, the company’s tariff rate in 1995 was 58% of what it was in 1986 [Cordeiro 1996].

In the new Vale D power station in Guernsey, the setup comprises a nine-cylinder 9RTA58 two-stroke engine from New Sulzer Diesel Ltd, driving a 15 MW, low-speed alternator. The speed is 136 r/min [Mullins 1995]. The power station is designed eventually to house two power units of 15 MW each. Guernsey Electricity has long experience with Sulzer engines, since the older Vale C power plant houses four earlier-model R-type engines, the first three being valveless and the last one valved.

Chellini [1995] reports that a low-speed two-stroke New Sulzer Diesel type 12 RTA84C is being installed at the EDF-Pointe des Carrières power station in Martinique. The engine rating is 45.84 MW at 100 r/min.

3.1.2 Medium- and high-speed engines

On Iles de la Madeleine in Canada, the diesel power station is based on six medium-speed Sulzer 16ZA40S diesel engines, each producing 11.52 MW at 514 rpm [Diesel... 1996]. Since the utility has total responsibility for producing power 24 hours a day and 365 days a year, the diesel plant is capable of supplying approximately twice the island's winter electrical requirement. The company can run up to four engines to handle peak loads, and keep one in reserve and one for overhaul. The plant began operations four years ago [Diesel... 1996], and it is nowadays driven with medium-heavy HFO.

In the U.K., a diesel engine CHP plant was commissioned in 1989 at the University of Nottingham on the main campus [Pickering 1994]. The plant is based on a Mirrlees Blackstone eight cylinder medium-speed diesel engine running on heavy fuel oil. The plant generates 3.7 MW of electricity at 11 kV and meets the base electricity load of the campus. The rated speed of the engine is 600 rpm and rated output 3937 kW. HFO is supplied to the station from the existing central boiler house system. The engine is started and stopped with gas oil. In order to isolate vibration, the engine and generator are mounted together on a steel base plate that is supported on antivibration mountings on a concrete block, independent of the building foundations.

There have been some engine related problems in the plant [Pickering 1994]. The original turbocharger has been replaced with one with a greater surge margin. The first turbocharger required daily water wash cleaning at a reduced load (exhaust temperature of 400°C) to maintain sufficient performance to prevent surging. This continual cycling led to damage of turbine components through thermal shock. To reduce the thermal shock, the wet wash was replaced by a dry washing technique using blown nut kernels. After a few months the turbocharger fouled up, however, and it had to be completely stripped for cleaning. At this point the turbocharger was changed and a new cleaning procedure introduced. Since then, the revised wet washing procedure has been carried out twice a week at an exhaust temperature of 250°C, with a daily short dry wash. Over six months of trouble free running had been achieved with no turbocharger/surging outages at the time of the paper.

There was also a series of other problems with the plant, many of them related to vibration and high engine temperatures, e.g., insufficient engine cooling and high engine vibration causing failure of various components bolted to the engine [Pickering 1994]. It was found that more reliable performance could be achieved by operating the engine at a reduced load. Since July 1992 the engine has run at around 3 MW_e and much more reliable operation has been obtained. As experience is gained and problems overcome, the authors assume that load may be increased again to around 3.6 MW_e. A fuel additive is also being considered to improve the performance of the oil treatment plant. Use of a cleaner fuel could result in reduced maintenance of the engine, concerning particularly exhaust valve cage life and turbocharger cleaning.

In Wilthen, Germany, an MAN engine burns a mixture of DFO and fusel oil as fuel [Drunk... 1996]. Up to now, the engine designers have been able to increase the percentage of fusel oil to 30%, but the target is 40%. The plant output is 843 kW. Gaskets, fuel lines and exhaust passages had to be changed when the diesel engine was modified to run on the mixture containing, e.g., lots of water. Combustion should also been investigated.

3.2 Gas engines

3.2.1 Engine concepts

A commonly used definition of different gas engine concepts has been presented in Ref. [Goto 1995]:

- Spark ignition (SI) engines burning stoichiometric or lean gas-air mixtures. Homogeneous or stratified charge.

- Dual fuel (DF) engines. The lean, homogeneous mixture is ignited by a small amount of pilot fuel injected directly into the combustion chamber just before the top dead center (TDC).

- Gas injection (GI) engines or gas diesels (GD). The high-pressure gas is injected directly into the combustion chamber at the end of the compression stroke. The mixture is ignited by pilot fuel or by means of glow plugs.

Fig. 5 compares the brake mean effective pressures (BMEP) obtainable in different gas engines and in a diesel engine, as far as medium-sized medium-speed engines are concerned [Goto 1995]. It should be noted that even higher BMEPs have been achieved particularly in large medium-speed diesel engines. Pischinger [1995] gives a BMEP level of 16 bar for highly-turbocharged lean-burn gas engines, which is also a slightly higher value than in Fig. 5.

In the GD cycle, the oxygen content of the exhaust is slightly higher than in other gas engines according to O'Keefe [1995]. It is reported that the difference is sufficient to avoid the addition of supplementary air or oxygen in duct burners and down-stream boilers in combined-cycle operation.

Properties and suitability of different engine concepts have been more thoroughly reported by Niemi [1995]. In this chapter, only power plant engine concepts and

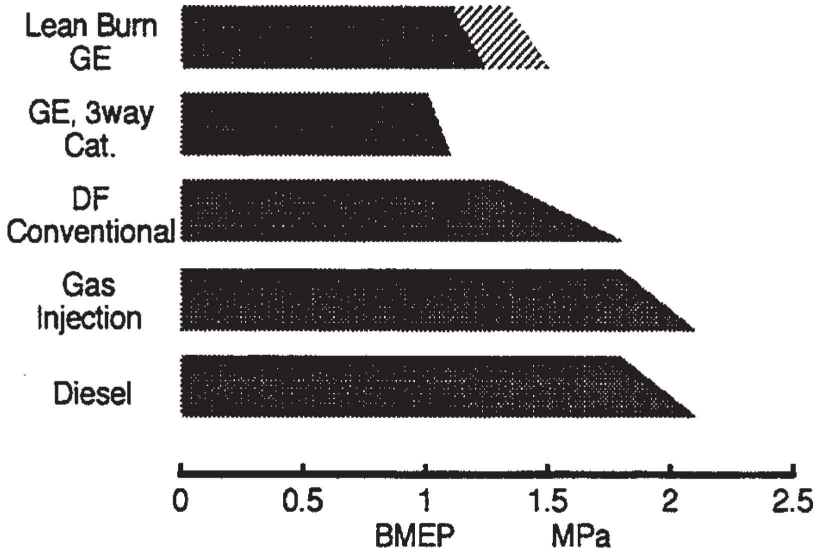

Figure 5. Comparison of BMEPs of different gas and diesel engines [Goto 1995].

experiences obtained from them are reviewed. Additionally, some features of new gas engines reported after the publication of [Niemi 1995] are presented.

3.2.2 German and British applications

Prussnat [1996] states that in a load range of 50 kW to 3.6 MW, SI gas engines are the main solution in Germany. DF and GD engines dominate an upper load range of 2 to 7 MW. Use of a diesel engine is a rather rare solution.

Two medium-speed 12-cylinder 12 PC 2-5 V400DFC dual-fuel gas engines form the core of the combined heat and power plant (CHP) of MHKW Resse in Gelsenkirchen, Germany [Röchling 1995]. The engines were taken into use in 1989. The main fuel, mining gas, is mixed with the intake air before the compressor of the turbocharger. When required, natural gas is fed into the intake manifold as a supplementary fuel. Natural gas is supplied to the plant at a gage pressure of < 4 bar, since the gas is mixed with compressed air without any further gas compression [Röchling 1995].

The original diesel engines were slightly modified for dual-fuel gas operation. The valve overlap was decreased to 40° in order to reduce gas leakage during the scavenging. The compression ratio was changed from 11.5 to 10.4 by modifying the piston crown and thickening the cylinder head gasket. The inlet manifold was reduced, strengthened, and equipped with explosion valves. The injection pumps were calibrated for an injection quantity of about 5% of full capacity. Sealing between the compressor and turbine was enhanced. Leaked compressed gas is led to the compressor inlet. A special engine control system was designed in order to make it possible to change over smoothly from one fuel to another in a load range of 35 to 100%. An extremely fast microprocessor-based control system controls actuators of mining gas, natural gas, air by-pass, and DFO settings [Röchling 1995].

Large effort was put into the development of the control system in order to make it possible to run the engines as economically as possible. This means that the individual cylinders must operate as close to the knocking limit as possible. Comprehensive measurements were first made to find out factors affecting the knocking phenomena. Mixture temperature in the intake manifold proved to be the main factor. Reduction of the temperature resulted, however, in ignition deterioration in natural gas operation due to the too lean air-fuel mixture. It was also noticed that the knocking sensitivity of individual cylinders was different, although the air-fuel ratios and mixture temperatures were very similar. The condition of the injection nozzle was also found to be an important factor affecting combustion. Mixture formation and cooling of the injector tip are problematic due to the small injection quantities of the pilot fuel [Röchling 1995] - Figure 6 shows the peak pressure variation cycle-by-cycle and between individual cylinders in different operation modes.

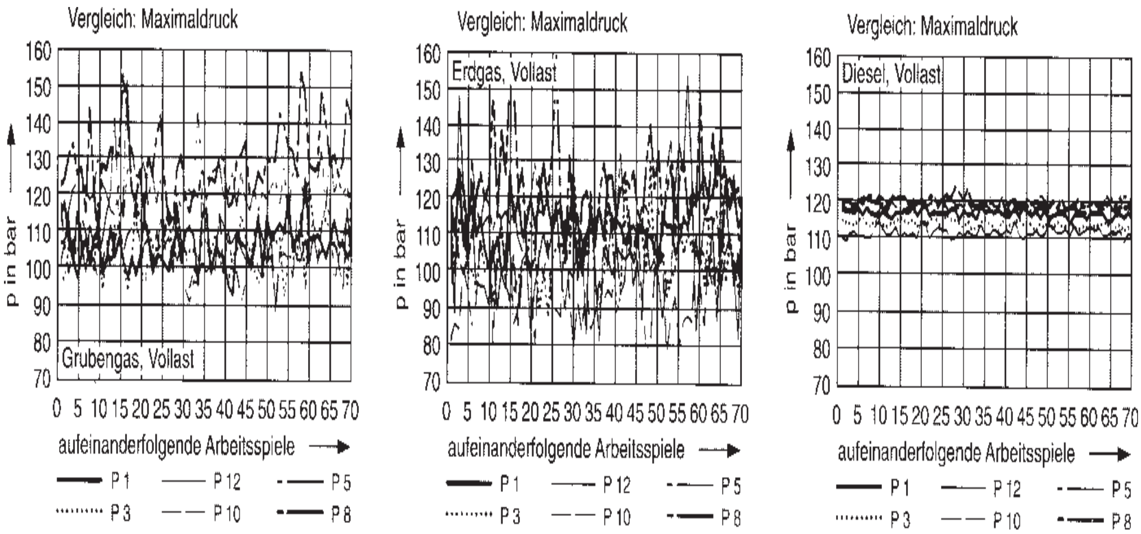

Figure 6. Peak pressure variation cycle-by-cycle and between individual cylinders in the dual-fuel gas engines of the CHP plant MHKW Resse [Röchling 1995].

As a result of the basic tests, it was concluded that knocking can be detected by means of individual knock sensors and cylinder pressure signals. Knocking should be watched within a crank angle range of 5 to 40° and analyzed in a frequency range of 2 to 10 kHz [Röchling 1995].

In the case of knocking, the mixture is first made weaker - to a λ value of 1.9 with natural gas and 1.85 with mining gas. If this does not help, the mixture temperature is reduced. Output reduction is the third step [Röchling 1995].

The CHP plant Haar in Germany comprises three 16-cylinder lean-burn spark-ignited gas engines, having a gross electric output of 1.29 MW each [Juptner 1996]. Natural gas is the main fuel of the plant. The medium-speed SI gas engine Ruston RK270GS has proved its reliability in landfill gas operation in Calvert power station in the U.K. [Mullins 1994]. According to Ref., the engine has been running for just under 18.000 h and has been operational for 94.9% of its possible running time. The output of the engine is 2.8 MW at 1000 r/min.

3.2.3 Engine development

O'Keefe [1995] reviews the dual-fuel engine development work made by Fairbanks Morse. Gas fuel is now introduced into the air-intake elbow, Fig. 7. Pilot fuel consumption was reduced to 1% using a prechamber. Prechamber cooling has been improved so that low-carbon steel can be used for construction, rather than the customary high-temperature alloys. The unusually low pilot quantity called for two injection means, with an electrohydraulic valve to control the two injectors.

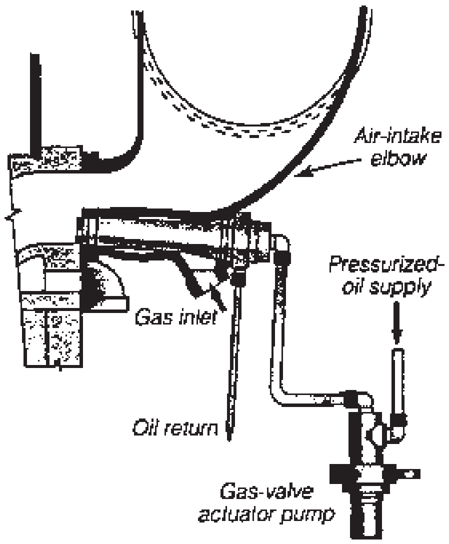

Figure 7. Gas valve in air-intake elbow [O´Keefe 1995].

Fincantieri has developed one of the largest SI gas engine type which is said to be particularly well-suited for co-generation plants [Chellini 1995a]. The bore of the engine is 320 mm and the stroke 390 mm. With natural gas of 100 methane number, the engine produces 360 kW/cyl at a speed of 750 rpm. The brake thermal efficiency of this lean-burn engine is said to be 41%.

The gas engine was developed in parallel with the A32 diesel engine program [Chellini 1995a]. The gas engine design is very similar to that of the diesel version. Minor modifications have been made to the cylinder heads, and a spark-ignition system has been added. A two-stage intercooler has been incorporated into the gas engine.

The cylinder head incorporates a small prechamber, Fig. 8 [Chellini 1995a]. A small quantity of rich air/fuel mixture is ignited in the prechamber through an electronically-controlled spark plug. This small charge - with its high ignition capability - expands into the main chamber, and ensures safe flame propagation to the very lean charge in the main chamber. The lean mixture containing 100% excess air with respect to the stoichiometric ratio burns at a low temperature. The pressure head of the fuel gas is kept constant in relation to the turbocharger boost pressure in order to obtain a constant air/fuel ratio in the charge.

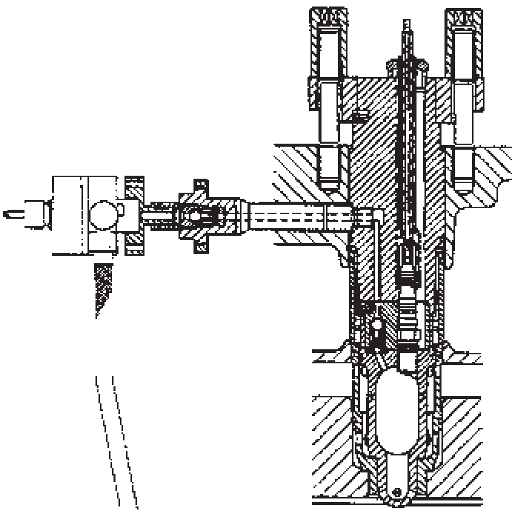

Figure 8. The prechamber ignition and injection systems for the GMT A32G gas engine [Chellini 1995a].

For small gas engine power plants, Wärtsilä Diesel has brought a new series of SI gas engines onto the market [Kinnunen 1996, Kunberger 1996]. The smallest unit with an output of 1 MW is a high-speed engine without any prechamber. In this engine, gas is exceptionally injected into the suction side of the turbocharger compressor. The engine uses two venturi-type carburetors. A gage pressure of 0.15 bar is sufficient for the gas supply.

Units of 2, 3, 4.5 and 5.5 MW are based on medium-speed engines utilizing prechambers [Kinnunen 1996, Kunberger 1996]. As an example, the Wärtsilä 28SG engine, launched in 1996 and used for the 4 and 4.5 MW plants, has turbochargers with wastegates and receiver throttle valve. The engine is designed to operate with jacket cooling water up to 110°C to satisfy different cogeneration requirements. The compression ratio is 12 and the BMEP 16 bar.

A common feature for the Wärtsilä SG engine range is a fully processor-controlled engine management system [Kunberger 1996]. It includes a knock detection system, monitors the combustion quality and controls the air/fuel ratio and the ignition continuously in each cylinder. The system makes it possible to operate the engine near the knocking limit even at varying gas qualities.

MAN Dezentrale Energiesysteme GmbH (MDE) of Augsburg, Germany, offer small-size high-speed spark-ignited gas engines for cogeneration [Lean-Burn... 1995]. Plants driven with such engines are said to be very popular in Germany today. MDE states that their systems can operate at one-half of the German Clean Air Legislation (TA-Luft) requirements. The engines use either λ=1 fuel-air control and a three-way catalyst or a lean-burn concept equipped with an oxidation catalyst.

To achieve a constant air-gas ratio at different engine loads, MDE utilizes venturi-type air/gas mixers in combination with a gas pressure regulator [Lean-Burn... 1995]. The intake manifold of the engine is designed to provide a homogeneous mixture. The micro-circuit-based digital ignition system uses the high-energy, capacitor discharge principle.

Two MDE turbocharged lean-burn engines are said to be capable of operation on natural gas down to a methane number of 70 without any restriction [Lean-Burn... 1995]. By varying the charge cooling temperatures, the engines can run even on lower methane number gas without any limit in power. The MDE naturally-aspirated stoichiometric engine models with a compression ratio of 12.5 require gas fuel with a methane number of 80 or better to provide continuous full load. Engine operation is possible even with gases having a methane number of slightly below 80, but the ignition timing should be adjusted and the engine output must be slightly decreased. MDE provides these engines with a lower compression ratio of 10 for gas fuels with a methane number down to 34 (e.g., propane). The engines are also offered for sewage and landfill gas down to a lower calorific value of 4 kWh/m^3 (0°C, 101.3 kPa).

The antiknock control and monitoring system of MDE consists of knock sensors, a control device for ignition timing and power reduction, and an ignition system that provides timing control with a current signal [Lean-Burn... 1995]. If the methane number of the fuel decreases, the system reacts in the following steps: determination of knocking combustion, retardation of spark timing, reduction of engine power, and cut off of the engine under continuous knocking. After monitoring normal combustion, the system will readjust spark timing and power output to their normal values.

The Austrian company Jenbacher Energiesysteme is offering high-speed, four-stroke, lean-burn and stoichiometric SI gas engines with air/gas supercharging [Cogeneration... 1995]. The electric power output per module ranges from 280 kW to 1.55 MW with gas engines running at 1500 rpm. Gas and combustion air are mixed by a turbocharger in the company's Leanox lean-burn technology.

The same company has also done intensive development work to make their engines able to run even on a low calorific value (LCV) gas having a lower heating value of only 1/20 of that of natural gas [Energy... 1996]. Before entering the cylinders, the gas is compressed by the turbochargers up to a pressure of 3.7 times that in the pipeline. The burning components of the gas are hydrogen, the share of which varies from16 to 21%, and carbon monoxide that the gas contains only 0.5 to 1.5%. The intake air is not compressed. The electrodes of the spark plugs are made of platinum alloy to ensure strong enough sparks. The 20-cylinder engine version produces an electric output of 1.05 MW with natural gas, but with LCV gas, it only reaches 588 kW.

4 POWER PLANT CONCEPTS

One of the most interesting issues of diesel power plant development is the exploitation of the waste heat generated by the engine and charge air cooling and in the exhaust gas boiler. In marine installations, waste heat of main and auxiliary engines is traditionally utilized for heating, evaporation and cleaning purposes. Sometimes steam produced in the boiler has been fed into steam turbines to increase electricity production.

In regions with a cold climate, the energy production is largely based on combined heat and power (CHP) plants. Waste heat of the electricity production is used for district heating. A small share of these CHP plants have diesel engines as prime movers. Waste heat recovery is rather simple to arrange effectively, and the total efficiencies are high. Similar concepts are also adopted for industrial applications, waste heat now being utilized for drying, heating, cooling, and other process purposes.

In pure electricity production, demand for higher fuel conversion efficiencies has forced different combined cycle solutions to be found even in the diesel power sector. During recent ten years, turbo compound systems (TCS) have been largely adopted particularly in low-speed two-stroke engine installations. Most recently, steam turbine units have also been incorporated into diesel power plants to exploit waste heat of the engines in the form of superheated steam and to increase the electric output of the plants.

4.1 Plants with pure electricity production

4.1.1 Turbo compound systems (TCS) and unfired steam power concepts

In the Coloane Power Station in Macau, waste heat from the exhaust gas of the diesel engines is used for power production just by means of turbo compound systems and by steam turbine generators driven by the steam produced in exhaust gas boilers [Cordeiro 1996].

Due to the high efficiency of the turbochargers, part of the exhaust gas can bypass the turbochargers and be used in a small power turbine coupled to an asynchronous generator, which feeds electricity into the grid through a transformer. In order to intensify the heat recovery, saturated and superheated steam is produced in exhaust gas boilers. Saturated steam is utilized for heating fuel oil, etc. The superheated steam is used for power production in a steam turbine that drives an asynchronous generator. Each diesel unit of the plant has its own TCS and exhaust boiler. The steam turbine units are common for each pair of diesel units. In a distiller plant, waste heat of the cooling water of two engine units is additionally used to produce fresh water from seawater for boiler make-up water [Cordeiro 1996].

In Table 2, engine, TCS, and steam turbine outputs of each pair of engine installations of the Coloane Power Station are presented. One can see that rather small additional power outputs (4.5 to 5.5%) can be obtained with TCSs and waste heat steam turbines. Nevertheless, the improved exploitation of raw energy is said to obviously compensate for the increased initial costs of the plant [Cordeiro 1996].

Table 2. Energy recovery of the Coloane Power Station [1996].

Units	Engines	Output of Engines	Turbo Compound System	Steam Turbine	Additional Output, Total	Additional Output, % of Total Output
		MW	kW	kW	kW	
G03/G04	9K80MC-S	2 x 24	2 x 400	1400	2200	4.5
G05/G06	12K80MC-S	2 x 37	2 x 800	2600	4200	5.5
G07/G08	12K90MC-S	2 x 50	2 x 1000	3800	5800	5.5

A similar turbo compound system is incorporated into the two-stroke low-speed engine in the Vale D power station in Guernsey [Mullins 1995]. The Sulzer efficiency booster system comes into use at 40-50% engine load. The maximum output of the power turbine is 420 kW of electricity, the diesel generating set having an output of 15 MW. The TCS hence increases the electric output slightly less than 3%. Waste heat of the engine's exhaust is utilized by generating steam in a heat recovery boiler for the process of fuel treatment and fuel preheating.

Wärtsilä has tested a power turbine in connection with a gas diesel engine burning natural gas as main fuel and generating an electric output of 7.1 MW [Westergren 1989]. The output of the TCS was 375 kW when 17% of exhaust was led via the power turbine. The additional output was thus ca. 5% of the total electric output. The fuel consumption decreased by ca. 6 g/kWh in a load range of 80-110%. Simultaneously, exhaust temperatures increased, however, by 30 to 40°C at full load. The authors emphasize that a power turbine can be used only when the combustion chamber scavenging is sufficient. In principle, the TCS can be devised when the engine output exceeds 5 MW.

The gas-driven diesel power plant Ringgold in the USA comprises three Wärtsilä 18V32GD gas diesel sets, each having an electric output of 5.4 MW [Paro 1995]. The three exhaust gas boilers produce steam for a turbine aggregate, generating an additional output of 1.2 MW_e (1.4 MW [O'Keefe 1995]), which is around 7% of the total electric output.

As a conclusion, some 3 to 7% of the total electric power of the plant can possibly be produced utilizing the excess exhaust energy in a power turbine and waste exhaust and cooling heat for steam power production. One important limit is that only low quality steam can be generated if the exhaust boiler is unfired.

4.1.2 Other combined schemes

Different diesel combined technology (DCC) systems have been presented by Shelor [1994], including those for pure electricity production and for combined heat and power production. The authors state first that the exhaust temperature of the engines is quite low due to their high shaft efficiency. It is therefore necessary to increase the temperature of the exhaust gas when high quality steam is required for either process requirements or for the production of electricity. This is accomplished by the use of a special burner and boiler system which allows a portion of the engine exhaust gas to be directed through a burner and the rest of the gas to be led through special over fire air ports in the boiler.

With this unique boiler configuration, it seems that the burner design will allow 80% of the engine exhaust gas to bypass the burner provided while burner stability is maintained and sufficient NO_x dilution occurs to comply with other environmental requirements [Shelor 1994]. Effects of the bypass quantity of the engine exhaust gas have been thoroughly investigated. Results are shown in Table 3. The authors consider that it is probably not practical to try bypass flow rates higher than 80%, the limiting factor being the gas temperature required to make superheat [Shelor 1994].

To demonstrate the concepts, the authors present three plant examples for pure electricity production. In the first one, a diesel generating set of about 15 MW is running on No. 6 fuel oil. The boiler is fired with the same fuel. Combustion air is added to maintain 14.6 mass-% (wet) oxygen in the burner windbox. The burner is fired with a minimum excess oxygen of 10% at burner exit. This results in an approximate temperature of 1540°C leaving the burner. The windbox temperature is kept at around 295°C. Steam is generated in conditions of 90 bar/510°C. The ambient conditions are 30°C, sea level, 60% relative humidity [Shelor 1994].

The second example utilizes six medium-speed diesel engines along with a single steam turbine. At the lowest boiler entering temperatures (670°C), only 20% of the diesel exhaust passes through the burners and the rest passes through the over fire air ports. The boiler is a three-pressure boiler. The reheat steam turbine will operate at 102 bar/538°C/ 538°C. The turbine output will be either 40 MW or 70 MW depending on the quantity of bypass. The total output - diesels plus turbine - will thus be either 130 MW or 160 MW depending on the bypass level. In the latter case, 40% of the diesel exhaust gas is led to the burner [Shelor 1994].

In the third case, four medium-speed diesel engines generate 60 MW electricity. The engines are running on heavy fuel oil. A single boiler is designed to receive the exhaust from the four engines. In the boiler, either HFO or orimulsion can be used as fuel. In the former case, approximately 40% of diesel exhaust will be sent through the burners and 60% will be bypassed to the secondary air ports. If orimulsion is used as boiler fuel, about 30% of the exhaust gas is led through the burner. A higher level of supplementary outside

Table 3. Oil fired diesel combined cycle: typical operating parameters [Shelor 1994].

Portion of Diesel Exhaust Mass Flow to Burner		100 %	80 %	60 %	40 %	20 %
Portion of Total Boiler Gas Mass Flow from Burner		100 %	83.6 %	65.7 %	45.4 %	24.0 %
Portion of Total Boiler Gas Mass Flow Overfire		0 %	16.4 %	34.3 %	54.6 %	76.0 %
Diesel Exhaust O_2	Vol.-%, wet	11.6 %	11.6 %	11.6 %	11.6 %	11.6 %
Windbox O_2	Vol.-%, wet	13.1 %	13.1 %	13.1 %	13.1 %	13.1 %
	Mass-%, wet	14.6 %	14.6 %	14.6 %	14.6 %	14.6 %
O_2 at Burner Exit	Vol.-%, wet	1.2 %	1.2 %	1.2 %	1.2 %	1.2 %
O_2 at Economizer Outlet	Vol.-%, wet	1.2 %	2.9 %	4.8 %	6.9 %	9.1 %
Gas Temp. at Boiler Inlet	°C	1540	1360	1160	930	670
Gas Temp at Economizer Outlet	°C	150	150	150	150	150
Fresh Air to Total Diesel Exhaust Flow Rate	kg/kg	0.218	0.174	0.131	0.082	0.043
Steam Produced per Mass of Burner Fuel	kg/kg	14.96	15.32	15.91	17.19	20.77
Steam Produced per Diesel Output	kg/kW	5.3	4.3	3.4	2.4	1.5
Steam Produced per Mass of Diesel Exhaust	kg/kg	0.776	0.635	0.496	0.351	0.214

air is thus employed to increase the burner temperature. A net additional output of 40 MW is produced with the steam cycle, the steam conditions being 103 bar and 510°C [Shelor 1994].

In these power plant schemes, diesel jacket heat is used to heat condensate from the condenser before the deaerator [Shelor 1994].

A diesel/coal combined cycle (DCCC) power plant is a further interesting diesel power station concept. Exhaust gases from one or more engines are ducted into a coal-fired boiler with specially designed micronized coal burners [Smith 1994]. Micronized coal has more exposed coal surface. It therefore requires less oxygen for combustion, and the oxygen in the exhaust gas can be used to combust the coal in the boiler. The author says that a DCCC industrial cogeneration system could have a net output of 11.6 MW while providing steam at 3.8 kg/s. This size of unit would have two 3.15 MW diesel generators fired with fuel oil. The remainder of the electricity (5.3 MW) would be produced by an extraction-condensing steam turbine-generator.

O'Keefe [1995] mentions two diesel plant solutions where hot exhaust gas from the engine is led to the boiler to supply a part of the combustion air and reduce air heating before combustion. There are no separate waste-heat boilers in these concepts. In the first one, the engine driven generator produces 2.13 MW, and the existing steam turbine generator 1.95 MW. The heat of the engine jacket water is not utilized. In the second plant, engine jacket water is recovered to heat process water. The boiler fires coal, which is micronized at the plant for improved combustion.

As shown by these examples, a lot of different combined schemes can be designed, if additional fuel is burned in the waste heat steam boiler. Different fuel can be utilized in the engine and the burner. No very general data can be given concerning distribution of the total output between the engine and turbine. Obviously, the parameters should be optimized case by case. It is clear, however, that diesel and gas engine plants can be built to boost the electric output of the existing plants and to improve their energy economy.

4.2 Combined heat and power (CHP) plants

4.2.1 German applications

In Germany, local and regional CHP plants have become very popular in recent years. The plants usually include natural gas driven engines, peak load boilers, and thermal storage. Below, some examples are given of such plants, called "Blockheizkraftwerk" (BHKW) in German.

Juptner [1996] reports on a German, natural gas driven CHP plant Haar equipped with three 16-cylinder lean-burn SI engines. Each engine has an electric output of 1.29 MW. The three CHP modules cover less than one third of a total thermal energy demand of about 20 MW of the network. The engines operate thus as base load producers. They are connected in parallel with two heating boilers of 7 MW and thermal storage. The engines are run based on the thermal load.

A very similar power station, the CHP plant Am Kranzberg in Werdau, Germany, comprises three turbocharged natural gas driven lean-burn SI V-16 engines with an electric output of 1.06 MW and a thermal output of 1.5 MW each [Koch 1995]. The engines started energy production on the 1st of January 1995. In addition to the engines, there are two hot water boilers and a thermal storage system in the plant. Fig. 9 shows the heat demand in Werdau and the adaptation of the plant Am Kranzberg to it.

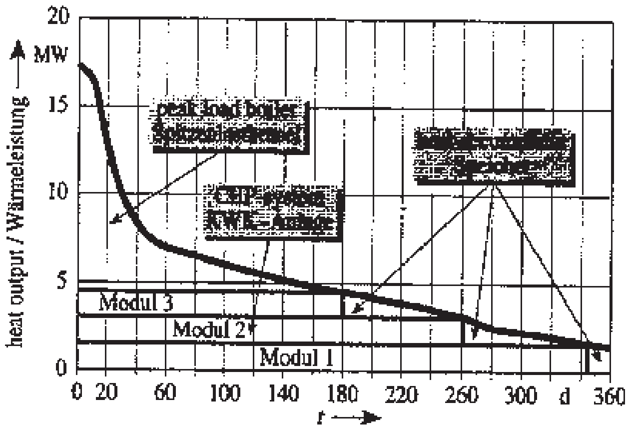

Figure 9. Classified annual duration curve of the heat demand in Werdau split according to basic cogeneration load, heat accumulator and peak load boiler [Koch 1995].

The dual-fuel gas engine power station MHKW Resse is also a real CHP plant. The gross electric output of the plant is 10.23 MW. Waste heat of the engines can generate at its highest a steam output of 11.90 MW (22.5 bar, 270°C) or a district heating output of 12.52 MW [Röchling 1995].

In the town Forst in Germany, a decentralized energy service solution was selected, when the old coal-fired heating and power plant had to be replaced [Görzig 1995]. Several modern CHP plants were built within the residential areas in order to minimize losses in the transfer pipes. Consequently, the noise emissions of the plants had to be reduced to below 35 dB, required by the German standards “TA Lärm” for residential areas. The contract was made in June 1993 and the engine-driven CHP units were connected in April and May 1994.

The four CHP plants comprise altogether ten natural gas driven engines, five of which produce 575 kW electricity and 960 kW heat [Görzig 1995]. The output of five other units is 210 kW_e and 348 kW_t. The total power output of 4 MW of the plants forms about 30% of the total power sales of the company. In addition to the engines, each plant has one or more heating boilers and two or three thermal storage systems. In the summer time, one engine per plant covers the base thermal load. The thermal storage was dimensioned so as to accept an excess heat of two hours of operation from an additional generating set serving as a peak load producer.

The flow scheme of the CHP plants Forst is in Fig. 10 [Görzig 1995]. The temperature of the water leaving the plant is kept at 90 to 110°C depending on the requirements of the consumers. The engine driven modules produce the base load. There are three methods to keep the heating water temperature at the desired level:

* The engines generate such thermal output that the outgoing temperature is 90°C. The excess heat is stored.

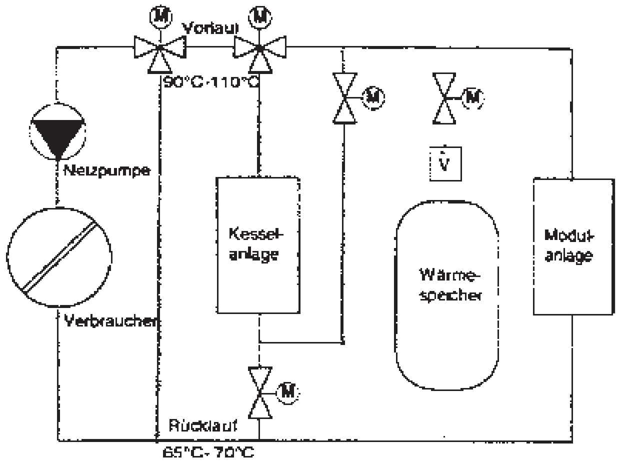

Figure 10. Flow chart of the CHP plants in Forst [Görzig 1995].

* The water from the engine plant is led to the inlet of the boiler and the latter increases the temperature to 98-110°C.

* The engine and the boiler run parallel. The waters are mixed by means of a three-way valve and the outgoing temperature is adjusted to 91-98°C. This operation concept is more favorable than the previous one in this temperature range, since the unfavorable use of the boiler at minimum load is avoided [Görzig 1995].

A very energy-efficient natural gas driven CHP plant has been built in Hochheim, Germany [Every... 1996]. The electric output of the plant is only 50 kW, but the advanced waste heat recovery system increases the thermal output up to 122 kW. The plant heats a swimming hall, a fire-service house, and a block of flats. Engine waste heat is recovered not only from cooling water and the two exhaust heat exchangers, but also from the oil cooler, the water-cooled catalyst cover, and from the alternator. Warm air removed from the noise protecting capsule of the engine is exploited as preheated combustion air in a gas fired heating boiler. The second exhaust heat exchanger cools the engine exhaust down to 37°C, i.e., below the dew point. Water from a swimming pool is heated from 28 to 32°C in this heat exchanger.

Convective and radiative waste heat of the engines is also utilized in another small German CHP plant in Biesenthal [Exner 1995]. The plant, with two natural gas driven engines, produces 140 kW electricity and 270 kW heat. Additionally, altogether 28 kW thermal energy is recovered from the surfaces of the engines during the winter period for defrosting and heating purposes. The engines are run based on the thermal energy demand and in parallel with the network. Two natural gas fired boilers with a thermal output of 1.34 MW each and thermal storage of 8 m^3 are also included in the new plant, replacing the old coal-fired heating station.

List [1996] states that operational hours of above 5500 h/a are considered as a feasibility limit by the professional authorities in Germany. The author reminds us that there are engine-driven CHP plants that can be run either based on the thermal or the electricity demand. According to [Every... 1996], a German energy company calculates that each of their five remote-controlled CHP plants must run at least 6000 h/a. As an example, one of the company's CHP plants covers 48% of the total heating energy demand concerned.

4.2.2 CHP plant at the University of Nottingham in the U.K.

In the U.K., a diesel driven CHP plant has been in use at the University of Nottingham since 1989 [Pickering 1994]. The plant running on HFO generates 3.7 MW of electricity at 11 kV and meets the base electricity load of the university campus. High grade heat, recovered from the exhaust gases, is exploited in the high pressure hot water district heating system of the campus. Lower grade heat is recovered from the engine jacket cooling system to meet a thermal energy demand local to the CHP plant comprising academic buildings and greenhouses.

The campus has a high pressure hot water (HPHW) district heating system operating with flow and return temperatures of 160°C and 120°C, respectively [Pickering 1994]. A diagram of the cooling and heat recovery systems of the plant is shown in Figure 11. The exhaust gases flow through a vertically mounted fire-tube boiler to produce HPHW at a temperature of 150°C. Some of the recovered heat is used in the CHP plant for fuel preheating and oil treatment, but most is utilized for district heating. Various cooling systems of the engine - cooling the exhaust valve cages/injectors, engine jacket and lubrication oil - operate at around 80°C off engine. There is thus no opportunity to recover high grade heat for the district heating system and it is not seen as practical to reduce the operating temperature of the system to allow more heat recovery. However, thermal energy is recovered from the engine jacket cooling system at about 80°C, through a heat exchanger, and used for space heating and hot water in the buildings and greenhouses in the nearby Life Sciences Department. A secondary cooling system removes the remaining heat rejected from the jacket cooling system, the lubrication oil, the valve cage, the intercooler and - when required - an exhaust gas dump heat cooler and dissipates it through a single stage evaporative cooling tower.

The CHP plant was designed to run 24 hours per day 7 days per week for 11 months in a year at constant load [Pickering 1994]. The remaining 4...5 weeks were planned for an annual overhaul. At night the electricity demand of the site can be less than the output from the alternator and the surplus is sold back to the electricity company. During the summer the engine can be kept operating even when the HPHW district heating scheme is shut down, by opening a bypass damper to divert a proportion of the exhaust gases around the heat recovery boiler and by dumping excess heat from the HPHW system into the secondary cooling system.

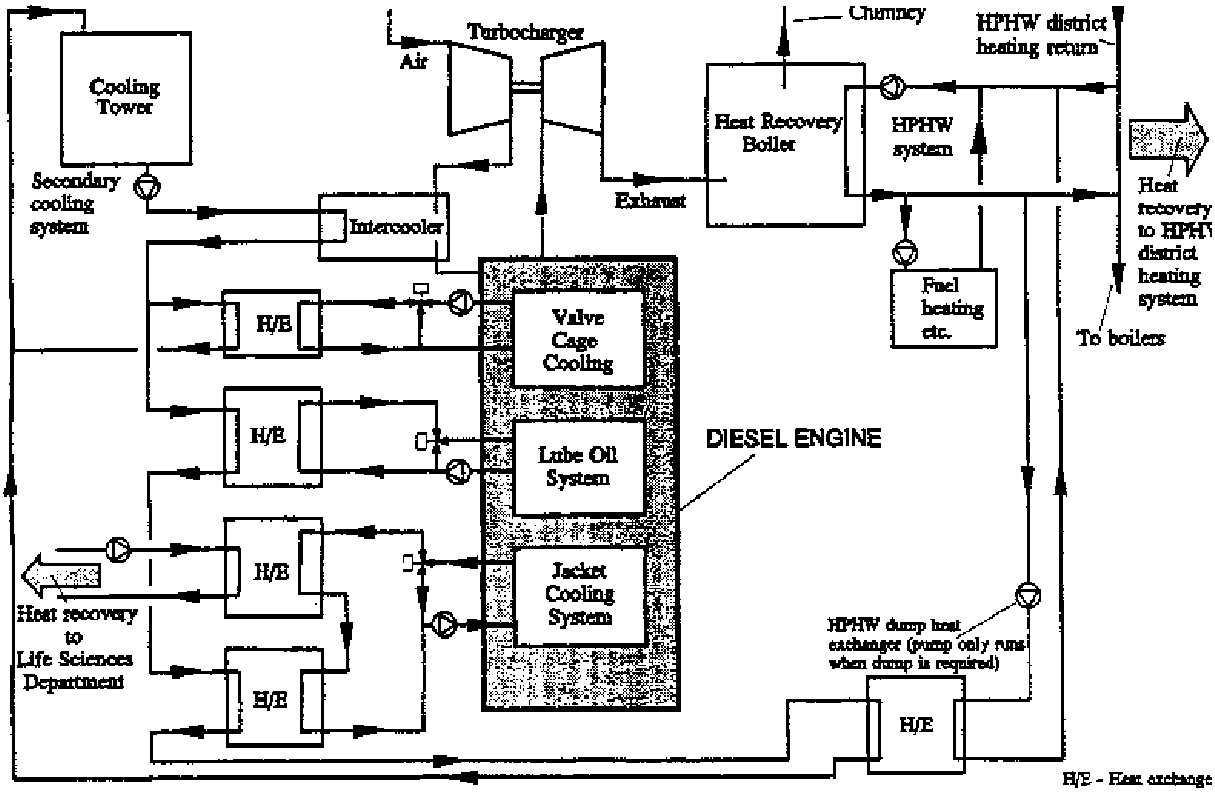

Figure 11. Cooling and heat recovery systems of the CHP plant in Nottingham [Pickering 1994].

The exhaust gases leave the engine turbocharger at about 360°C and they are cooled in the heat recovery boiler to about 220°C [Pickering 1994]. This is said to be a practical lower limit for the exhaust temperature to prevent acid condensation in the boiler, ducting and chimney. Based on operation at 3 MW_e generator load, there is 900 kW of heat recovered in the boiler and about 850 kW is exported from the CHP plant to the campus district heating system. The remainder of the thermal energy is used within the CHP plant for fuel heating and cleaning purposes. Up to 270 kW of heat from the engine jacket cooling system can be utilized in the Life Sciences Department.

Only a proportion of the thermal energy rejected in the jacket cooling system is hence recovered and there is additional heat rejected from the valve cage and the lubricating oil systems [Pickering 1994]. The authors state that it would be possible to recover a further 800 kW of heat at a temperature of about 80°C. This temperature is too low for the HPHW district heating system but the possibilities for utilizing it in nearby buildings is being investigated. The charge air is cooled currently by a single-stage intercooler. The temperature of the water leaving the intercooler is of the order of 30-40°C depending on the ambient air temperature. If a two-stage intercooler were adopted, heat could be recovered from the first stage and utilized in the district heating system. There is further potential to recover 400 kW of thermal energy in this way. The additional weight of a new large intercooler fitted onto the engine needs, however, to be considered.

4.2.3 CHP plant in Lahti, Finland

In Finland, the energy company Lahti Energia Ltd. has built a gas engine power plant for combined district heating and power production [Lahti... 1996]. The plant is based on a Wärtsilä 16V25SG spark-ignited engine, the electric output of which is 2.8 MW. The power station started operation in December 1995. Fig. 12 shows the simplified connection scheme of the plant. The design point has been selected so that the return water temperature is 50°C and the outlet temperature 95°C. One part of the return water is fed directly to the cold part of the exhaust boiler. This way, the exhaust gases can be cooled down to about 70°C. The thermal output is 3.65 MW at the design point.

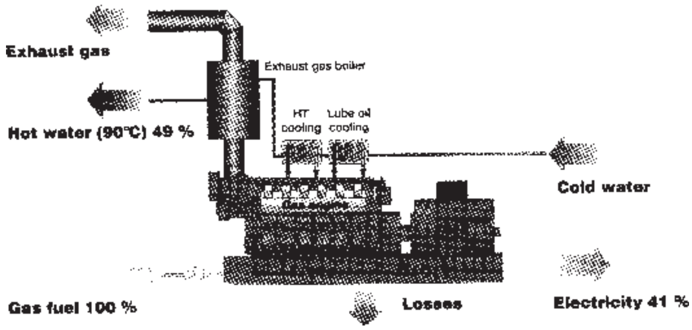

Figure 12. Scheeme of the Lahti CHP plant [Lahti... 1996].

Lahti Energy Ltd. will build two more CHP plants, similar to the first one [Lahti... 1996]. The one increases the power output of the district heating plant to 5.6 MW and the total thermal output of the engines to 7 MW. The peak duration time of the engines will be around 3500 h/a at the beginning, but it will be lengthened when the thermal demand increases.

The third power station will be installed at a food company [Lahti... 1996]. The exhaust gases of the engine are first led to a biscuit oven of the factory. Thereafter, the gases flow through a heat recovery boiler which produces district heat.

4.2.4 Further concepts and projects

In Austria, a gas engine driven CHP plant produces 588 kW electricity per module for a chemical factory [Energy... 1996]. The plant comprises four 20-cylinder engines running on low calorific value gas wasted by the factory. Pro module, 200 kW of waste heat is utilized to generate 97 g/s of saturated steam at a pressure of 11.5 bar and a temperature of 186°C. The plant comprises two unfired exhaust boilers. The power plant will operate ca. 7000 h/a.

Wärtsilä have standardized the most common heat recovery solutions in their Pure Energy Plant concept [Kunberger 1996]. In one system, hot water is produced up to

110°C by engine cooling and exhaust; in another, steam up to 10 bar by the exhaust (condensate and water can be preheated by engine cooling); the third concept utilizes exhaust gas directly, just like that used for drying in industrial processes (engine cooling can be adopted for preheating fresh air).

The utility company Göteborg Energi AB has ordered a Wärtsilä Pure Energy Plant to be built at the site of the Pripps Brewery in Gothenborg, Sweden [Kunberger 1996]. It will consist of three Wärtsilä 18V28SG lean-burn spark ignited gas engines and generate 13 MW electric power and 16 MW thermal output. Start of production is scheduled for 1997. Fig. 13 shows the flow diagram of the plant.

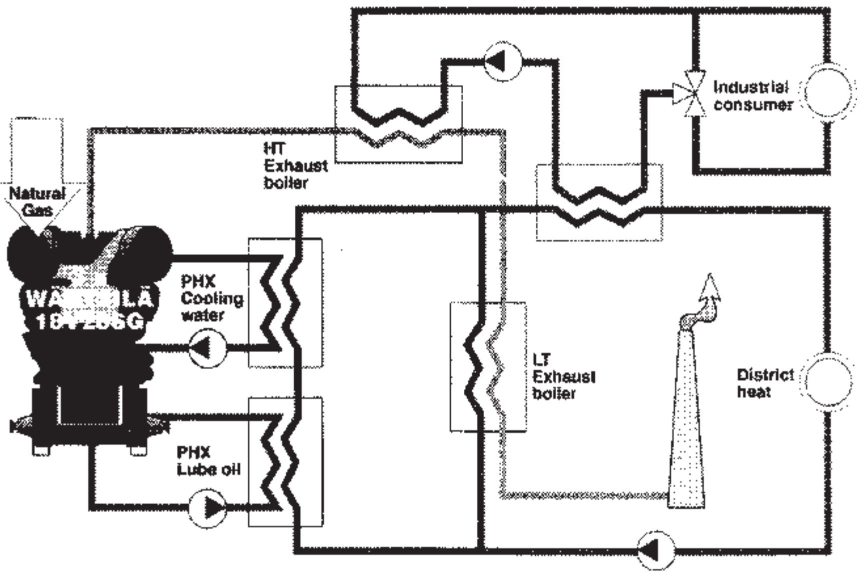

Figure 13. Flow scheme of the Göteborg Energi plant [Kunberger 1996].

The medium-speed SI gas engines GMT A32G, developed by Fincantieri, are said to be particularly well-suited for cogeneration plants [Chellini 1995a]. Thermal energy is obtained from various engine heat sources, lube oil, cylinder water cooling system, charge-air cooler and exhaust line. The engine generates an electric power of 346 kW per cylinder and a thermal output of 411 kW/cyl, assuming the hot water used in the cogeneration system is returned to the engine at around 60°C. A net electric efficiency of 39.4% is achieved.

Westergren [1989] presents a CHP concept equipped with a dual-fuel gas diesel engine. The electric output of the plant is 7.1 MW. The total thermal output of 7.5 MW includes 4.0 MW from the exhaust boiler, 2.67 MW from jacket and charge air cooling, and 0.78 MW from lubricating oil. The concept can be changed to produce process steam at about 11 bar, 195°C. Waste heat of the second charge air cooler - lower temperature level - can be utilized to preheat the intake air of the engine room. This enhances the district heating output.

A project reported in [Shelor 1994] is designed to generate up to 24 MW electricity either as base load or only during peak periods with the diesel exhaust acting as combustion air for a 38 kg/s micronized coal fired boiler. The boiler provides steam at 75 bar, 450°C.

4.2.5 Air CHP systems

The company Aircogen Ltd has developed a combined heat and power system claimed to be capable of providing electrical and thermal power with efficiencies in the region of 95% [Mullins 1996b]. Power output of the plants ranges from 50 kW to 1.5 MW. Even in this system, convective and radiative heat is recovered: air for heating the building is heated direct from the gen-set. The plant is said to lend itself especially to swimming pools, supermarkets, leisure centers, etc. It can be applied to any system where more than 2 m^3/s of air is moved. Designs for 60 m^3/s or more are said to be practical.

In the basic system, the gen-set is placed in the return air section of an air handling unit [Mullins 1996b]. The return air flows through several heat exchangers: first, the radiator dealing with the engine cooling water; second, the heat exchanger handling the exhaust gases; and third, an air-to-air heat exchanger extracting heat from the oil cooler and engine surfaces, Fig. 14. The heated air is then mixed with fresh air to reduce it to a suitable ambient temperature.

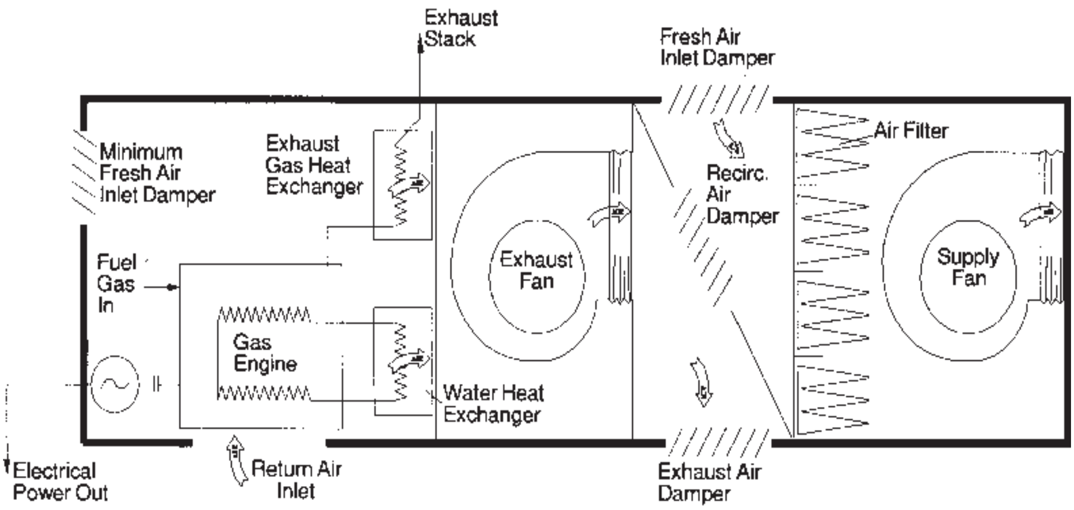

Figure 14. A schematic of the Air CHP system [Knowles 1994].

Where hot water is needed, an optional water heating pack can be provided [Mullins 1996]. The system can be tailored to specific applications with a variety of options available. As an example, electric elements can be set in the air stream or water circuits to allow variations of the heat/power ratio. An additional exhaust heat exchanger can also be fitted to vary the water/air heating ratio.

Depending on the case, electric power output can range from 50 kW to 1.5 MW, air heating from 100 kW to 4.0 MW, water heating from 40 kW to 1.0 MW, air/chilled water from 60 kW to 2.0 MW, and low-temperature cooling from 100 kW to 1.0 MW [Mullins 1996]. The last one is only available on units with electric output of 250 kW or more.

Two of the first applications of the air CHP system are mentioned in [Mullins 1996]. One is in a new superstore development for Safeway in Milton Keynes, U.K. The installation is comparatively modest in size, using a VarityPerkins SI gas engine producing 145 kW of electricity with a maximum heat transfer to air of 336 kW.

The other is in a new postal distribution center for the UK Royal Mail in Peterborough, U.K. [Mullins 1996b]. This plant is based on two naturally-aspirated 12-cylinder SI gas engines, each generating 216 kW of electrical power. The air CHP system was selected, since it can provide heated air and low-pressure hot water simultaneously. As soon as the air heating requirement is met, the system diverts heat into the hot water circuit.

4.2.6 Concluding remarks and power-to-heat ratio

Again, there are very many innovative and efficient concepts in the area of diesel and gas engine driven CHP plants. The design of a plant depends on the local requirements and circumstances. One of the most important issues of the CHP plants is the power-to-heat ratio of the plant. It is very beneficial to join as much power production as possible to a given thermal energy demand. Ref. [Rumpel 1996] sees that the industry will show growing interest in the plants with high electricity ratio in the future, since in some areas, the specific heat consumption has decreased in recent years, whereas the specific power consumption has increased.

A great advantage of a diesel CHP plant is the high power-to-heat ratio. In the largest plants, roughly as much thermal energy can be obtained as electricity. The power-to-heat ratio should be evaluated, however, always in connection with the total efficiency, since high power-to-heat ratios can be obtained by wasting a great portion of the surplus heat.

The power-to-heat ratios of the reviewed CHP plants vary rather largely. All the plants incorporating an efficient waste heat exploiting system are gas driven. In the biggest plants, the power-to-heat ratio ranges from 0.80 up to 0.95. The cylinder output of these engines is in the range of 250 to 450 kW. The smallest plants with a very efficient waste heat utilization show lowest power-to-heat ratios of 0.4 to 0.45. In these cases, the total electric output of the engines is 50 to 150 kW.

In the large HFO driven CHP plants, the power-to-heat ratio can be higher than one, since the shaft efficiencies of the engines are very high resulting in rather low exhaust temperatures, and the exhaust gases can not be cooled in the waste heat boiler to as low a temperature as in gas driven plants. The total efficiencies can still be rather high, at a level of 80%. Karlemo [1993] states that a CHP plant in Sweden reaches a total efficiency of 83.4%, the share of electricity being 41.7% and of heat 41.7%, as well. No accurate data on the efficiency definition are given, nor on the fuel. The output of the engine is 4.0 MW.

4.3 Engine-driven heat pumps

Engine-driven heat pumps form one plant concept reported every now and then. In a narrow sense, the plants are not "power" plants at all, since they do not necessarily produce any electricity, at least to be sold outside of the plant. However, the plants utilize gas or diesel engines, are rather energy-efficient and can contribute to the CO_2 reduction. So it was decided to include a brief survey of these plants.

Ref. [Heat... 1996] reports on an engine-driven heat pump system developed in Germany. The plant utilizes ambient air as heat source. The refrigerant flows through the coolers at a temperature which is around 8°C below that of the ambient air. The air cools about 3°C, flows thus downward, and draws new warmer air to the coolers. The temperature of the refrigerant is then raised to ca. 65°C which is the highest temperature required at an ambient temperature of -10°C. The heat pump is driven with a gas engine, waste heat of which is also utilized for heating purposes. A supplementary engine CHP plant generates power for the electric equipment like motors, pumps, or regulation devices.

The heat pump plant in Essen-Freisenbruch in Germany, taken into use in August 1996, comprises a Caterpillar engine driven CHP plant with an electric output of 100 kW and three York heat pumps, also driven by Caterpillar engines [Heat... 1996]. The heat pumps can produce up to 650 kW each. There are two peak load boilers in the plant, as well. The representatives of the manufacturer calculate that up to 40% of the heating energy can be obtained "without costs" from the ambient air, while the CO_2 emissions decrease by up to 70%.

The heat pump plant in Greifswald, Germany, has produced heating energy for two clinics of the University since the beginning of 1995 [Heat... 1996]. The thermal output of the plant is 650 kW. The power from another CHP plant, 700 kW, is fed into the public network. In Duisburg-Meiderich, Germany, the heat pumps deliver ca. 80% of the heating energy demand of 600 residences with a total living area of more than 30,000 m^2. The total output of the heat pumps is 3.25 MW. Two gas fired boilers take care of the peak load with an output of 1.07 MW each [Heat... 1996].

5 EFFICIENCY AND SOME OPERATIONAL EXPERIENCES

One of the greatest advantages of reciprocating internal combustion engines is the high brake thermal efficiency (BTE). The largest two-stroke diesel engines convert more than 50 per cent of fuel input to shaft power. The BTE of unthrottled engines (diesel and gas diesel) remains high even at part loads, down to around 40% load. The BTE is also insensitive to the ambient conditions in contrast to gas turbines.

The BTE depends of course strongly on the engine size and type, but BTEs of above 40% are today common even in the truck engine class. Automotive spark ignition (SI) gasoline engines usually remain below 40 per cent due to their lower compression ratios and intake throttle required for the control of mixture strength. However, greater lean-burn SI gas engines reach BTEs of more than 40%.

5.1 Oil burning engines

5.1.1 Combined cycles

Table 4 gives valuable information about operational data of the Coloane Power Station, which has three pairs of low-speed two-stroke diesel engines [Cordeiro 1996]. Data shown in the Table are based on accumulated values for each year. Calculated figures are hence average values for the year. The figures for produced energy are the total energy generated, including energy produced by the turbo compound systems and steam turbines, cf. Table 2. The specific fuel oil consumption rates (SFOC) include the consumption during starts and stops and the removal of sludge and any other waste. The figures thus represent the total fuel oil consumption of the plant. They are not corrected for lower calorific value; the average heating value has been some 5 to 6% lower than that of the ISO reference condition. The lubrication oil consumption rates (SLOC) are also gross figures, including sludge removal, oil replacements, etc. [Cordeiro 1996].

The given figures thus stand for real results, obtained in the known actual local conditions. Omitting the first year data of each pair of units, we can calculate that the peak duration of the units G03&G04 and G05&G06 has been 5600 to 6900 h/a and 6200 to 7000 h/a, respectively. The engines have hence been run at rather high loads during the given years. The overall efficiencies have varied from 45.3% to 46.8%, i. e., rather slightly. It is interesting that there is no clear difference in the overall efficiency between the oldest units and the units G05&G06. The overall efficiency is calculated as net generated power divided by total fuel consumption [Cordeiro 1996].

Table 4. Operational data for the Coloane Power Station [Cordeiro 1996].

		1987	1988	1989	1990	1991	1992	1993	1994	1995
	Generation (GWh)	118.6	287.8	319.7	345.4	336.2	294.6	296.1	312.9	282.5
Units	Auxiliaries (%)	3.6	3.4	3.2	3.5	3.9	4.1	4.1	4.0	4.6
G03	SFOC (g/kWh)	186.1	186.0	189.2	190.7	186.8	188.6	187.6	187.0	185.4
&	SLOC (g/kWh)	1.7	1.5	1.7	2.0	1.9	2.1	3.2	2.9	2.5
G04	Average Load (MW)	21.9	22.6	22.6	22.6	22.0	21.3	21.5	21.3	20.6
	Reliability (%)	99.7	91.4	94.9	96.3	99.0	97.9	95.7	96.1	98.6
	Availability (%)	93.0	79.8	79.7	86.6	88.2	88.9	84.0	88.5	92.9
	Overall Efficiency (%)	46.1	46.5	45.8	45.3	46.1	45.5	45.8	46.0	46.1
	Generation (GWh)					206.6	502.6	487.5	541.2	508.9
Units	Auxiliaries (%)					4.0	3.5	3.5	3.5	3.9
G05	SFOC (g/kWh)					189.1	184.6	185.7	186.2	185.9
&	SLOC (g/kWh)					1.9	1.7	2.2	1.9	2.1
G06	Average Load (MW)					36.0	34.9	34.7	34.9	34.3
	Reliability (%)					98.0	97.3	94.7	98.0	96.6
	Availability (%)					91.6	89.1	85.1	89.7	90.7
	Overall Efficiency (%)					45.3	46.8	46.5	46.4	46.3
	Generation (GWh)									237.0
Units	Auxiliaries (%)									2.7
G07	SFOC (g/kWh)									190.5
&	SLOC (g/kWh)									1.9
G08	Average Load (MW)									47.5
	Reliability (%)									87.3
	Availability (%)									84.8
	Overall Efficiency (%)									45.6

A maximum energy efficiency of 54% is given for the combination of the low-speed two-stroke engine and the turbocompound system (TCS) of the Vale D power station in Guernsey [Mullins 1995]. The main contributor to thermal economy is the TCS, but improvements in the fuel consumption are also achieved by appropriate adaptation of the fuel injection and turbocharging systems. The improved design of the pumps for cooling water, lube oil and fuel supply enhances operating economy of the power station as well, since the electrical load imposed by these pumps is said to be some 40% less than previous designs [Mullins 1995].

A diesel combined technology (DCC) system with two-thirds diesel output and one-third steam turbine output would be near the optimum configuration for the lowest overall heat rate according to Shelor [1994]. If the system is configured to have two-thirds steam output and one-third diesel output, it would utilize nearly all of the diesel exhaust as combustion air and most of the fuel would be burned in the boiler. This configuration is not as efficient, but may still be close to the point of the best economic efficiency, if the boiler fuel is much cheaper than the diesel fuel.

For a combined cycle plant with six diesels and a total output of 130 MW, [Shelor 1994] gives a total net efficiency of 49%. In this concept, a reheat steam turbine with an output of 40 MW is used. Reduction of the by-passing exhaust amount will increase the boiler fuel input and consequently the turbine output. However, the efficiency decreases at the same time. For a plant with a total output of 230 MW, a net efficiency value of 44% is given.

For another DCC plant with a diesel output of 60 MW and steam turbine output of 40 MW, a total net efficiency of slightly below 44% is given, if the boiler fuel is heavy fuel oil and 40% of the diesel exhaust is sent through the burner. If orimulsion is used as boiler fuel and more additional air is employed to produce higher burner temperature, the net efficiency will decrease to slightly below 43% [Shelor 1994].

According to Smith [1994], optimum overall cycle performance of a diesel/coal combined-cycle (DCCC) plant can be achieved by adding fresh air to the diesel exhaust. Enough fresh air ensures stable and complete combustion of the micronized coal. An exhaust/air mixture temperature of 230 to 260°C is said to be ideal for use in conventional boiler equipment, and this temperature level is achieved by fresh air addition.

A test has been set up to simulate firing of micronized coal with ambient air and diesel exhaust. The percentage of oxygen in the combustion air was varied between 11 and 18.5%. The tests showed that stable combustion can be achieved at values lower than 15% oxygen on a wet-weight percentage basis [Smith 1994]. The author emphasizes the importance of air ratios and O_2 content of the exhaust/air mixture on plant size and heat rate. The increased oxygen content results in a larger but less efficient power cycle, Fig. 15.

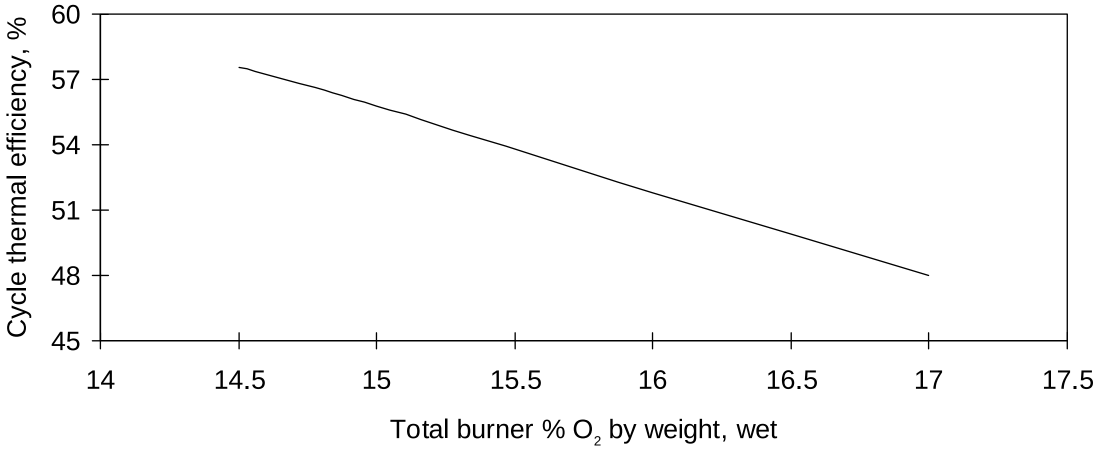

Figure 15. Cycle thermal efficiency of a diesel/coal combined-cycle plant with various O2 contents of the diesel exhaust/air mixture; steam production 3.8 kg/s. Data based on [Smith 1994].

5.1.2 Single cycles

For medium-speed diesels as single prime movers, [Shelor 1994] gives efficiencies of 48% and higher. This figure covers both HFO and natural gas driven dual-fuel engines.

Chellini [1995b] mentions that the low-speed two-stroke Sulzer 12 RTA84C power station diesel unit in Martinique offers high efficiency of an order of 50% without heat recovery. The engine can burn low-cost heavy fuel.

Rosser [1994] gives one example of the efficiencies realized in actual diesel power plant operation. During a period of 28 months in 1991-1993, the Esperance 14 MW diesel power station produced 3.76 kWh electricity with one liter of diesel fuel. Depending on the fuel oil quality, the average efficiency has thus been 35 to 38%. Based on the little given information, one can conclude that the engines have been run at an average load of around 40 to 45% during that period, which accounts for the rather low efficiency level.

For the diesel-generator of the Nottingham CHP plant, the electrical generation efficiency is shown as a function of load in Fig. 16 [Pickering 1994]. The efficiency is defined as alternator output divided by fuel energy input, and is based on a measured gross calorific value (GCV) for HFO of 42.5 MJ/dm^3 at 15°C. The figure shows that the engine has a generation efficiency of about 37.5% (based on GCV) at 3.6 MW which decreases to about 36% at 3.0 MW. These efficiency figures should be multiplied by about 1.06 to make them ready for comparison with those based on lower heating value (LHV) of the fuel. The range is thus 38 to 40% based on LHV. The engine is a medium-speed one running at 600 rpm. Waste heat is not utilized very thoroughly, so the overall efficiency varies from 47 to 50% based on GCV (50-53% on LHV).

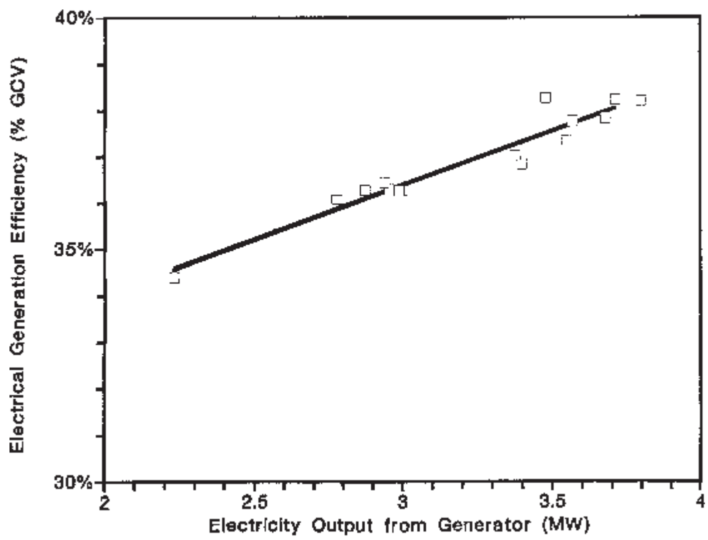

Figure 16. Electric efficiency versus output of the HFO driven CHP plant at the University of Nottingham [Pickering 1994].

In Sweden, a CHP plant is said to reach a total efficiency of 83.4 per cent, the share of electricity being 41.7% and that of heat the same, 41.7 per cent [Karlemo 1993]. No accurate data on the efficiency definition are given, however, nor on the fuel. The output of the engine is 4.0 MW.

In a sophisticated diesel power plant concept, using pyrolysis oil as fuel, an efficiency of about 25% can be achieved in a small scale of operation according to Solantausta [1994]. The efficiency is then calculated from wood to power, based on the lower heating value of wood.

5.2 Gas engines

5.2.1 Spark-ignited (SI) engines

The electric efficiency of SI gas engines depends on the engine size. In small high-speed SI gas engines up to 200 mm bore, efficiencies of around 35% can be achieved in electricity production according to Vuorinen [1995]. With comprehensive waste heat utilization, a total efficiency of 90 per cent can be reached. The fuel economy is improved when the engine size increases and speed decreases: for medium-speed prechamber SI gas engines with a bore of above 200 mm, electricity efficiencies of 38 to 39 per cent are given in [Vuorinen 1995]. In the combined production of heat and power (CHP), around 90% is again obtainable.

Kinnunen [1996] gives a peak electric efficiency of 42% for a lean-burn SI gas engine with a bore of 340 mm and an output of 5.5 MW. The total efficiency of the plant is said to be 85%. In the natural gas driven CHP plant in Lahti, Finland, the total efficiency will be 88%, when slightly smaller lean-burn SI engines are used, having a bore of 250 mm [Lahti... 1996]. The power-to-heat ratio will be 0.8. In [Chellini 1995a], a brake thermal efficiency of 41% and an electric efficiency of 39.4% are given for an approximately similar size medium-speed lean-burn SI gas engine with a bore of 320 mm and stroke of 390 mm. The electric efficiency of the turbocharged lean-burn SI engines of the CHP plant Am Kranzberg is said to be 36% and the total efficiency 87% [Koch 1995]. The output of the natural gas driven engines is 1.06 MWe, so they are clearly smaller than those of [Kinnunen 1996] and [Chellini 1995a].

The small-size natural gas driven CHP plant in Hochheim, Germany, has the highest total efficiency reported: 98% [Every... 1996]. The electric output of the plant is only 50 kW, but the thermal one rises to even 122 kW, since the exhaust gases are cooled down to 37°C - below the dew point - in an additional exhaust heat exchanger to heat water for a swimming pool. Without this latest heat recovery, the total efficiency is 92%.

An electric efficiency of 34% is given for an Austrian CHP plant burning low calorific value (LCV) gas as fuel [Energy... 1996]. The SI engines have 20 cylinders and generate

588 kW electricity each. The lower heating value of the LCV gas wasted by a chemical company is 0.5 to 0.6 kWh/m^3 only. The combustion temperature is said to be as low as 900°C, since the gas contains a lot of nitrogen (74 to 78%) and carbon dioxide (3.5 to 5.5%)

5.2.2 Gas diesels (GD)

In GD engines, the electric efficiency varies strongly depending on the power required for gas compression. Ref. [Paro 1995] states that a gas diesel (GD) engine Wärtsilä V46GD can achieve an optimum shaft efficiency of 49.6% in gas operation at about 85 to 90% load. At MCR, the efficiency is around 49%, and at half load, ca. 47%. The figures correspond to ISO 3046 conditions. The engine driven pumps are excluded. The speed of the engine is 500 rpm, and the full load BMEP 22.5 bar. The author says that the energy consumption of the gas compression is around 2.5% of the shaft output, if the inlet gas pressure is 16 bar. When the gas is delivered in liquefied form, the gas compression takes only 0.8%.

Plant heat rate of the gas-driven Ringgold cogen power station is about 9300 kJ/kWh corresponding to an efficiency of around 39% [O'Keefe 1995]. The availability has been on average 90.1% during 48 months of operation. The GD engines of the plant run on natural gas boosted to 250 bar for injection. The pilot oil amount is about 5%. The specific lubricating oil consumption averages about 0.8 g/kWh. The tests have revealed significant differences in oils. If the ash formation rate is high, the turbocharger performance is impaired and the boiler tubes can be fouled.

Table 5 gives gross electric and total efficiencies for a GD power plant [Westergren 1989]. The plant was designed as a CHP plant in a town in Southern Finland. Natural gas is used as the main fuel. It is compressed to a pressure of 250 bar before injection into the cylinders. A small amount of pilot fuel is used for ignition.

Table 5. Efficiencies of a natural gas driven gas diesel power plant [Westergren 1989].

Load	%	100	90	80	70	60	50
Gross electric efficiency	%	42.6	42.5	42.3	41.8	41.1	40.1
Total efficiency	%	84.0	83.3	82.7	82.5	81.7	81.0

One can see that the gross electric efficiency remains above 40% in a load range of 50-100%, the maximum value being 42.6% at full load. At the same time, the total efficiency varies between 81.1 and 84.0%. The authors state that the gross electric efficiency can be kept at above 41%, if the optimum point is adjusted to 85% load. The internal electricity consumption of the plant is rather high, however, since the gas pressure must be raised from 4 bar to 250 bar by means of the compressors. A range of 370-490 kW is given for a 7.1 MW_e plant. If the gas is delivered to the plant at a pressure of 16 bar, the internal electricity consumption falls to about 310 kW [Westergren 1989]. - It should be noted that the engine manufacturer later increased the gas injection pressure to around 350 bar.

5.2.3 Dual-fuel engine

Röchling [1995] reports that a net electric efficiency of about 40% was achieved in the CHP plant MHKW Resse in 1993. The dual-fuel engines use mining gas (MG) as the main fuel. The peak duration time was 8024 hours. The design value for the gross electric efficiency is 43.5% and the net one 39.4%. Fig. 17 shows how the electric efficiency varied with load in the measurements made for control system design. In gas mode operation, efficiencies of slightly above 40% were obtained at full load. At half load, the efficiency was still over 35% [Röchling 1995].

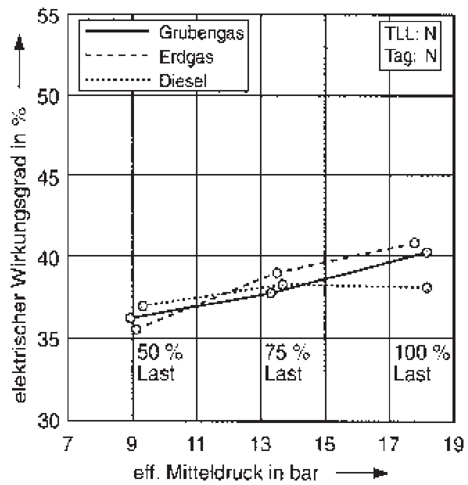

Figure 17. Electric efficiency of the CHP plant MHKW Resse versus load with different fuels [Röchling 1995].

5.3 Summary

Table 6 shows electric and total efficiencies given for gas engine driven power plants in the reviewed papers. As can be seen, the electric efficiencies vary from 35 to 47% and the total efficiencies from 85 up to 98%. It should be noted that the efficiencies are perhaps defined in slightly different ways and the circumstances do affect them, so too accurate conclusions must be avoided. In addition, the extent of heat recovery always depends on the local requirements and conditions.

Table 7 presents similar data for the oil driven engines. Two-stroke low-speed engines show the highest electric efficiencies, the combined cycle plant reaching up to 54% and the single cycle engine to 50% at rated power. Efficiencies slightly below 50% are

Table 6. Efficiencies of reviewed gas engine driven power plants.

Engine type	Fuel	Bore	Engine output	Electric efficiency	Total efficiency
		mm	MW	%	%
SI	NG		0.05		92...98
SI	NG		1.06	36	87
SI	NG	< 200		35	90
SI	NG	> 200		38...39	90
SI	NG	250	2.8		88
SI	NG	320		39.4	
SI	NG	340	5.5	42	85
SI	LCV		0.588	34	
GD	NG	320		42	
GD	NG	320		39	
GD	NG	460		45...47	
DF	MG	400		40	
DF	NG	400		41	

achieved with four-stroke medium-speed engines with combined cycle concept. Average values obtained in the Coloane Power Station (first row in Table 7) give valuable information about the real fuel economy of a diesel power plant, as does the bottom row. In the latter case, the average load has, however, most probably been very low. The diesel engines are generally somewhat larger than gas engines in Table 6, which partly explains differences in the electric efficiencies, but the fact is that oil driven engines usually show somewhat better fuel economy than gas engines.

Figure 18 shows efficiency regions for different power plant engines [Pischinger 1995]. The values agree rather well with those given in Tables 6 and 7 and confirm that shaft efficiencies of above 40% can be achieved even with modern highly-turbocharged lean-burn gas engines.

For a combined engine-steam turbine power plant, a maximum efficiency of 54% is given by Wiese [1996], Fig. 19, which also complies with information given in Table 7. The authors state that particularly large two-stroke and medium-speed four-stroke engines are not very favorable for combined cycles, since their exhaust temperatures are low, 250-290°C for the former, and 300 to 350°C for the latter. In addition, the temperatures

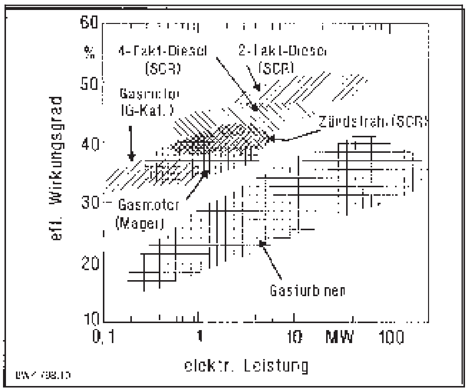

Fig. 18. Shaft efficiency of different internal combustion engines [Pischinger 1995].

decrease with load. Besides this, the exhaust energy represents only a rather small share of the wasted thermal energy, a great deal of waste coming out at a low temperature level in the cooling water. However, as can be seen in Fig. 19, the efficiency of such combined cycle plant is competitive in an output range of, say, below 50 MW. Fig. 19 also shows the clear advantage of reciprocating engines at part loads.

Further efficiency regions are given for the main competitors of diesel and gas engines in Fig. 20. It can be seen that single gas turbines remain below 40% and single steam turbines reach slightly more than 45% at their highest. Combined gas and steam turbine plants are able to obtain, however, efficiencies of more than 55%, if natural gas is used as fuel in both the gas turbine and steam boiler.

Table 7. Efficiencies of surveyed diesel power stations (CC, combined cycle; SC, single cycle).

Engine and plant type	Fuel	Bore	Plant output	Electric efficiency		Total efficiency
				Average	Rated power	
		mm	MW	%	%	%
2-str., CC	HFO	800	128	45...47		
2-str., CC	HFO	580	15.4		54	
2-str., SC		840	44		50	
4-str., CC			130		49	
4-str., CC			230		44	
4-str., CC			100		43...44	
4-str., SC	HFO		3.6		40	
4-str., SC			4		41.7	83.4
			14	35...38		

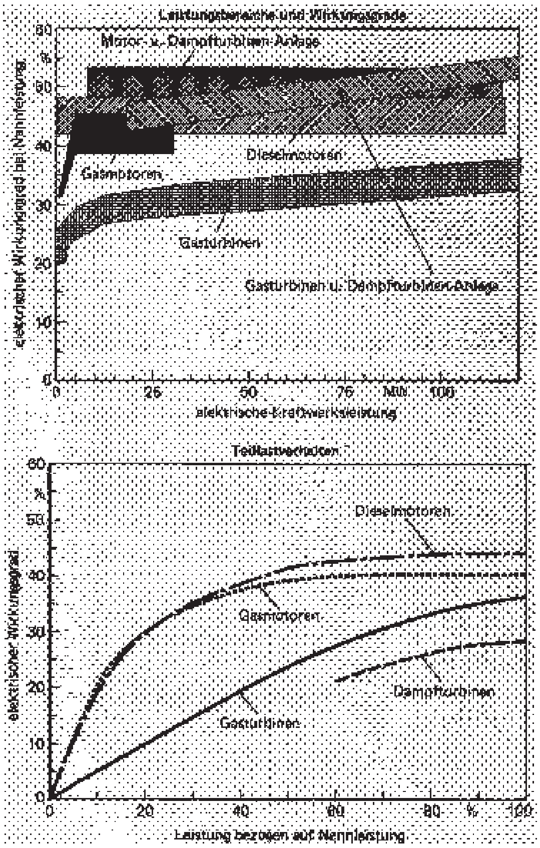

Figure 19. Electric efficiencies for different prime movers versus output (upper) and load [Wiese 1996].

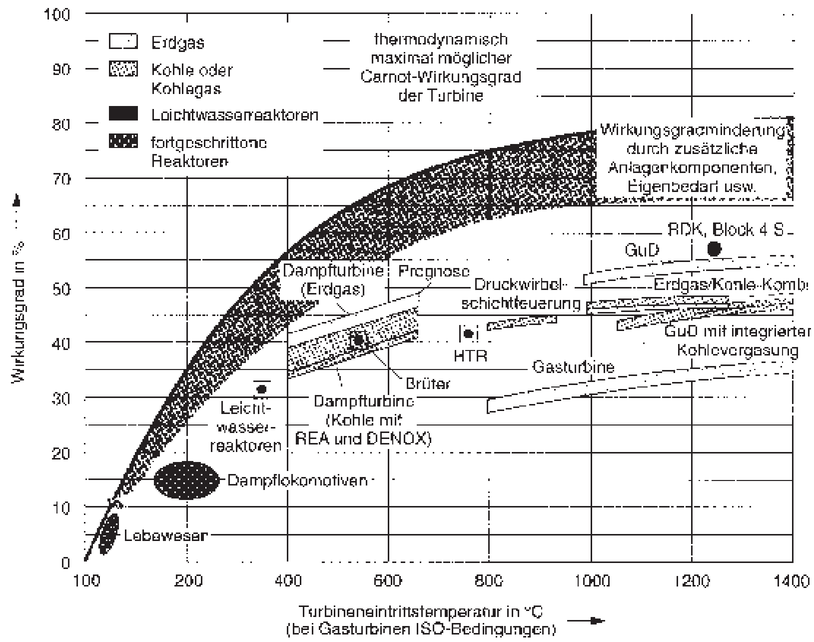

Figure 20. Ideal and real efficiencies of different turbine processes [Hässler 1996].

6 EXHAUST EMISSIONS

The emissions regulations are becoming more stringent both on land and at sea. Efforts to reduce pollutant formation are primarily internal, i.e., by developing combustion technologies. Nevertheless, the emissions limits are so strict, particularly in the industrialized countries, that the internal methods are usually insufficient. Post-combustion methods must hence be adopted for exhaust cleaning. Consequently, some companies have specialized in the development, production and installation of turnkey exhaust gas cleaning systems for diesel and gas engines of power plants, ships and locomotives [Turnkey... 1994].

Below, emissions of modern power plant diesel and gas engines are first examined. Required and available purification systems and their performance are then presented.

6.1 Emissions of diesel and gas engines

6.1.1 Pollutant species

Diesel and gas engines emit pollutants when operating. The main toxic components are carbon monoxide (CO), oxides of nitrogen (NO_x), unburned hydrocarbons (HC), sulfur dioxide (SO_2), and particulates. Especially in industrialized countries, rather stringent limits have been set for these pollutants. However, the engine exhaust contains other components, the quantities of which are small, but which may be more toxic than the above mentioned species. Some detailed hydrocarbons and, for instance, nitrous oxide belong to this group. Until now, these pollutants have not usually been regulated by the authorities, and they are therefore called unregulated emissions.

Different engine concepts emit different amounts of these pollutants. In the exhaust of a stoichiometric gas engine, high raw emissions of CO, NO_x, and HC can be found. Utilization of the lean-burn concept in gas engines reduces all these emissions considerably, if the mixture is kept safely within the inflammable limits. Particulates usually form no problem in gas engines. In natural gas driven diesel power plants, the sulfur emissions are negligible [Vuorinen 1995].

Oxides of nitrogen and particulates are the worst pollutant components in the exhaust of heavy or diesel oil driven engines. CO and HC are usually negligible. SO_2 depends on the fuel sulfur content. With the worst heavy oils, the raw SO_2 emissions can be significant.

In recent years, growing attention has been paid to carbon dioxide (CO_2) and other species that are considered greenhouse gases and contribute to global warming. The reciprocating internal combustion engines are rather favorable in this respect, since they emit relatively less CO_2 due to their high thermal efficiency. The combined heat and power production offers further advantages as far as CO_2 minimization is concerned [Koch 1995, Prussnat 1996, Wiese 1996]. As an example, CO_2 emissions of the new gas-driven air CHP plant in the Royal Mail installation are said to be 23.6% lower than for a similar building using a traditional system [Mullins 1996b].

6.1.2 Emissions level

Requirements

The German emissions standards "TA Luft" have been accepted as a reference by most European countries [Chellini 1995a]. These emissions standards require emissions be less than

- 500 mg/m^3 for NO_x,

- 650 mg/m^3 for CO, and

- 150 mg/m^3 for non-methane hydrocarbons (NMHC).

All are given at 0°C, 101.3 kPa; NO_x and CO at 5% O_2, HCs at actual O_2. [Kunberger 1996]. For other units these correspond roughly to values of

- 1.8...2.3 g/shaft-kWh or 220...280 mg/fuel-MJ for NO_x,

- 2.4...2.9 g/kWh or 290...360 mg/MJ for CO, and

- 0.75...0.93 g/kWh or 90...120 mg/MJ for NHMC.

In a Swedish 4 MW_e engine driven CHP plant, the target of the NO_x emissions is below 100 mg/MJ_{fuel} after the exhaust cleaning system [Karlemo 1993]. This is less than half of the limit of the TA Luft. In German CHP plants, NO_x emissions of the engines have had to be reduced from a level of 4000 mg/m^3 (0°C, 101.3 kPa) to a level of below 500 mg/m^3 during the last ten years [Pischinger 1995]. Emissions of 4000 mg/m^3 corresponds roughly to a level of 15-18 g/shaft-kWh or 1.8-2.2 g/fuel-MJ.

Carbon dioxide

In [Wiese 1996], global warming potential of the engine and gas turbine CHP plants has been studied. In the eco balance, direct exhaust emissions were included, as were those formed in the fuel supply. The emissions generated in the power plant manufacturing are said to be small. Of greenhouse gases, only CO_2 and methane were included, others were assumed to be negligible. Effects of methane were converted to correspond to those of CO_2 using proper correction factors. The advantages of the CHP production were taken into account so that the CO_2 emissions of a gas-driven boiler plant - efficiency 95% - were subtracted from the total CO_2-equivalent emissions of engine or gas turbine power plants.

As a result, the engine CHP plants show specific CO_2 equivalent emissions of 290 to 656 g/kWh. The lower limit corresponds to a gas driven power station equipped with a high-efficiency engine. Use of a combined cycle reduces the figures to a range of 271-566 g/kWh. The lowest CO_2 emissions of gas turbine plants are very close to those of engine power stations [Wiese 1996].

Oxides of nitrogen from engines with liquid fuels

Diesel power plants are not very competitive as regards the raw emissions of oxides of nitrogen (NO_x). The 3500 series B diesel models of Caterpillar are said to generate, on average, about 6.7 g of NO_x per output-kWh [Exhaust... 1995]. This corresponds to around a value of 800-900 mg per fuel-MJ. Ref. [Wärtsilä... 1995] gives NO_x emissions values of 10 to 12 g/kWh for medium-speed Wärtsilä diesel engines tuned for best fuel consumption. The emissions level depends on the speed, faster engines emitting less NO_x than slower. Initial test bed results indicate a NO_x emission level of 13 g/kWh for the company's largest medium-speed 64 engine [Wärtsilä... 1996].

With advanced combustion systems and innovative concepts, the engine manufacturers have, however, managed to reduce the raw NO_x emissions further. Mitsubishi Heavy Industries have developed the combustion system in a single cylinder test engine, the cylinder bore of which is 170 mm, the stroke 180 mm, and the output 300 kW [Lanz 1996]. The concept is based on a prechamber combustion system and a stratified fuel-water injection system, but many other NO_x reducing methods are also utilized. After large optimization work, the engine NO_x emissions fell to 88 ppm, the brake thermal efficiency still reaching 39%. No accurate load data are given, however. At full load, the content of 88 ppm would correspond roughly to 1.0...1.1 g/kWh.

According to Lanz [1996], Wärtsilä Diesel of Finland has also been looking for ways of reducing exhaust emissions. By following a systematic approach, the company has come to a solution with which a NO_x reduction of 50% has been achieved with no increase in fuel consumption. Wärtsilä opted for a direct water injection system. It has been tested in a ferry. It is said that NO_x values of 4.5 g/kWh would be obtainable, if the results follow those recorded on the test engine.

Use of the mixture of diesel fuel and fusel oil as fuel seems favorable for most exhaust emissions [Drunk... 1996]. The measured emissions values of the MAN engine in Wilthen, Germany, were mainly lower with this mixture than with pure diesel fuel.

Emissions of SI gas engines

Regarding gas engines, the amount of raw NO_x emissions depends on the combustion concept. So-called lean-burn gas engines exploiting spark ignition (SI) are favorable as to NO_x emissions, since the combustion temperatures are low, and no exhaust catalytic cleaning is necessarily required [Kinnunen 1996, Vuorinen 1995]. Kinnunen [1996] gives the following figures for a medium-speed lean-burn SI gas engine with a bore of 250 mm and without any catalyst: NO_x < 1.0 g/kWh, CO < 2.0 g/kWh, HC < 1.0 g/kWh. According to [Kunberger 1996], the Wärtsilä Pure Energy Plant engines are designed to meet stringent emissions limits without aftertreatment. Wärtsilä says that the 2 MW, 3 MW and 4.5 MW units even achieve less than one-half of the TA Luft levels.

Fig. 21 shows the performance and exhaust emissions for such a Wärtsilä engine, installed in a CHP plant in Lahti, Finland [Lahti... 1996]. The engine is operated with a relative air/fuel ratio of around 2.1. The CO emissions are then somewhat more than 2 g/kWh (not very obvious in the Figure), the non-methane HCs below 1 g/kWh, and the oxides of nitrogen around 1 g/kWh. The NO_x emissions have been guaranteed to be less than 150 mg/MJ_{fuel} and reached easily thanks to the electronic control system [Lahti... 1996].

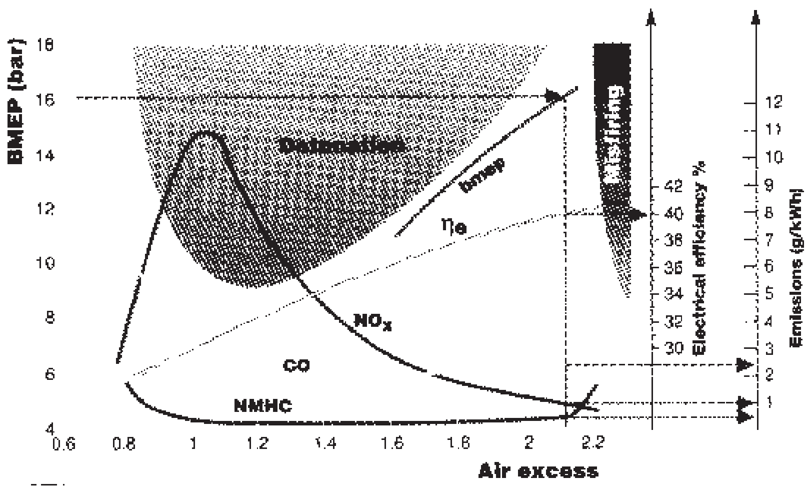

Figure 21. Performance and exhaust emissions of a Wärtsilä lean-burn SI gas engine [Lahti... 1996].

The new generation of high-speed gas engines of the Jenbacher Energiesysteme are said to provide an electrical efficiency of more than 40% without jeopardizing emissions [Cogeneration... 1995]. The engines typically meet emissions values below 500 mg/m^3 NO_x and 300 mg/m^3 CO (at 0°C, 101.3 kPa) with an oxidation catalyst. These figures correspond roughly to values of 2.3-2.5 g/kWh and 1.3-1.5 g/kWh, respectively.

The medium-speed lean-burn SI gas engine GMT A32G of Fincantieri has also been designed to provide low emissions levels, so that it can be operated without a catalytic converter within the TA Luft emissions limits [Chellini 1995a]. The maximum NO_x emissions of the engine are said to be 500 mg/m^3, and maximum CO emissions are 650 mg/m^3 (at 0°C, 101.3 kPa). A small oxidation catalytic converter can be used, if further reductions of CO are required.

A very low exhaust NO_x level of 5 mg/m^3 is given for a CHP plant in Germany, equipped with a low-calorific-value gas driven SI engine [Energy... 1996]. The value has been defined after the oxidation catalyst, but the catalyst does not usually reduce oxides of nitrogen, so the main reason for the low NO_x value is a low combustion temperature of about 900°C.

Gas diesels and dual-fuel engines

In gas diesels (GD), the NO_x level is higher. In a natural gas driven GD power plant in Finland, a catalytic exhaust purification system is required to reduce the NO_x emissions from a level of 1300 mg/MJ (fuel) (ca. 11 g/shaft-kWh) to a target level of 200 mg/MJ, comparable to small gas turbines [A catalyst... 1990]. Westergren [1989] gives a NO_x level of even 1.96 g/MJ$_{fuel}$ for a GD engine at full load.

Fig. 22 shows raw emissions of the dual-fuel engines of the diesel CHP plant MHKW Resse [Röchling 1995]. Operation with mining and natural gas is favorable as regards oxides of nitrogen, but CO and HC emissions are smaller when running on diesel fuel. There are no great differences between mining gas and natural gas as far as raw emissions are concerned. With both fuels, the HC and CO emissions increase with decreasing load, whereas they remain rather constant with DFO. At full load, the CO emissions are at a level of 0.6-0.8 g/m^3, the NO_x emissions at a level of 1.5 to 2 g/m^3, and the hydrocarbons are around 1 g/m^3 (0°C, 101.3 kPa, exhaust O_2 content 5%) when running on gas fuels. In diesel mode, the engine emits some 4.5 g/m^3 of oxides of nitrogen. Other pollutant contents are very small or negligible.

Emissions results with a similar trend have been reported by O'Keefe [1995]. A gas driven engine was inherently better for NO_x in dual-fuel mode than in diesel mode. It also retained that advantage over a range of BMEP values. As in the MHKW Resse, the CO in exhaust was somewhat higher for the dual-fuel mode.

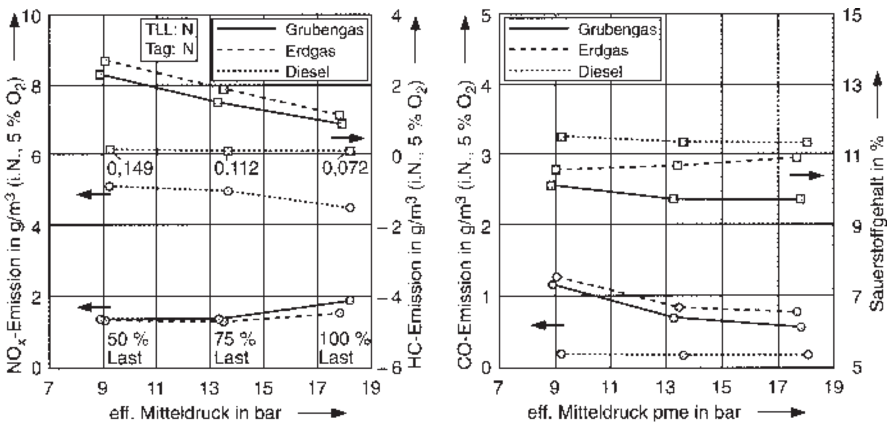

Figure 22. NO_x, HC, and CO raw emissions of dual-fuel engines versus engine load with different fuels [Röchling 1995].

Table 8 shows a summary of raw emissions given for gas driven engines. As can be seen, the lean-burn SI gas engines are very favorable as to exhaust emissions, since the raw emissions of all surveyed engines are below or very close to the TA Luft limits, and no exhaust aftertreatment systems are necessarily needed. Gas diesels emit considerably higher amounts of oxides of nitrogen and do not comply with the TA Luft standards without catalysts. NO_x and HC emissions of the only dual-fuel engine exceed those limits as well, but the CO emissions are rather close to the requirements.

Table 8. Raw emissions of examined gas engines (OC, oxidation catalyst).

Engine type	NO_x	CO	HC	Catalyst
	g/kWh	g/kWh	g/kWh	
SI, lean-burn	< 1.0	< 2.0	< 1.0	w/o
SI, lean-burn	1	2.4	< 1.0	w/o
SI, lean-burn	< 2.3	< 1.3		OC
SI, lean-burn	< 2.3	< 2.9		w/o
GD	11			w/o
GD	16			w/o
DF	5.5...9	2.2...3.6	4	w/o

Oil driven engines emit too high quantities of oxides of nitrogen. Without any reducing methods, the reviewed engines show raw NO_x emissions of 6.7 to around 19 g/kWh. With water injection and a systematic approach, however, the emissions have been reduced to a level of 4.5 g/kWh and - combined with further measures - even down to 1.0...1.1 g/kWh.

6.2 Exhaust purification

There are two main control strategies available to minimize NO_x emissions of diesel and gas engines: to modify the combustion process and to treat the exhaust gas stream [Bretz 1989]. More and more, combustion modifications alone cannot achieve the NO_x emissions limits in place today or planned for the future. Selective catalytic reduction (SCR) and selective non-catalytic reduction (SNR) processes are then required.

In this paper, it was decided to deal only with post-combustion exhaust cleaning systems - including those adopted for NO_x, CO, HC and particulate removal - since in power plants, more efficient methods are required than combustion modifications. Regarding the latter, more information is available in a number of recent publications, e.g., in [Bretz 1989, Collins 1990, Lanz 1996, Miyano 1995, Nakano 1995, Pischinger 1995, Remmels 1995, Schnohr 1995, Tsukamoto 1995, Velji 1995, Yoshikawa 1995].

6.2.1 Requirements of different engines

Lean-burn SI gas engines emit very low levels of pollutants and do not necessarily need any post-combustion methods for exhaust cleaning, as mentioned earlier and reported, e.g., by Kinnunen [1996], Prussnat [1996], and Vuorinen [1995]. A further advantage of these engines is that they can be driven even with gases containing sulfur, chlorine, fluorine, and arsenic, which are said to be harmful for catalysts [Prussnat 1996]. Landfill and sewage gases are mentioned as examples of gaseous fuels containing catalyst poisons in [Lean-Burn... 1995]. Nevertheless, oxidation catalysts have been incorporated, e.g., into the CHP plants Am Kranzberg and Haar in Germany, although lean-burn natural gas driven SI engines are used as prime movers [Koch 1995, Juptner 1996].

Consequently, the development in Germany has generally led to a larger use of the lean-burn concept in gas engine driven power stations and SCR technology in diesel and dual-fuel engine installations [Pischinger 1995]. Earlier, more three-way catalysts (TWCs) were adopted in gas engine plants.

Prussnat [1996] mentions that TWCs are used up to an output of 300 kW_e. There are TWCs available, for which a lifetime of 16,000 h is guaranteed. In optimum use, pollutant levels remain below the German restrictions even for a period of 40,000 h [Rumpel 1996]. In the new natural gas driven CHP plant in Hochheim, Germany, a TWC has been installed into the exhaust system of the 50 kW_e gen-set [Every... 1996]. The CO and NO_x emissions are said to be at a level of 50% of the limits given in the emissions standards of TA Luft.

TWCs can not, however, be adopted if landfill gas or gas from sewage disposal plant is burned as fuel, since the latter contain harmful species [Prussnat 1996]. Even Rumpel [1996] mentions that problems have been reported with the catalysts in some cases when gas from a sewage disposal plant has been burned.

SCR catalysts are said to be profitable only above an approximate output of 2.5 MW. [Prussnat 1996] states that in the middle range - between 300 kW and 2.5 MW -, there are even gas diesel (most probably dual-fuel, not gas injection; author’s note) engines which comply with the German legislation even without any SCR catalyst, if the pilot fuel quantity is below 1%.

6.2.2 Cleaning systems

Jenbacher Energiesysteme has presented two new systems for a further reduction of exhaust NO_x and CO emissions [Cogeneration...1995]. The systems are independent of the engine combustion. The first system combines the lean-burn gas engine with an SCR catalytic converter and makes it possible to reduce exhaust gas NO_x values to 100-50 mg/m^3 (0°C, 101.3 kPa) still leaving freedom to optimize the engine efficiency and output. The injected urea (40% solution) acts as a reduction agent in the converter. It is transformed into ammonia at temperatures of 480 to 500°C, and reacts with the oxides of nitrogen to form nitrogen, steam and CO_2.

The other system is a regenerative heat exchanger with two reaction chambers and a switch mechanism [Cogeneration... 1995]. The exhaust gas flows at a temperature of ca. 530°C into the first storage medium of the reaction chamber and is heated to around 800°C. CO and HC form here CO_2 and H_2O by reacting with the available oxygen. The exhaust gas releases the added thermal energy for further use as heat in flowing through the second storage medium. It then enters the exhaust line at 550 to 570°C.

The exhaust flow is reversed after a few minutes so that it can take the heat from the second storage medium back to the first one [Cogeneration... 1995]. The energy requirement is minimized that way. The thermal losses are almost completely compensated for by the post combustion of carbon monoxide and hydrocarbons. The system is designed to reduce CO and HC emissions to values of about 150 mg/m^3 (0°C, 101.3 kPa) each. In addition, formaldehyde is also reduced considerably. The system is said to be particularly suitable for critical gases, such as sewage and landfill gases.

The turnkey exhaust gas cleaning systems of Ref. [Turnkey... 1994] consist of a selectively acting double catalyzer - SCR catalyzer and oxi-cat - in honeycomb form that can be used for NO_x reduction and for oxidation of CO and HCs. Conversion rates of 90-99% are said to be obtained, in a temperature range of 250-550°C and at a pressure loss of ca. 2 mbar. Ammonia or urea dissolved in water is adopted as reaction agent. It is applied at a dosage in which the mass flow is exactly stoichiometric to the NO_x/SO_2 mass flow. The cleaning system concepts have been differentiated between the type of fuel, since diesel plants have practically no HC components in the exhaust while gas/diesel plants produce less soot and SO_2 [Turnkey... 1994].

The Cat DeNO_x concept combines several emissions-reducing technologies into a single package, as well [Exhaust... 1995]. Fuel-grade ethanol is injected into the hot exhaust stream at the head of the catalytic converter/muffler. The injection pressure is 4.0 bar. The injectors are protected against high exhaust temperatures by routing engine coolant around them. In the presence of the catalyst, the ethanol and oxides of nitrogen combine to form water vapor, nitrogen and carbon dioxide. Based on the tests, NO_x levels of down to 1.3 g/kWh can be sustained (ca. 150-200 mg/MJ fuel) [Exhaust... 1995].

In the diesel CHP plant MHKW Resse, a SCR and oxidation catalyst has been installed between the turbocharger turbines and exhaust boiler for each engine. To reduce oxides of nitrogen, NH_3-water solution (25%) is injected into the exhaust before the SCR catalyst. After the oxidation part, the exhaust contains less than 500 mg/m^3 of NO_x and 150 mg/m^3 of CO in all operation modes (0°C, 101.3 kPa, exhaust O_2 content 5%), corresponding to a NO_x level of below 1.8-2.3 g/kWh and CO level of below 0.5-0.7 g/kWh.

In the NO_x removal system of Westergren [1989], NH_3-water mixture is injected into the exhaust gases of a GD engine immediately after the silencer. It is said that a mole fraction of 0.9 is enough for an 80-85% reduction of NO_x. The NH_3 consumption would thus be 39 kg/h for a 7.1 MW_e plant, and the water consumption 117 kg/h, since NH_3/water ratio is 1/3. The authors state that with a reduction level of 80-90%, the purified exhaust gas would contain130-250 ppm or 100-300 mg/MJ NO_x (0.8-2.5 g/shaft-kWh).

Regarding removal of NO_x and SO_x, O'Keefe [1995] mentions a zeolite-based high-temperature catalyst as an example of recent developments. The honeycomb structure is homogeneous. It is said to prevent harm from thermal cycling and particle abrasion. Bretz [1989] mentions that a zeolite-based molecular-sieve SCR reactor reduces the exhaust NO_x levels from 850 ppm in dual-fuel mode and from 1400 ppm in diesel mode to below 280 ppm for both modes in a cogeneration engine-powered plant in Germany. The figures are, however, based on the information of the system supplier.

Another new concept comprises a downstream circulating-fluidized-bed reactor to remove NO_x and SO_x. The reactor requires supplementary fuel, which can be inexpensive petroleum coke, to increase the incoming exhaust temperature to 870°C. The reactor is reported to be able to reduce NO_x and SO_x to below 13 mg/MJ and unburned hydrocarbons to below 2 ppm. The figures are based on an assumption that the exhaust of the diesel engine running on diesel fuel contains 800 to 1400 ppm NO_x and 200 to 300 ppm unburned hydrocarbons, and that residual oil produces 700-800 ppm SO_2 in addition [O'Keefe 1995].

The Wärtsilä systems for removal of CO and unburned hydrocarbons rely on either boiler-furnace combustion or some type of duct burner ahead of the boiler. A regenerative heat exchanger is mentioned as a typical example of a variant combustion device. The two reaction chambers ignite combustibles and store the waste heat, alternating the order every few minutes [O'Keefe 1995].

According to O'Keefe [1995], oxides of nitrogen of the diesel exhaust gas are decreased by reburning and by dilution when the gases are burned in duct burners and down-stream boilers in combined-cycle plants. Reduction figures of 50 to 70% are reported.

A post combustion cleanup system is used to control SO_2, NO_x and particulate emissions of the diesel/coal combined-cycle plant, presented by Smith [1994]. The purification system has a dry lime scrubber and baghouse. A selective catalytic (or non-catalytic) reduction system is adopted to control oxides of nitrogen.

Catalytic silencers have been applied to very small stationary engines in addition to vehicular ones [Bretz 1989]. They consist of small spheres in an annular basket. The spheres, which are porous inorganic substrates, are impregnated with noble metals - platinum, palladium and rhodium.

According to Bretz [1989], the concept is also applicable to larger stationary engines, because the same catalyst metals are used in more traditional ceramic honeycomb or stainless steel grid substrates and the chemical reactions are the same. For the catalyst to be effective, the following operating parameters are, however, necessary:

* Exhaust gas temperature must be between 430 and 650°C.

* Stoichiometric or slightly rich mixtures must be burned, since sufficient CO must be available to react with oxides of nitrogen.

* Free oxygen must be present at less than 0.5% in a mole basis [Bretz 1989].

As can be seen, these catalytic silencers can come into use in such power plants only where stoichiometric gas engines are used as prime movers.

Ref. [Collins 1990] recognizes the great advantages of the SCR method in NO_x reduction and states that the NO_x content can be limited to less than 100 ppm in most cases. The following drawbacks of the system are, however, listed:

1) The temperature range is limited to 320-400°C. At lower temperatures, the rate of reaction is too slow for effective removal. Above 430°C, ammonia is oxidized to form NO_x. Additionally, permanent damage occurs to some catalysts.

2) Ammonia is a hazardous substance. Potential for accidental release must be taken into account. Stack emissions of unreacted ammonia have proven a problem for some gas-turbine-based plants using SCR.

3) Catalyst disposal can be problematic. SCR catalysts can contain heavy-metal oxides, which are considered hazardous.

According to Bretz [1989], spent zeolite-based molecular-sieve catalysts are non-hazardous and safely disposable or recyclable as ceramic filler, since there are no heavy metals in the molecular-sieve body.

In marine installations, the size of an SCR unit forms one problem. Nevertheless, Wärtsilä has added an SCR into the machinery of a ferry operating between Sweden and Denmark [Lanz 1996]. The system was seen as feasible, since a much smaller unit could be used due to the reduced raw NO_x emissions of the engine (around 4.5 g/kWh). The "compact SCR" combines the functions of an SCR and silencer into one unit.

7 CONCLUSIONS

1) The market for gas and diesel engine power plants has grown strongly particularly during the ’90s. It can be estimated that total orders for reciprocating engines were at the 5000 unit level during a survey period from June, 1995, to May, 1996. This means a total ordered output of around 10 GW.

In industrialized and cold-climate countries, combined heat and power (CHP) production is preferred. In Germany, e.g., the number and output of engine driven CHP plants have increased rapidly.

Diesel-electric marine propulsion has become a good market for diesel generating sets, as well. Recently, even gas driven engines have been installed in marine equipment.

Advanced small-scale gas turbines are seen as the main competitors for reciprocating engines at least as long as gas is available as fuel.

2) Continually, most reciprocating engine power plants operate with diesel or heavy fuel oil. Different fuel alternatives have, however, been widely investigated. Natural gas is obviously the most important alternative. Liquid petroleum gas, gas from sewage disposal plant, landfill gas, other biogases, and low calorific value gases (e.g., from wood) are listed as further alternatives. Fusel oil and pyrolysis oil are mentioned as liquid alternatives. Use of coal as engine fuel is also being studied.

3) Engines with a speed of above 1000 rpm clearly form the largest category of the ordered power generating engines. Nevertheless, large two-stroke low-speed, as well as four-stroke medium-speed engines are being installed particularly for base load operation.

Recently, the proportion of different gas burning engines has increased. Special effort has been put into the development of lean-burn spark-ignited (SI) gas engines, since the raw exhaust emissions of these engines can be reduced to a very low level. Additionally, small stoichiometric SI gas engines and large dual-fuel and gas injection engines are used.

4) Combined cycle technology is also developed in connection with reciprocating engine power systems. Excess exhaust energy is utilized in turbo compound systems to increase the electric output of the plants. Steam is produced in exhaust boilers for steam turbines for further increase in power and efficiency. Supplementary firing in boilers is investigated in order to find a good balance between the state of steam and cycle efficiency. New concepts are being developed, including, e.g., combustion of coal in the burners of exhaust boiler.

In CHP plants, as efficient as possible means for heat recovery are being sought. In the most advanced systems, the engine exhaust is cooled below the dew point in the boiler, and even the convective and radiative heat of the engine is recovered.

The very efficient waste heat utilization increases the total efficiency to very high a level. On the other hand, specific heat consumption seems generally to decrease, whereas the specific power consumption increases. Therefore, it is important to develop systems with high power-to-heat ratios.

5) The highest power generation efficiencies given for the reciprocating engines in the surveyed material reach 54%. The prerequisite is that some kind of combined cycle is adopted. In a power plant with two-stroke HFO engines, actual electric efficiencies of 45 to 47% have been measured. The plant comprises turbo compound systems and steam turbines driven with exhaust energy of the engines.

Generally, oil driven engines seem to be more energy economic than gas driven. Concerning the largest gas diesels, high shaft efficiencies of slightly below 50% are promised. It is, however, difficult to evaluate the actual electric production efficiency, since the gas compression consumes more or less power, depending on the gas pressure in the supply line. The only given efficiency results show an average efficiency of 39% for a GD plant comprising, however, even a steam turbine.

With the largest lean-burn SI gas engines, peak electric efficiencies of around 42% seem to be achievable in single cycle plants.

The efficiencies of even gas driven reciprocating engines seem, however, to be competitive with those of gas turbines, say, at least below an output of 10 MW. The great advantage of the engines over gas and steam turbines is that the efficiency remains at a rather high level down to a part load of ca. 40%.

6) Spark-ignited gas engines utilizing lean-burn technology show very favorable exhaust emissions results. Without any post-combustion methods, emissions of oxides of nitrogen are said to remain below or equal to 1 g/kWh at their lowest, carbon monoxides 2 g/kWh or less, and hydrocarbons below or equal to 1 g/kWh. No actual results were, however, reported in the surveyed articles.

Gas diesels, dual-fuel engines and diesel engines are not as successful as to the oxides of nitrogen. These engines do not comply with the current emissions standards of TA Luft without exhaust aftertreatment. Nevertheless, with comprehensive measures, including direct water injection, very low NO_x figures of even 1.0 to 1.1 g/kWh were reported for a diesel engine with a bore of 170 mm. For a larger medium-speed diesel, a NO_x level of 4.5 g/kWh was given.

Reciprocating engines are usually found to be favorable regarding CO_2 emissions, since the shaft efficiencies of the engines are generally higher than those of other prime movers. Concerning CHP production, there is, however, no noteworthy difference between gas engines and gas turbines in CO_2 production according to a reported eco balance.

7) There are a lot of different exhaust aftertreatment systems available for diesel and gas engine driven power plants. Three-way catalysts, known from automotive engines, are common in small plants below an output of 300 kW.

Above 2.5 MW, selective catalytic reactors (SCR) are widely used to reduce oxides of nitrogen. Ammonia or urea dissolved in water is commonly adopted as reaction agent, but even fuel-grade ethanol is exploited. Very often, some kind of oxidation catalyst has been connected to the SCR reactors. Sometimes, silencers are also incorporated into the same package. Boiler-furnace combustion and duct burners are also used for oxidation of CO and HCs.

Use of a downstream circulating-fluidized-bed reactor represents a more advanced exhaust purification system. In a diesel/coal combined-cycle plant concept, the cleaning system has a dry lime scrubber and baghouse in addition to the SCR reactor.

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