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Gas Turbines above 150 MW for Integrated Coal Gasification Combined Cycles (IGCC)

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ABSTRACT

Commercial IGCC power plants need gas turbines with high efficiency and high power output in order to reduce specific installation costs and fuel consumption. Therefore the well proven 154 MW V94.2 and the new 211 MW V94.3 high temperature gas turbine are well suited for this kind of application. A high degree of integration of gas turbine, steam turbine, oxygen production, gasifier and raw gas heat recovery improves the cycle efficiency. The air used for oxygen production is taken from the gas turbine compressor. The N_2-fraction is recompressed and mixed with the cleaned gas prior to combustion. Both features require modifications of the gas turbine casing and the burners. Newly designed burners using the coal gas with its very low heating value and a mixture of natural gas and steam as a second fuel are developed for low NO_x and CO emissions. These special design features are described and burner test results presented.

INTRODUCTION

Over the past two decades, the efficiency of the gas turbine has been improved and its reliability verified, with the result that it has gained considerable significance as a means of generating electricity. Where natural gas or fuel oil are used, the most economical type of power plant is either a gas turbine alone, as a peak load plant, or a combined gas and steam (GUD) cycle as an intermediate and base load plant. By dividing the process into two temperature ranges, 1100 to 550°C in the gas turbine and 530°C to ambient temperature in the steam turbine, high overall efficiencies exceeding 50% are now being achieved [1]. Even with the most complex configurations, the straight steam cycle power plant will never be able to reach this level.

For that reason the gas turbine has gained a significant fraction of the market in recent years. The gas turbine is dependent on the availability of low cost, clean fuel because of the direct combustion of the fuel in the compressed air. The chief objective of the coming decade is utilization of coal for the combined cycle process. At present both direct combustion in a pressurized fluidized

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bed and gasification and cleanup of fuel gas prior to combustion are being investigated simultaneously to achieve this end.

NEW IGCC DEMONSTRATION POWER PLANTS

Since the plants Cool Water and Dow Chemical, Plaquemine in the USA verified the suitability of the integrated coal gasification combined cycle (IGCC) several years ago, several large European utilities are now showing interest in this new technology. In Europe, fuel costs have always been relatively high. For this reason, plant efficiency is a far more important consideration here. Furthermore, German coalfired power plants have been fitted with flue gas desulphurization and DeNOx plants at high capital costs. Any new plant designs to be installed are therefore expected to meet, preferably to even surpass, the emissions performance achieved in these plants.

The largest German utility, RWE, operates primarily lignite-fired power plants in the base load range. As shown in the diagram in Fig. 1, published a year ago, the development

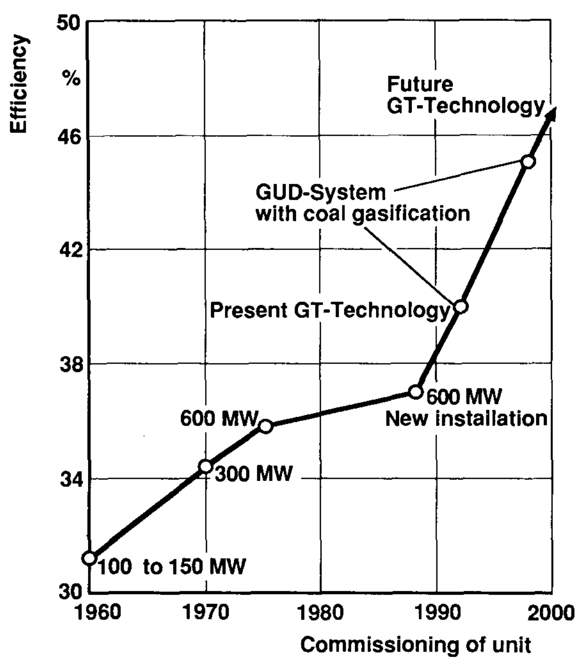

Fig. 1 Net Efficiency of RWE-Bituminous Coal Power Plants

of this configuration as a straight steam cycle power plant over the past decades has culminated in a 600 MW unit with about 38% net efficiency (based on the lower heating value) [2]. Since further improvements in steam cycle efficiency by increasing pressure and temperature are possible, but require substantial capital investments and longer start-up times, the present design can be regarded as fully optimized in economic terms.

Further significant efficiency improvements can only be achieved by utilizing the combined cycle process. Even implementation of today's standard gas turbines would substantially improve these performance levels. Since the 300 MW "Projekt KoBra" demonstration power plant calls for full exploitation of the efficiency potential of the increased turbine inlet temperature, RWE opted for the newly developed 211 MW Siemens V94.3 gas turbine.

In the neighboring Netherlands, the decision to build a 283 MW hard-coal-fired demonstration power plant was made some six months earlier. In this case, proven technology was given a high priority and the Siemens V94.2 gas turbine was selected. The V94 has been in commercial service since 1974 and by September 1990, 37 V94 gas turbines had begun operation in 16 power plants, amassing some 400,000 operating hours. The Buggenum IGCC power plant (in Dutch KV-STEG) will be built by Demcolec B.V. adjacent to the existing "Maascentrale" (Maas power station) at Buggenum, near Roermond, Province of Limburg in the south of the Netherlands. Start-up and operation are scheduled for mid 1993. During the demonstration period, which will span about three years, the IGCC plant will be tested under different conditions and with various types of coal.

GAS TURBINES

Table 1 lists the thermodynamic data of the two gas turbine designs under standard ISO conditions with natural gas fuel. Development of the new gas turbine model series has been described in several publications [3], [4]. Since the V64.3 model is smaller than the V94.3 by a factor of 1.8, but thermodynamically similar and its efficiency and output potential have been demonstrated in both test-bed and commercial power plant operation, it was possible to increase the the performance specs of the 50 Hz turbine over those previously published.

1	Type		V64.3	V84.2	V84.3	V94.2	V94.3
2	Frequence	s^{-1}	50/60 1)	60		50	
3	Turbine inlet temperature according to ISO 2314	°C	1120	1040	1120	1050	1120
4	Power output at terminals (ISO-Base load) 2)	MW	60	103	139	150	200
5	Pressure ratio	-	15.6	10.9	15.6	10.9	15.6
6	Exhaust mass flow	kg/s	187	355	420	509	605
7	Exhaust temperature	°C	534	532	534	538	534
8	Efficiency at terminals (ISO-Base load) 2)	%	35.2	33.8	35.7	33.8	35.7

1) With gear (Gas turbine: 90 s^{-1})

2) ISO-Conditions, $\Delta p_{Vi} = \Delta p_{TH} = 0$

Fuel : Natural gas

Tab. 1 Thermodynamic Data

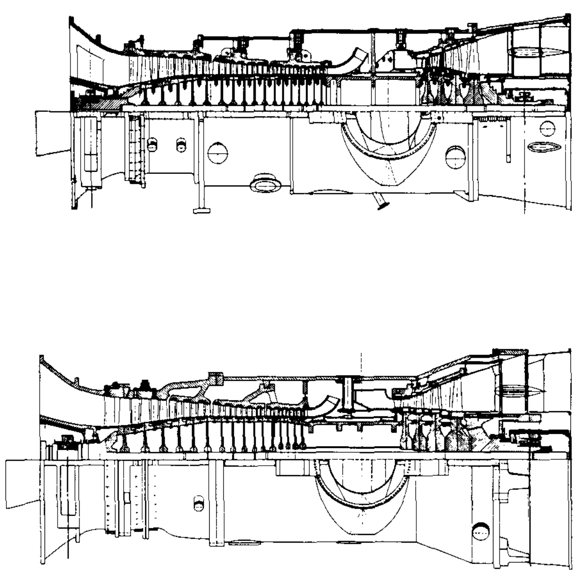

Fig. 2 Comparison of Longitudinal Cross Sections

Figure 2 compares the longitudinal sections of the V94.2 and the V94.3 two gas turbines. The distance between bearings is virtually identical but the blading diameter of the higher-output V94.3 was increased by about 10 % to allow efficient exploitation of the increased enthalpy difference. To achieve the pressure ratio increase in the compressor from 10.9 to 15.6, it was therefore only necessary to adjust the number of compressor stages from 16 to 17. In the case of the turbine, the previous four stages represented an optimum.

Since the new gas turbine is primarily envisaged for GUD power plant applications, particular emphasis was placed on high part load efficiency. Four rows of variable-pitch compressor blade rows ensure a wide range of mass flow adjustment, making it possible to maintain a constant exhaust gas temperature from approx. 50% load to base load. In a power plant with at least two gas turbines it is therefore possible to maintain a constant steam temperature from 100 to 25 % load, since one of the gas turbines can be shut down after simultaneously reducing the load of both to 50 %. The other continuing operation with in-let blading fully open at 100 % power output, has still a further 50% power reduction reserve.

Adjustable pitch stator blading and three bleed points (as compared to the previous two of the V94.2) ensure unproblematic start-up of the compressor, which has a very high pressure ratio for a single-shaft axial compressor.

By careful selection of the extraction pressures it was possible to optimize the supply of cooling air to the turbine blading. In concrete terms, the second row of turbine vanes is supplied from the last extraction point and the third row from the middle extraction point. Two different cooling air flows were selected for supplying the rotor. Whereas the first row of both the stator and rotor blading is supplied with precooled air, subsequent stages are cooled with air extracted downstream of the 13th compressor stage.

The pressure drop of this flow which travels from the perimeter inward, in opposition to centrifugal force, is minimized by long bores oriented oblique to the circumference. A cooling system design of such complexity is surely only possible where single disks and hollow shaft elements are used.

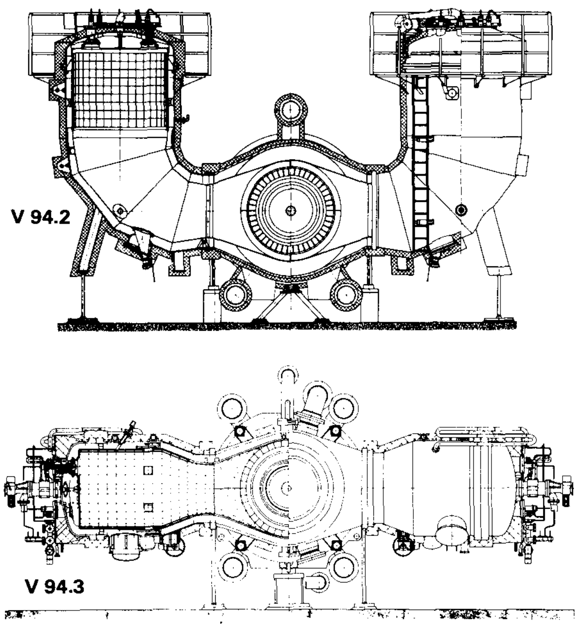

Fig. 3 Comparison of Combustor Cross Sections

Figure 3 compares the combustion chamber arrangement. The number of burners of the V94.3 (2x8 = 16) corresponds to that of the V94.2. Owing to the higher pressure it was possible to slightly reduce the flame cylinder diameter in spite of the higher burner power. At base load the average flame temperature is kept so low, for reasons of NO_x reaction kinetics, that no dilution air other than cooling air is required to achieve a mean gas temperature of about 1300°C at the turbine inlet. This made it possible to reduce the combustion chamber volume and simplify geometry.

COMBUSTOR DESIGN FOR COAL GAS

In high-output boilers for steam turbines it has always been standard practice to work with high residence times, i.e. with large combustion chamber volumes, in order to reduce CO emissions. This was especially important when gas turbines for blast-furnace gas were constructed in Europe in the earaly sixties [5]. The coal gas generated during steel production has a heating value which is approximately 5 percent of that of natural gas. The combustion temperatures are accordingly lower, thus delaying the reaction of CO with O_2 to form CO_2 and rendering good burn-out impossible in the small can-type combustion chambers of an aircraft derivative engine.

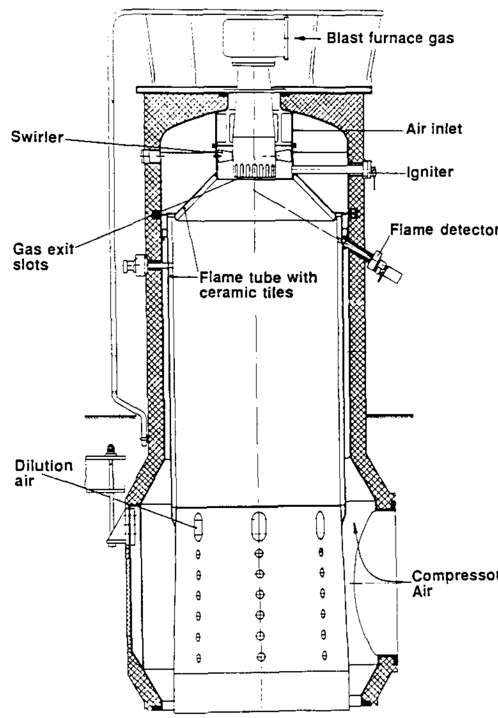

Fig. 4 Combustion Chamber for Low-BTU Gas

Fig. 4 shows the combustion chamber of a 7 MW gas turbine which was developed for blast-furnace gas and is still functional after 83,000 hours of operation. This combustion chamber was fitted with a ceramic lining of the type used in present-day turbines, with the tiles suspended in such a way that thermal expansion and contraction are permitted. The immediate vicinity of the burner is also fitted with suspended ceramic tiles which form a barrier between the metal wall behind them and the radiant heat of the hot gases. There are no instances of a hole being burned through a flame cylinder protected in this manner.

As the development of gas turbine technology has progressed, there has been a step by step increase in the maximum output. Maintaining the principle of one burner in one combustion chamber means that the burner becomes extremely large, resulting in long residence times in the near stoichiometric range and thus in increased No_x formation. This effect can be minimized by using several burners per combustion chamber, which also permits much more even temperature distribution [6].

With the exception of one manufacturer whose gas turbines have burners with thermal outputs ranging from less than 1 MW to more than 400 MW [7], the typical thermal output per burner in gas turbines with electrical outputs of over 30 MW is between 15 an 30 MW. Different gas turbine outputs are achived by fitting various numbers of similar burners, thereby making use of the results achieved in the development of individual components. Consequently, each of the 16 V94.2 burners has a thermal output of 28 MW.

The thermal output of the old blast-furnace gas burner is 24 MW. If one allows for the increase of pressure ratio from about 4 to 11, thermal output can be increased without changing burner dimensions. On the basis of this comparison it is evident that the design of burners developed some 30 years ago does not differ significanly from coal gas burners of today's large gas turbines.

BURNER DESIGN FOR SEVERAL FUELS

A large percentage of the V94.2 gas turbines now in service are equipped with burners (Fig. 5) which permit the following three modes of operation:

1. natural gas diffusion burner operation
2. natural gas premix burner operation
3. fuel oil diffusion burner operation.

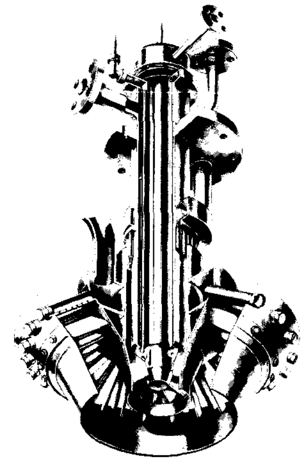

Fig. 5 Hybrid Burner for Low NOx and CO Emissions

Switchover from one mode to another can be made without interruption of the plant process. Natural gas and fuel oil can be combusted simultaneously in diffusion burner operation in nearly all possible combinations (natural gas > 20 %, fuel oil > 10 %). Natural gas premix burners achieve NO_x concentrations below 25 ppm without water or steam injection [8]. H_2O is used with diffusion burners to control NO_x emissions.

For the IGCC plants, this burner was to be equipped with a coal gas injection system in addition. Since the fuel has a very low heating value (LHV approx. 4300 to 5000 kJ/kg), a fuel volume is required which is equivalent to 47 % of the primary air which passes through the outer swirler during base load operation. This made modification of the air flow patterns around the central diffusion nozzle system unavoidable. Furthermore, at the high H_2 concentration of the coal gas, spontaneous combustion of the air and fuel would occur on premixing. It was therefore necessary to arrange the coal gas nozzle system such that ignitable fuel-air mixtures cannot form near the walls.

The design developed with these parameters (Fig. 6) has an additional concentric ring of coal gas nozzles which open into the primary air duct downstream of the outer swirler. Since localized recirculation occurs in this region during natural

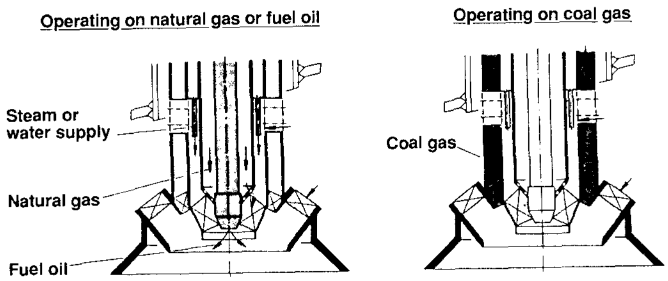

Fig. 6 Burner for Coal Gas and Conventional Fuels

gas operation, air-natural gas -premixing is not performed for safety reasons. The additional coal gas duct does not, however, influence operation of diffusion burners with natural gas or fuel oil. This burner is therefore suitable for operation with three fuels, even any two simultaneously. In the Buggenum plant, the fuel oil system is not required. In the KoBra plant, fuel oil will possibly be the second fuel. It had to be demonstrated by 1 July 1990 that these burners can operate with coal gas reliably and in a manner which is benign to the environment. For gas turbine start-up and for achieving instantaneous load rejection to station service load, the alternative fuel natural gas may be used. The aim here is, however, to accommodate these operating conditions under operations with coal gas alone.

RESULTS Of BURNER TESTING

Investigations with the newly developed coal gas burner were conducted on a low pressure burner test rig (Fig. 7) at the Mülheim Works at atmospheric pressure. This test equipment permits preheating the combustion air to 230 °C to investigate the effects of elevated temperatures in the gas turbine on stability and emissions. Considerable effort was required to provide the equipment for the supply of coal gas to the test rig. This gas, synthesized from CO, H_2 and N_2, is stored in a 95 m^3 vessel. Maximum

Fig. 7 Coal Gas Burner and Test Rig

vessel pressure was 45 bar. This vessel pressure and volume did limit burner operation to around 40 minutes at an excess air ratio equivalent to base load. Gas volumes required for filling the vessel, in particular CO, could only be delivered once a week within Western Europe.

Tests were conducted both with and without air preheating. It was demonstrated that the coal gas burner can be operated from gas turbine start-up, through idling and up to peak load stably and reliably. Natural gas or propane is required only for burner ignition since the spark igniter used is located in front of the nozzles of the central natural gas diffusion burner.

Other special measures are, however, necessary to accommodate load rejection. When the gas turbine is suddenly isolated from the grid, trip should not be initiated, but operation continued under the control of the speed governor to cover plant auxiliary requirements. Before this new steady-state conditions is achieved, non-steady-state speed increases occur which are limited and controlled by reducing the fuel flow to below the amount required for idling. Under such conditions the air flow in the combustion chambers slightly increases and the fuel flow is reduced to about 10 % of the base load flow rate. Excess air in the primary zone reaches extremely high levels and there is a risk that flame out will occur, and the flame monitors will then initiate trip.

Tests with the coal gas burner have demonstrated that the stability range of the coal gas flame can be extended significantly by also feeding coal gas to the centrally located natural gas nozzles. Results of tests conducted at 1 bar are not adequate, however, for making definitive predictions on the effectiveness of the process envisaged for implementation in the power plant. For this reason, alternative plans call for injection of a small volume of natural gas or other fuels with high heating value on load rejection. Many V94.2 gas turbines plant tests have repeatedly demonstrated that load rejections can be accommodated with a natural gas flame.

When operating in the gas turbine load range the burner can be switched over from coal gas to natural gas or vice versa in less than five seconds. This permits continued operation of the turboset at load and operating speed on loss of coal gas supply or shutdown at a rate which minimizes life-limiting effects on the plant without tripping.

Results of NO_x measurements taken during tests at atmospheric pressure with air preheating are compared with calculated reaction kinetics data in Fig. 8. Measured NO_x levels were achieved without the addition of water or steam for reduction. Emissions measured with synthetic coal gas (3846 kJ/kg) were below 2 ppm, thereby in the range of the zero drift of the instrumentation equipment used. NO_x levels in a V94 at a much higher pressure can therefore be expected to be below 20 ppm.

Carbon monoxide emissions of the burner were also very low. In spite of the comparatively low heating value of the coal gas less than 10 ppm CO were measured at excess air ratios equivalent to 40 to 100 % gas turbine load. When used in the gas turbine, the low-emissions operating range of this burner is more likely to become even greater due to the increased pressure level.

Investigations included varying coal gas composition. Particular attention was directed to operation with a nearly nitrogen and steam-free clean gas with a heating value of about 9960 kJ/kg. Gas of this composition

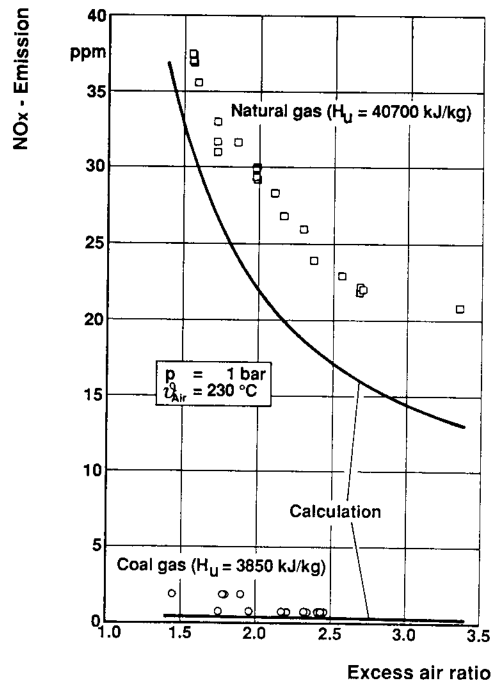

Fig. 8 NO_x-Emissions of the Coal Gas Burner in Atmospheric burner Test Rig

is produced in Buggenum on gasifier start-up and during certain faults when the admixture of the N_2 fraction produced by the air separation unit or the coal gas saturator has not yet begun operation. This high heating value coal gas can also be reliably combusted under all operating conditions.

These investigations have demonstrated that the requirements specified by Demkolec B.V. in the letter of intend have been satisfied on schedule. In the meantime, the contract for the supply of the gas turbine generator has been awarded.

INTEGRATION OF THE GAS TURBINE INTO THE IGCC POWER PLANT

The air separation unit in the Buggenum plant is supplied with compressed and recooled air from the gas turbine compressor during normal operation. The gas turbine must therefore be equipped with an air bleed point located on the lower part of the outer casing (Fig. 9).

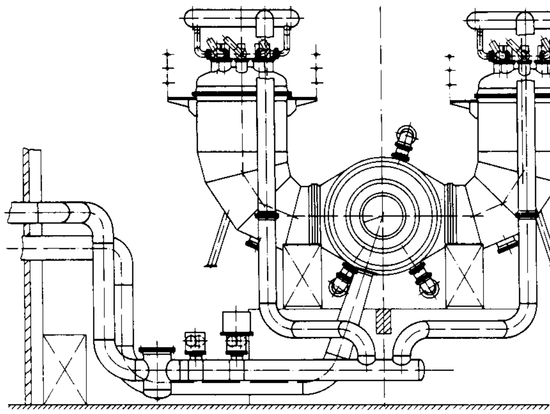

Fig. 9 V94.2 Buggenum, Coal Gas Piping

In the Buggenum plant gas turbine and steam turbine have only one common generator. The compressor end is rigidly coupled to the high pressure stage of the steam turbine by an intermediate shaft which passes through the air intake duct.

In the KoBra plant, an existing steam turbine will be used in the combined cycle. Details of the coal gasification system designed by Rheinbraun are not yet published.

CONCLUSION

Two IGCC power plants will be built in the next two years in the Netherland and Germany with efficiencies of 43 to 46 %, these plants will operate more efficiently than conventional lignite or hard-coal fired plants.

Extremely low environmental impact will be achieved because of the very efficient desulphurization and cleanup of the coal gas in the gas clean-up system. This fuel will be free of fuel-bound nitrogen and because of the low combustion temperaure only small amounts of thermal No_x will be formed. Due to the long residence time in the primary zone, the low combustion temperature will not lead to formation of significant amounts of CO in the normal operating range of the combined cycle.

The successfull demonstration of the 283 MW Buggenum ICG-GUD block with the 154 MW V94.2 from 1993 onward and of the 300 MW KoBra IGCC block with the 211 MW V94.3 from 1995 onward will provide a decisively important impulse towards the adoption of coal gasification for central power station generation. Just as combined cycles have already replaced straight steam cycles when noble fuels i.e. natural gas or distillate oils are utilized to generate electric power, they are likely to become predominant also when coal is the fuel, because integrated gasification into a combined cycle yields higher operating efficiency as well as the most environmentally benign conversion of coal to electricity.

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