AT0100381

15th International Plansee Seminar, Eds. G. Kneringer, P. Rödhammer and H. Wildner, Plansee Holding AG, Reutte (2001), Vol. 1

Externally Fired Gas Turbine Cycles with High Temperature Heat Exchangers Utilising Fe-based ODS Alloy Tubing

Fredrik Olsson*, Sven-Åke Svensson*, Roddy Duncan+

*Sycon Energikonsult AB, Malmö, Sweden

+Mitsui Babcock Energy Limited, Renfrew, Scotland, UK.

Summary

This work is part of the BRITE / EuRAM Project "Development of Torsional Grain Structures to Improve Biaxial Creep Performance of Fe-based ODS Alloy Tubing for Biomass Power Plant". The main goal of this project is to develop heat exchanger tubes working at 1100°C and above.

The paper deals with design implications of a biomass power plant, using an indirectly fired gas turbine with a high temperature heat exchanger containing Fe-based ODS alloy tubing. In the current heat exchanger design, ODS alloy tubing is used in a radiant section, using a bayonet type tube arrangement. This enables the use of straight sections of ODS tubing and reduces the amount of material required.

In order to assess the potential of the power plant system, thermodynamic calculations have been conducted. Both co-generation and condensing applications are studied and results so far indicate that the electrical efficiency is high, compared to values reached by conventional steam cycle power plants of the same size (approx. 5 MW_e).

Keywords

Biomass, efficiency, PM2000, ODS alloy, indirectly fired gas turbine.

1. Introduction

There is presently a strong commitment, in Europe and in other parts of the world, to renewable energy generation as a means to reduce CO_2-emissions. Technologies and efforts for developing biomass-fuelled plants with higher energy conversion efficiencies are, hence, essential.

15th International Plansee Seminar, Eds. G. Kneringer, P. Rödhammer and H. Wildner, Plansee Holding AG, Reutte (2001), Vol. 1

Advanced, large-scale indirectly fired Combined Cycle Gas Turbine (CCGT) systems offer overall energy conversion efficiencies of 45 % and above, compared with approximately 35 % for conventional biomass steam plants. However, to attain this level of efficiency in an indirectly fired CCGT, it is required to develop a heat exchanger capable of operation at temperatures and pressures of above 1100°C and 15 – 30 bars, respectively.

Hence, a project group[1] has been formed to conduct the BRITE / EuRAM Project "Development of Torsional Grain Structures to Improve Biaxial Creep Performance of Fe-based ODS Alloy Tubing for Biomass Power Plant", Project No.: BE97-4949. The project is partly financed by the European Union and the main goal is to develop heat exchanger tubes working at 1100°C and above.

As a part of this project, Sydkraft and Mitsui Babcock have investigated the feasibility of using these tubes in an advanced, indirectly fired combined cycle power plant fuelled by biomass. This paper deals with the preliminary results of this investigation.

Other parts of the project, not presented here, include material tests, material and tube manufacturing and field tests of tubes.

2. Power plant configuration

The prevailing method for generating electricity from biomass is combustion of solid wood to produce steam for a steam turbine. For modern small-scale plants (5 – 10 MW_e) the electrical efficiency is typically 25 – 30 %. In a co-generation application with production of hot water for district heating, the electrical efficiency is normally 3 – 5 percentage points lower. To improve the electrical efficiency, research and development is conducted to enable the use of biomass as a fuel for gas turbines. Possible approaches are indirectly fired gas turbines, direct combustion of wood powder in the gas turbine combustor, or gasification and subsequent combustion of the generated syngas.

[1] Plansee GmbH, MSR GmbH, Mitsui Babcock Energy Limited, Sydkraft AB, University of Liverpool, Risoe and University of Cambridge.

15th International Plansee Seminar, Eds. G. Kneringer, P. Rödhammer and H. Wildner, Plansee Holding AG, Reutte (2001), Vol. 1

Indirectly (or externally) fired gas turbines differ from conventional gas turbines in that the products of combustion do not enter the expander. This is shown in Figure 1. Another difference is that the combustion in the indirectly fired case takes place at atmospheric conditions.

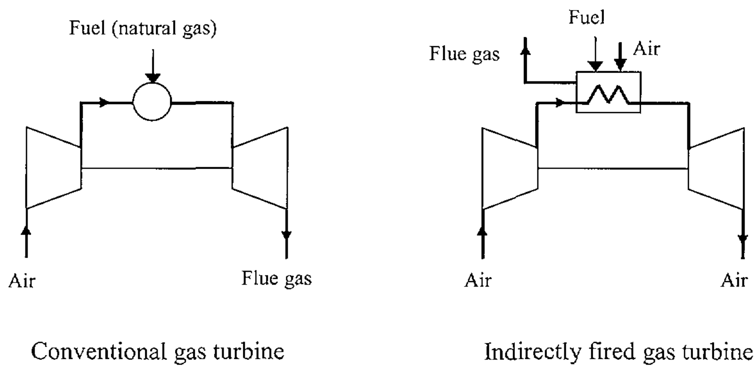

Figure 1. Conventional and indirectly fired gas turbines.

Conventional gas turbines require very clean fuels in order to avoid fouling, erosion or corrosion of the hot parts in the expander. The limits for particulates and alkali species in the working gas are normally on the ppm level. The advantage of the indirectly fired approach is that solid fuels can be used without any risk of damage to blades and vanes in the expander.

Indirectly fired gas turbine cycle systems can be arranged either as serial-coupling systems or as parallel-coupling systems. In a serial-coupling system, the exhaust gas from the gas turbine is used as preheated combustion air for the solid fuel combustor. This results in high-temperature heat being recovered into the gas turbine.

In a parallel-coupling system, on the other hand, heat in the gas turbine exhaust gas is recovered only in a bottoming system. Combustion air is then provided by means of a separate fan. Schematic pictures of these coupling modes are given in Figure 2.

The systems studied in this work are all designed as serial-coupling systems, i.e. gas turbine exhaust gas is used as preheated combustion air in the solid fuel combustor. However, only a fraction of the gas turbine exhaust flow is

15th International Plansee Seminar, Eds. G. Kneringer, P. Rödhammer and H. Wildner, Plansee Holding AG, Reutte (2001), Vol. 1

used as combustion air. The rest of the gas turbine exhaust bypasses the combustor and mixes with the flue gases downstream of the combustor. Residual heat in this mixed gas flow is then recovered in a bottoming steam cycle.

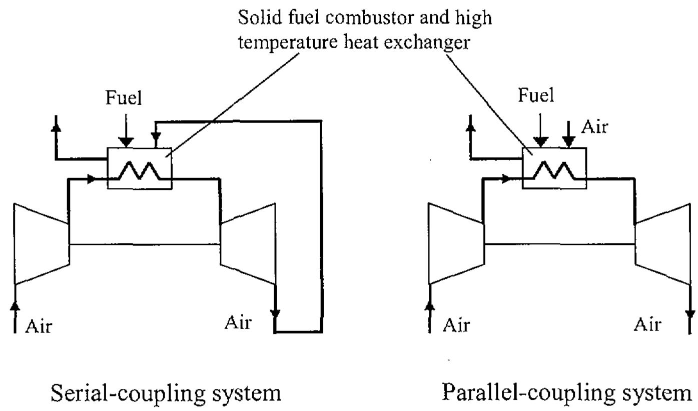

Figure 2. Different layouts of indirectly fired gas turbines.

Serial coupling is thermodynamically advantageous since heat in the gas turbine exhaust is recovered at the highest possible temperature level. The reason for not using the entire gas turbine exhaust flow as combustion air is that only a limited amount of heat is required to heat the compressed air in the high temperature heat exchanger. Any additional heat from solid fuel combustion is recovered in the bottoming system, at lower efficiency. Using more of the exhaust flow to burn more fuel is, hence, of no interest.[2]

The systems studied are based on a fictive gas turbine with performance data similar to VT4400 from Volvo Aero Corporation. Performance data for this machine, at ISO conditions[3] and natural gas fuel, are presented in Table 1.

Modelling of the gas turbine in externally fired applications is based on this information. It has been assumed that the mass flow through the turbine is to be kept at design value. It has also been assumed that the polytropic efficiencies for both turbine and compressor are the same as in the design case. Based on design data, these efficiencies are estimated to 0.850 for the compressor and 0.859 for the turbine.

[2] This is unless a higher output is required from a system with a given gas turbine.
[3] 15°C, 1.013 bar, 60 % relative humidity.

15th International Plansee Seminar, Eds. G. Kneringer, P. Rödhammer and H. Wildner, Plansee Holding AG, Reutte (2001), Vol. 1

Output	4.4 MW
Thermal efficiency	30.5 %
Compressor inlet air flow	19.7 kg/s
Compressor pressure ratio	12.2
Exhaust temperature	474°C
Compressor outlet temperature*	385°C
Turbine entry temperature*	1085°C
Cooling flow, as fraction of compressor air flow*	10 – 15 %

Table 1. Performance data for selected gas turbine at ISO conditions and natural gas fuel. *Estimated by Sydkraft, not given in public sources.

The cooling flow to the gas turbine expander, presented as percentage of design compressor flow, has been estimated to 12.5 %. The resulting cooling mass flows are kept constant in the calculations.

An externally fired combined cycle (CC), based on this gas turbine, has been modelled and calculated by means of the software PROSIM, developed by Endat Oy, Finland. Calculations have been performed for both condensing and co-generation units.

Co-generation is in this context equivalent with production of hot water for district heating. The co-generation system has been designed to give maximum electrical efficiency, given the limits imposed by the temperature levels in the district-heating network. The supply/return temperatures are assumed to be 90/45°C.

The model fuel used in the calculations is wood powder with a very low moisture content, 3%. Hence, starting from raw wood chips, both size reduction and extensive drying is required. These operations have not been included in the studied power plant concepts. It is, however, possible to integrate a dryer, using steam or low temperature heat in the flue gases to dry the fuel.

3. High-temperature heat exchanger

Since the thermodynamic efficiency of the gas turbine cycle depends on the gas temperature and pressure at the expander inlet, a heat exchanger that

15th International Plansee Seminar, Eds. G. Kneringer, P. Rödhammer and H. Wildner, Plansee Holding AG, Reutte (2001), Vol. 1

can withstand high temperatures, in a chemically rather harsh environment, is required. The original specifications for the heat exchanger included a duty of approximately 14 MW_{th}, an increase in air temperature from 385°C to 1085°C and a pressure of about 12 bar(a) on the air side, in order to comply with the data for the chosen gas turbine. In order to reach the final air temperature of 1085°C, the Fe-based ODS alloy PM2000, which can be used at material temperatures of up to 1100°C[4], is to be used in the high temperature part of the heat exchanger. In the low temperature part, where air is heated from 385°C to approximately 650°C, austenitic and ferritic tube materials can be used instead.

The fuel, dry wood powder, can be burnt with a low percentage of excess air. It does, however, have a high adiabatic flame temperature. Preheating the combustion air or using air that has passed through the gas turbine will increase the flame temperature even further, but this is necessary to get a high performance according to discussions in the previous section. The wood powder has low ash content and a melting temperature of ash of around 1250 - 1300°C. This means that the flue gas temperature should be kept below 1200°C to avoid having molten ash in the heat exchanger. The conclusion was that the flame has to be cooled either by radiation in a cooled combustion chamber or by flue gas re-circulation.

Initial calculations showed that a design with a cooled combustion chamber required less ODS material than a design with flue gas re-circulation. Therefore, a design with heat transfer to the pressurised air in two separate heat exchangers was chosen. The high temperature part is a radiant section where ODS material is used for the tubes. The second part, working with a flue gas temperature below 1200°C, is in the form of three conventional convective banks of the cross flow type, with tubes of austenitic and ferritic material.

The present design of the high temperature heat exchanger is based on a small square furnace enclosure (3x3 m) with a vertical burner at its centre and panels of free-hanging ODS tubes (approx. 6 m long) at the walls. The radiant section dimensions were based on the required heat transfer area with assumptions regarding slagging and fouling. The height of the radiant

[4] Normally, this material is highly anisotropic, exhibiting its best mechanical properties in the axial direction. Work is, however, conducted within the aforementioned BRITE / EuRAM Project to improve the material's properties in the circumferential direction.

15th International Plansee Seminar, Eds. G. Kneringer, P. Rödhammer and H. Wildner, Plansee Holding AG, Reutte (2001), Vol. 1

section depends on the flame length and the maximum allowable flue gas exit temperature.

The ODS alloy is restricted in its applications by its mechanical properties. For example, it cannot accommodate bending, it is difficult to weld and the tube length is limited with the present manufacturing method. To overcome these problems, the design of the radiant section uses a bayonet type tube arrangement with an outer tube of PM2000 material and an inner tube of ceramic material, Figure 3. This enables straight sections of PM2000 tubing to be used and reduces the amount of material. The arrangement also means that the tubes are top-supported and allowed to expand individually without constraints. To facilitate assembly and tube exchange, the tubes are connected to the header via threaded connection pieces.

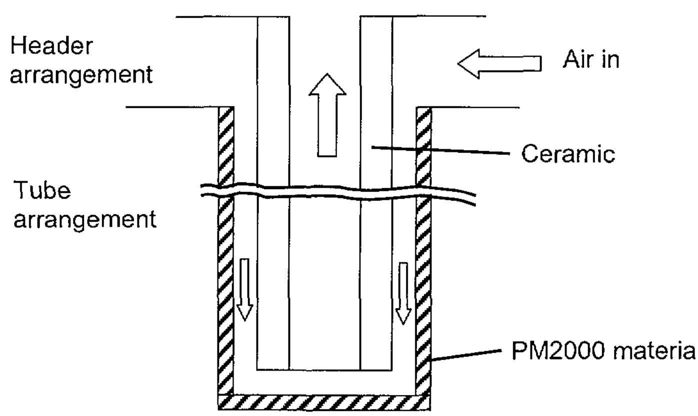

Figure 3. ODS bayonet tube arrangement.

The combustion chamber is followed by a convective heat exchanger. The design is such that conventional materials can be used in this area. However, the outlet flue gas from the radiant section of the furnace will enter the first convective bank at a temperature of 1200°C. Hence, rapid fouling of the hot end of the convective bank must be expected. The convective bank is, therefore, designed with wide tube spacing and soot blowers located between adjacent tube banks, to allow on-line cleaning.

Figure 4 shows the layout of the complete heat exchanger.

15th International Plansee Seminar, Eds. G. Kneringer, P. Rödhammer and H. Wildner, Plansee Holding AG, Reutte (2001), Vol. 1

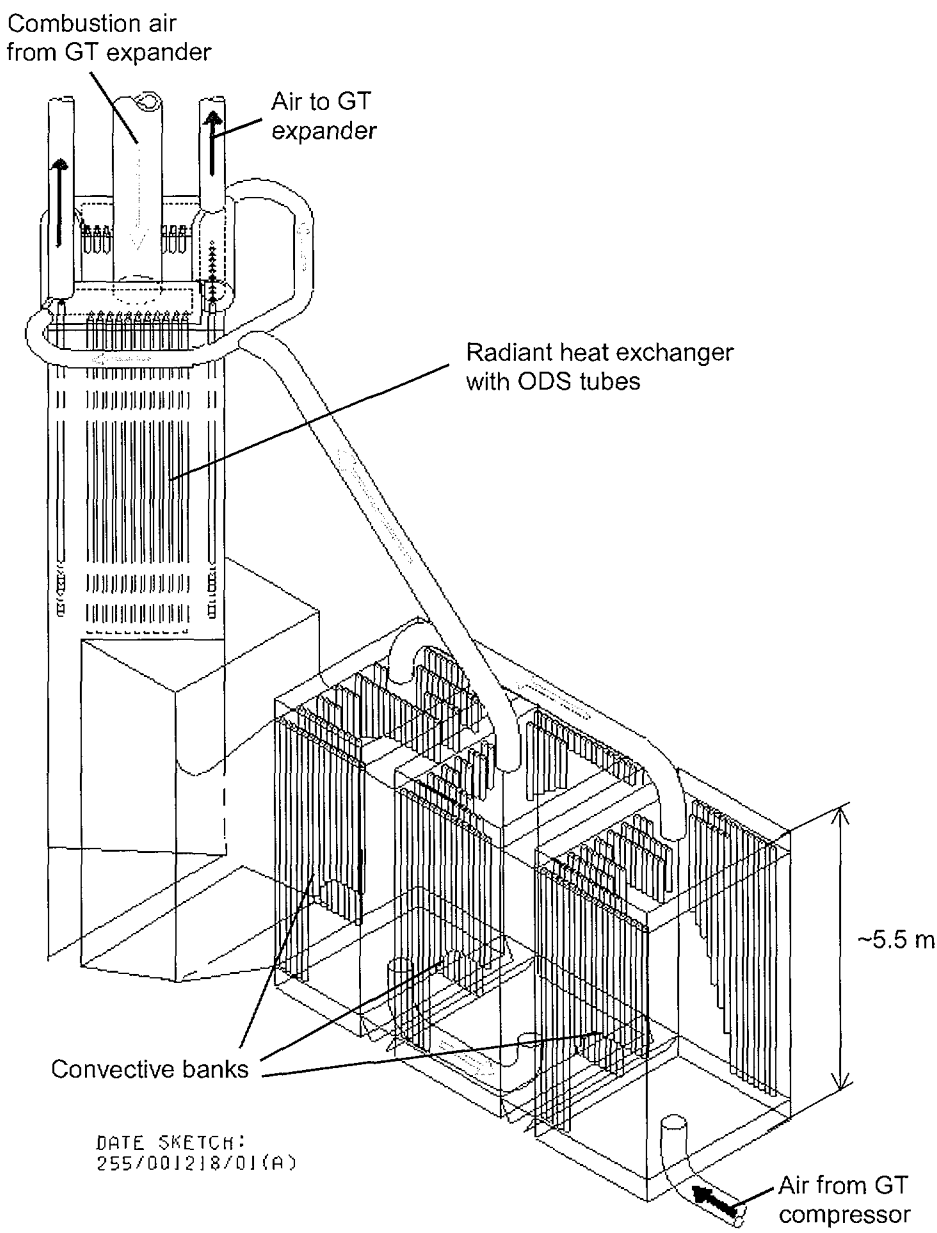

Figure 4. Layout of high temperature heat exchanger.

15th International Plansee Seminar, Eds. G. Kneringer, P. Rödhammer and H. Wildner, Plansee Holding AG, Reutte (2001), Vol. 1

The layout is a result of an iterative process, where results from thermodynamic cycle calculations at Sydkraft have been used as input for the heat exchanger design work at Mitsui Babcock and vice versa. The result of this process is a heat exchanger design with the following characteristics.

Radiant section	
Air inlet temperature (°C)	656
Air outlet temperature (°C)	1013
Air side pressure drop (bar)	0.59
Flue gas pressure drop incl. burner (bar)	0.03
Convective section	
Air inlet temperature (°C)	384°C
Air outlet temperature (°C)	656°C
Air side pressure drop (bar)	0.085
Flue gas pressure drop (bar)	0.0003

Table 2. High temperature heat exchanger data.

Due to the limitation of the maximum temperature in the PM2000 material (1100°C), the final air temperature is 1013°C instead of the desired 1085°C. This is the result of an optimisation of air temperature versus pressure drop. In the heat exchanger the heat transfer is increased by higher air velocity. Higher heat transfer means smaller temperature difference between heat exchanger material and air and, thus, means that a higher air temperature out from the heat exchanger can be reached. This would be beneficial for the gas turbine efficiency. Higher air velocity, however, also means higher pressure drop, which results in lower gas turbine efficiency.

4. Overall plant performance

The power plant concept studied is based on a combined gas and steam turbine cycle. In this section the calculated values of output power and efficiency for condensing and co-generation units are presented.

Design properties for the gas turbine were presented in Table 1. The major difference in the indirectly fired configuration calculated here is that the turbine entry temperature is decreased to 1013°C, in order to comply with the specifications of the ODS heat exchanger. Other differences, compared to

15th International Plansee Seminar, Eds. G. Kneringer, P. Rödhammer and H. Wildner, Plansee Holding AG, Reutte (2001), Vol. 1

design specifications, are increased pressure drops between compressor and turbine and at the turbine outlet.

As was mentioned above, a serial-coupling system is applied. Hence, part of the exhaust gas (air) leaving the gas turbine is used as preheated combustion air in a wood powder burner. In order to introduce the wood powder into the combustion chamber, additional air supplied by means of a roots blower, is used to convey the wood powder to the burner.

4.1 Base case condensing unit

This system is presented in Figure 5, which is a simplified drawing taken from the Prosim software.

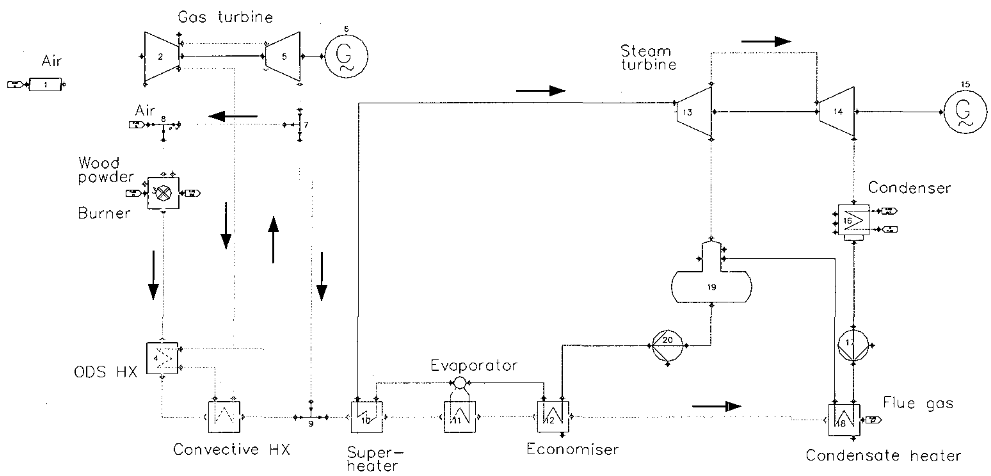

Figure 5. Indirectly fired combined cycle, base case condensing unit.

After heating the high pressure air in the high temperature heat exchanger, the combustion gases are mixed with the excess air from the gas turbine exhaust. The remaining heat in these flue gases is used to produce steam in a single-pressure heat recovery steam generator (HRSG). The steam is expanded in a steam turbine with an extraction for feed water deaeration. In order to recover more energy from the flue gases, a condensate heater is

15th International Plansee Seminar, Eds. G. Kneringer, P. Rödhammer and H. Wildner, Plansee Holding AG, Reutte (2001), Vol. 1

added between the economiser and the stack. This reduces the amount of steam extracted from the turbine.

To perform the calculations, a number of input parameters have to be assigned proper values. In appendix A, the most important input parameters, and the values used in these calculations, are presented. The air temperature at the ODS heat exchanger outlet, and the pressure drops in the radiant and convective heat exchanger parts, are results from the heat exchanger design presented earlier.

The electrical efficiency of this system varies slightly with live steam pressure. With the present live steam temperature, 428°C, the maximum is close to 16 bar and, hence, this value is chosen. The resulting steam flow is only 2.20 kg/s.

The gas turbine output is 3.01 MW_e and the steam turbine output 1.70 MW_e. Taking into account the roots blower and pump work, the net output is 4.67 MW_e. Based on the lower heating value of the fuel, the resulting electrical efficiency is 34.8 %.

The gas turbine output is considerably lower than the design value presented in Table 1. This is due to the decreased turbine inlet temperature and the increased pressure drops.

4.2 Base case co-generation unit

This system is very similar to the system presented above. The main difference is that the condensate heater has been substituted by a heat exchanger for district heating water. This heat exchanger is connected in series with the condenser, where the pressure is now higher than in the system for power production only (0.48 bar versus 0.042 bar).

The electrical efficiency of this system also varies slightly with live steam pressure. With the present live steam temperature, 428°C, the maximum is close to 24 bar and, hence, this value is chosen. The resulting steam flow is 2.10 kg/s.

The gas turbine output is 3.01 MW_e and the steam turbine output only 1.14 MW_e. The net electrical output is 4.10 MW_e and the heat output is 7.13 MJ/s.

15th International Plansee Seminar, Eds. G. Kneringer, P. Rödhammer and H. Wildner, Plansee Holding AG, Reutte (2001), Vol. 1

Based on the lower heating value of the fuel, the resulting electrical efficiency is 30.6 % and the total fuel utilisation is 83.7 %.

4.3 Topping combustion, condensing unit

Presently, the air temperature at the high temperature heat exchanger outlet (1013°C) is lower than the design value of the gas turbine inlet temperature. In order to utilise the full potential of the gas turbine, the addition of a natural gas fired topping combustor has been investigated. As presented in Figure 6, a fraction of the airflow is bled from the outlet of the convective heat exchanger and used to burn natural gas in a separate combustor. The flow is adjusted so that the final air temperature, after mixing with the air from the radiant section, is 1085°C. This is consistent with the design value of the gas turbine inlet temperature.

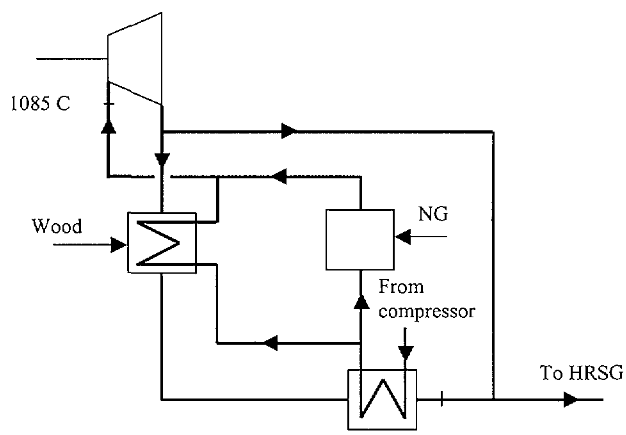

Figure 6. Integration of a natural gas fired topping combustor.

The reasons for taking only part of the air-flow to the topping combustor are that the inlet air temperature is lower, the resulting pressure drop is lower, and it is possible to have a near stoichiometric combustion.

Another option is to extract air for the topping combustor immediately after the compressor. This would, however, increase the amount of natural gas required, since the air would then be colder.

15th International Plansee Seminar, Eds. G. Kneringer, P. Rödhammer and H. Wildner, Plansee Holding AG, Reutte (2001), Vol. 1

Input parameters for this system are presented in appendix A. As a result of the increased gas turbine inlet temperature, the outlet temperature and, hence, the steam temperature out from the super-heater have been increased. The steam turbine inlet temperature is 460°C and the steam flow is 2.45 kg/s.

The gas turbine output is 3.63 MW_e and the steam turbine output 1.98 MW_e. The net output is 5.57 MW_e. Based on the lower heating value of the fuel, the resulting electrical efficiency is 37.4 %. The amount of natural gas added in the topping combustor corresponds to 13.4 % of the total fuel input.

The gas turbine output is still lower than the design value presented in Table 1, even though the turbine inlet temperature is at design value. The reasons for this are the increased pressure drop between compressor and turbine, and the increased pressure at the gas turbine outlet.

4.4 Topping combustion, co-generation unit

Finally, a co-generation unit with a topping combustor has been studied. The result is a gas turbine output of 3.63 MW_e and a steam turbine output of 1.32 MW_e. The net electrical output is 4.91 MW_e and the heat output is 7.69 MJ/s. Based on the lower heating value of the fuel, the resulting electrical efficiency is 33.0 % and the total fuel utilisation is 84.7 %. Again, the amount of natural gas added in the topping combustor corresponds to 13.4 % of the total fuel input.

5. Discussion

Using ODS alloy tubing, it is possible to design a high temperature heat exchanger operating with metal temperatures of around 1100°C. The proposed design consists of a radiant section, where ODS material is used for the tubes, and a convective section using conventional tubing materials.

The results of the thermodynamic calculations, summarised in Table 3, indicate that the indirectly fired gas turbine systems have a potential for comparably high electrical efficiencies. The corresponding figure for a condensing steam cycle power plant of this size is normally around 25 %. Hence, the development of a high temperature heat exchanger could make an important contribution to the development of systems converting solid fuels to electricity with high efficiency.

15th International Plansee Seminar, Eds. G. Kneringer, P. Rödhammer and H. Wildner, Plansee Holding AG, Reutte (2001), Vol. 1

	Electric output (MW)	Heat output (MJ/s)	Electrical efficiency (%)	Fuel utilisation (%)	Natural gas (%)
Base case, condensing	4.67	-	34.8	34.8	0
Base case, co-generation	4.10	7.13	30.6	83.7	0
Topping combustion, condensing	5.57	-	37.4	37.4	13.4
Topping combustion, co-generation	4.91	7.69	33.0	84.7	13.4

Table 3. Thermodynamic performance of studied concepts.

Due to the present limitation of the maximum temperature in the ODS material, the final air temperature is lower than what was originally aimed for (1013°C instead of 1085°C). In order to utilise the full potential of modern gas turbines, further development of the ODS material and the heat exchanger design is required. Meanwhile, top firing with natural gas is an interesting option.

This work will continue during 2001 with detailed studies of the heat exchanger design and its integration with the gas turbine. Also, the economy of an indirectly fired combined cycle power plant of this type will be assessed.

6. Acknowledgements

We wish to thank our colleagues Erik Skog and Rolf Öberg, at Sycon Energikonsult, and Alex Fleming, at Mitsui Babcock, for their valuable contributions to this paper.

The financial support from the European Union is also greatly acknowledged.

15th International Plansee Seminar, Eds. G. Kneringer, P. Rödhammer and H. Wildner, Plansee Holding AG, Reutte (2001), Vol. 1

Appendix A. Input parameters for the thermodynamic calculations.

	Base case, condensing	**Base case, co-generation**	**Topping combustion, condensing**	**Topping combustion, co-generation**
High temperature heat exchanger (radiant + convective)				
Heat loss (% of heat transferred)	1.5	1.5	1.5	1.5
Air outlet temperature (°C)	1013	1013	1013	1013
Air inlet temperature (°C)	384	384	384	384
Pressure drop air side (bar)	0.915	0.915	0.915	0.915
Pressure drop flue gas side (bar)	0.08	0.08	0.08	0.08
Minimum temperature difference (°C)	75	75	75	75
Burners				
Wood powder burner lambda	1.10	1.10	1.10	1.10
Natural gas burner lambda	-	-	1.02	1.02
Natural gas burner heat loss (%)	-	-	1	1
Gas turbine				
Expander inlet temperature (°C)	1013	1013	1085	1085
Inlet pressure drop (%)	1	1	1	1
Steam cycle				
Live steam pressure (bar)	16	24	16	24
Superheater min. temp. diff. (°C)	30	30	30	30
Evaporator pinch-point (°C)	15	15	15	15
Economiser approach temp. (°C)	10	10	10	10
HP turbine isentropic efficiency	0.65	0.65	0.65	0.65
LP turbine isentropic efficiency	0.75	0.75	0.75	0.75
Feed water tank pressure (bar)	1.21	1.21	1.21	1.21
Condenser min. temp. diff. (°C)	5	5	5	5
District heating heat exchanger min. temp. diff. (°C)	-	15	-	15
Generators				
Gas turbine generator efficiency	0.965	0.965	0.965	0.965
Steam turbine generator efficiency	0.96	0.96	0.96	0.96