COBEM2005-007 – Performance and Optimisation of Externally Fired Gas Turbines Fuelled with Biomass

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Abstract. Biomass based fuels are attracting much interest recently due to their plentiful supply and their environmentally friendly characteristics if properly managed. The difficulties of burning low grade fuels such as biomass in gas turbines is that they contain some particulate matter which causes corrosion and erosion problems in the turbine blades; this can be overcome by indirectly firing the gas turbine. The work in this paper has been focused on Externally Fired Gas Turbine (EFGT) cycles fuelled with Bagasse, residue of sugar cane milling process, which is especially abundant in Brazil. In a first part, in order to assess the combined cycle performance, the analysis of the simple gas turbine cycle has been carried out. A sensitivity analysis has permitted the determination of an optimal gas turbine in terms of pressure ratio and turbine inlet temperature. Off-ambient and part load performances are also presented within this work. Then in a second part, the influence on the externally fired combined cycle has been determined, and the cycle optimised at design. The simulations have been carried out with the commercial code GateCycle, and the results compared with a conventional configuration of gas turbine and combined cycle fuelled with natural gas. The EFGT cycle presents some disadvantages in terms of gas turbine performance. However, due to its regenerative aspect, the externally fired cycle offers good steam cycle performance. The externally fired combined cycle (EFCC) promises to be competitive with other cycles, even when fuelled with low heating value fuels.

Keywords: Externally Fired Cycles, Combined Cycle, Optimisation, Sugarcane Bagasse

1. Introduction

Nowadays, the world faces a choice of unsatisfactory options for generating electricity. Traditional electricity generation technologies based on fossil fuels, hydroelectricity and nuclear power all face problems. Coal is cheap and abundant, but it emits more of the "greenhouse" gas carbon dioxide per unit of energy than oil or gas, as well as sulphur and nitrogen oxides causing "acid rain". Oil, because of its history of price volatility, has fallen from favourite as a major fuel for base load electricity generation.

Natural gas is the cleanest fossil fuel, but in many parts of the world the necessary infrastructure may not be available for decades, and the long term availability and cost of natural gas may depend on transport, sometimes over a thousand kilometres. Hydroelectricity is the classical renewable energy, but even where sites are still physically available, traditional hydroelectricity based on large demand has become intensely controversial, because it floods vast areas of land, displaces populations and upsets local ecosystems. Nuclear power emits no carbon dioxide from its reactors, but it faces problems of public acceptance, economic obstacles to financing in competition with other generating technologies, and raises questions about safety, waste disposal and the security of fissile materials.

Amongst renewable energy resources, biomass power technology may prove to be a valuable option. In theory, solid biomass may be used to fuel gas turbines by direct firing, indirect firing or via gasification. Direct firing of gas turbines with biomass is constrained by the corrosive and erosive nature of the combustion products. There is no gas turbine capable of withstanding these factors. In the gas turbine with biomass gasification, the biomass must first be converted to clean fuel gas, and then burned in the combustor. The Biomass Integrated Gasification Gas Turbine (BIG/GT) has been developed and put into practice in several places now.

However, the gasifier is relatively large, sensitive to the quality and size of solid fuel and costly. The main limitations of this complex component are the energy losses. The indirectly fired or externally fired gas turbine, EFGT, is very attractive for future applications in power generation. The solid biomass is burnt in an external combustor and the hot gases pass through a heat exchanger to heat the compressed air. This paper focuses on the analysis of this cycle, which offers various advantages and difficulties of operation.

2. Externally Fired Cycles

A key feature of the EFGT is an atmospheric combustor and a high temperature heat exchanger, which replaces the conventional combustion system in a standard gas turbine cycle. The air delivered by the compressor receives heat energy from combustion gases in the heat exchanger. It is then expanded in the gas generator turbine to provide work for the compressor, and then goes through the power turbine to generate useful work. After the power turbine, all the energy that has not been converted to shaft power is available in the exhaust gas. Therefore, the power turbine exhaust gases are directed into the combustor, diminishing the heat input required.

When used in conjunction with a steam bottoming cycle, as in the externally fired combined cycle (EFCC) (Figure 1), the heat rejected by the gas turbine heat exchanger can be used as the heat input to the steam cycle. The gases from the exhaust of the heat exchanger are still hot and will heat up the water in the Heat Recovery Steam Generator (HRSG) to produce steam at high pressure. This steam will then be expanded in the steam turbine to produce shaft power. At the exhaust of the steam turbine, the steam is just on the point to become water. With an air cooling system, or cold water from a river, for example, the steam will be cooled down and transformed into water in the condenser. The pump will then, with a very little need of energy, bring this water (liquid) to a high pressure so that the steam produced in the HRSG will be at high pressure as well. Figure 1 also represents the T-s diagram of the EFCC.
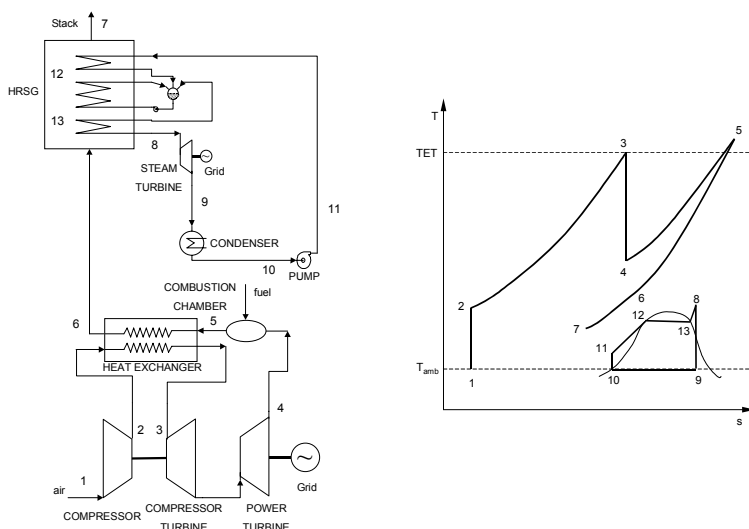

Figure 1 - Schematic of the EFCC cycle

The main drawback of this approach is the low efficiency of the cycle compared with the conventional natural gas fired combined cycle. This low performance is due to the use of metallic heat exchangers, which constrains the gas turbine entry temperature TET, far below the design temperature used in ordinary engine. The competitiveness of the EFGT depends strictly on the capability to operate the heat exchanger at very high temperature: hot gas entering at 1300-1600°C, compressed air exiting at 1100-1300°C Consonni and Macchi, (1996). This requirement rules out metallic materials and confines the choice to ceramics. Besides elevated temperatures, the ceramic heat exchanger must cope with the following: pressure difference between the cold air compressed by the compressor and the hot gases coming from the atmospheric combustor; the ensuing stresses cause structural and leakage problems; high temperature corrosion of the ceramic material; and erosion and possible rupture of the ceramic tubes caused by ash particles. To prevent this occurrence, it is mandatory to collect ash (or slag) upstream of the heat exchanger. Another aspect to bear in mind is that the thermal inertia of the heat exchanger reduces the flexibility for controlling the power output simply by acting on fuel flow rate Ranasinghe; Aceves-Saborio; and Reistad, (1989).

On the other hand there are several advantages in using the EFGT cycles fuelled with biomass. First, thanks to the heat exchanger, the gas turbine operates on clean air, so the gas path is never exposed to the corrosive elements in the fuel. This avoids the requirements for advanced hot gas clean-up system to protect the GT components. The maintenance costs will therefore drop and the engine lifetime will be augmented. Moreover, the gasifier is no longer needed. This device, necessary in the BIG/GT, is very costly and introduces large losses in the overall cycle efficiency, not to mention its bulk Ferreira and Pilidis, (2001).

The second major advantage of the EFGT cycle is the versatility of the combustion chamber. Many different solid fuels can be burnt, allowing the use of the fuel which is the cheapest or readily available in that season. The pre-treatment needed is modest when compared with other cycles Ferreira and Pilidis, (2001). These points are of great relevance when considering biomass fuels.

The flow through the turbine is slightly smaller than the compressor flow (difference is due to the need for sealing flows), unlike the BIG/GT; there is therefore no compressor stall problem due to the important mass flow increase in the turbine. Furthermore, with no flame upstream, the temperature profile of the flow entering the gas turbine is particularly uniform; this implies lower thermal stresses and lower cooling flow, for the same TET Consonni; Macchi; and Farina, (1996). It is also important to notice that no redesign of the expander, compressor or combustion system is needed; the conventional equipment and current technology are suitable to the EFGT engine.

The main design data for the gas and steam cycle models are coming from conventional data found in literature Zwebek, (2002), and are summarised in Table 1.

Table 1 - Design parameters for the EFGT and EFCC models

Design value	EFGT	Design value	EFCC
η_c	0.88	$T_{approach}$ (°C)	20
η_{cc}	0.99	ΔT_{pinch} (°C)	10
$(\Delta P/P)_{cc}$	0.02	T_{stack} (°C)	121
η_{turb}	0.89	η_{ST}	0.89
φ_{hx}	0.9	limit T_{steam} (°C)	540
TET (K)	1350	η_{HRSG}	0.85
W (kg/s)	140	P_{cond} (bar)	0.1

The fuel used is the Brazilian bagasse called Bagatex, which composition has been given elsewhere, Codeceira-Neto, (1999). Concerning the combustor pressure losses, they have been chosen lower than the usual values (4%), because the temperature in the combustor is lower than usual. The TET was fixed by the limitation of the heat exchanger. Despite all the attempts to develop a suitable heat exchanger for EFGT cycles such a device would be available at high costs for a long life prototype application. For this reason, the hot side inlet of the heat exchanger was set to 1435K (1162°C) which constrains the TET at 1350K (1077°C).

The approach temperature, which is the difference between the saturation temperature in the drum and the water temperature at the economiser outlet, has been set to 20°C, to avoid evaporation in the economiser at off design conditions, Dechamps, (1999). The stack temperature has to be high enough to prevent condensation, avoiding corrosion problem due notably to sulphur content. In this study, since the bagasse used as fuel has quite low sulphur content, the stack temperature was limited to 121°C, which is lower than the usual 140°C used for coal.

The superheated temperature is usually limited to 540°C because of the thermal capability limitation of the first blade of the steam turbine, which is uncooled. To avoid erosion problems, it is very important to limit the droplets content in the steam expanding through the last stages of the turbine. Therefore, the exhaust steam quality is maintained above a certain limit, fixed at 0.88.

3. Results and Discussion

Initially a performance analysis of the EFGT has been carried out to underline the specificities of the externally fired cycles. Then a pressure ratio sensitivity analysis has been done to determine the optimum combined cycle design point.

In this study, the design point was chosen at 25°C (typical temperature for countries in which sugarcane bagasse is plentiful) and 1atm. The following equations are reviewed to help to understand the explanation about the behaviour of efficiencies and shaft power for the calculations of the EFGT and EFCC.

$$\eta_{GT} = \frac{PO_{GT}}{HI} \qquad (1)$$

$$\eta_{CC} = \frac{PO_{GT} + PO_{ST}}{HI} \qquad (2)$$

$$\eta_{CC} = \eta_{GT} + (1 - \eta_{GT}) \times \eta_{HRSG} \times \eta_R \qquad (3)$$

Where:
$$HI = \dot{m}_{fuel} \times LHV$$

$$PO_{GT} = EW - CW$$

$$\eta_{HRSG} = \frac{T_{exhaustGT} - T_{stack}}{T_{exhaustGT} - T_{ambient}}$$

As a first approach, the EFGT was assessed at design; then an off-design analysis has been carried out to assess the performance of the engine over a complete range of operating conditions which it can experience.

The EFGT and the conventional cycle configuration models were run under the design mode over a range of pressure ratios varying from 2 to 30, while the TET was kept constant to its maximum limit of 1077°C. Figure 2, Figure 3, and Figure 4 show the results of these simulations for both cycles, in terms of efficiency, power output and exhaust cycle temperature.

As far as the specific power output is concerned, the results obtained with the EFGT retain essentially the same shape as for the conventional cycle: the specific power output increases as the TET and pressure ratio increase, up to its maximum of 254 (kJ/kg), achieved for a pressure ratio of 12. The specific power output obtained with the EFGT is lower than for the standard natural gas fired turbine. In the EFGT cycle, the presence of the heat exchanger increases the pressure drop, consequently, the pressure at the inlet of

the turbine is lower than with the conventional cycle, hence, the expansion work obtained is lower. Moreover, at the exit of the EFGT cycle power turbine, the pressure is fixed to a value higher than the atmospheric pressure, to allow for the pressure drop in the combustor, ducts and heat exchanger before the exhaust. As a consequence, the expansion work obtained through the power turbine is lower, for the same TET and air mass flow. However, in the EFGT cycle, the variation of the efficiency with the pressure ratio presents a major difference compared with that of the simple conventional cycle, as illustrated in Figure 3.

In the conventional cycle, the efficiency increases with the pressure ratio and starts to decrease only for high pressure ratios where the reduction of power output is higher than the reduction of heat input. On the contrary, in the EFGT the heat of the power turbine exhaust gas is used to heat the air in the combustion chamber, thus reducing the heat input. Or, as the pressure ratio increases, the power turbine exhaust gas temperature is reduced. As a consequence, even if the total heat input is reduced due to the increase of compressor discharge temperature (at constant TET), the heat introduced by the exit gas is reduced and the final heat input required increased. Finally, the efficiency of the externally fired cycle decreases with an increase of pressure ratio. Considering the case of an ideal regenerative cycle, from which the EFGT is derived, the efficiency can be expressed as, after Cohen; Rogers, and Saravanamuttoo, (1996):

$$\eta_{th} = 1 - \left[\frac{PR^{\frac{\gamma-1}{\gamma}}}{\left(TET / T_1 \right)} \right]$$
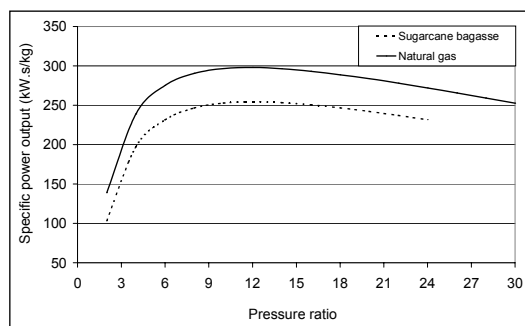

Figure 2 – Comparison between specific power output and pressure ratio for the EFGT and natural gas cycles
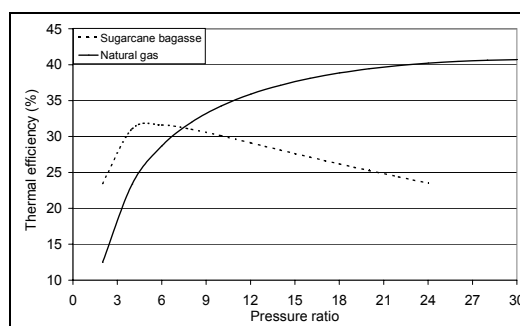

Figure 3 – Comparison between thermal efficiency and pressure ratio for the EFGT and natural gas cycles

For the low pressure ratio, instead of tending to infinity, the efficiency falls to zero because at this pressure ratio, the turbine provides just sufficient work to drive the compressor; at this point there is positive heat input with zero net work output. For the EFGT, an optimum efficiency of 31.5% was obtained for a gas turbine pressure ratio of 6.3. The highest efficiency for the EFGT cycle is achieved at a relatively low PR, which means that the compressor to be used does not need to be a state-of-the-art, consequently high cost, device. The efficiency achieved by the EFGT cycle is higher than the one achieved by direct firing cycle. The BIG/GT cycle for its respective optimum PR of 18, presents an efficiency of 28.5%, Ferreira; Pilidis; and Nascimento, (2001). However, the power output obtained with the BIG/GT cycle is higher than the one obtained with the EFGT cycle1, respectively 36.8 MW and 32.8MW, Fryer, (1995).

It is worth noticing that, in the EFGT, the exhaust gases from the power turbine are re-injected in the combustor and go through the heat exchanger before the final exhaust. Hence, even if the power turbine exhaust temperature is decreasing when the pressure ratio is increasing, as the compressor exit temperature is higher, the energy exchange needed in the heat exchanger, for the same TET, will be lower and the final exhaust temperature will increase.

To estimate the performance of the EFGT over a range of ambient conditions, the engine control system must be simulated. The control system of the usual industrial engines keeps the firing temperature essentially constant at all ambient temperatures, thus keeping the TET constant. Clearly, it is easier to keep the exhaust temperature constant, since it can be directly measured and no calculations are involved. However, keeping the TET constant is the control mode that will give the best performance and the lower losses in power and efficiency. Moreover, keeping the firing temperature constant will be more suitable for the heat exchanger, since it will introduce the minimum change in temperature, hence minimising possible cracks in the ceramic material. However, at lower ambient temperatures, the firing temperature must be reduced since the compressor corrected speed ($N / \sqrt{T_1}$) must not exceed a specified maximum value to prevent over speed, Palmer and Erbes, (1993).

In the GateCycle® models, a macro has been defined to set the firing temperature at the adequate value to get the constant TET of 1077°C. In the cases where the corrected speed limit was reached, the TET has been manually decreased until the iterations led to an acceptable value of corrected speed. The EFGT off-ambient performance are comparable to the conventional cycle performance; the

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[1] Considering the same air mass flow and optima pressure ratios.

shaft power and efficiency increase when the ambient temperature is reduced, whereas the exhaust gas temperature reduces (Figure 5).

The use of low heating value fuel has the effect of shifting the point where the different control limits actuate, due to the difference in compressor map used. The compressor maps were chosen based on their design point pressure ratio, hence for the EFGT cycle it is a low pressure compressor map, whereas for the conventional cycle fuelled with natural gas, it is a high pressure compressor map. In the high pressure compressor map, the corrected speed lines become straight and closer for a smaller flow function ($\dot{m}\sqrt{T}/p \cdot A$) than in the low pressure compressor map. Hence, the limit of corrected speed is achieved earlier as flow rises.

In the case of the natural gas configuration, the TET has been reduced to keep the corrected speed value below its limit. In the EFGT, the corrected speed limit has not been reached and the power output has not been reduced.
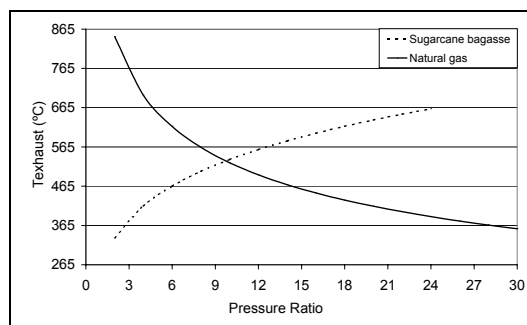

Figure 4 – Comparison between exhaust temperature for the EFGT and natural gas cycles
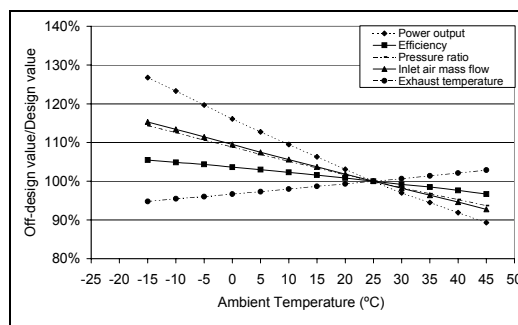

Figure 5 - Cycle parameters versus ambient temperature for the EFGT cycle

The output of a gas turbine may be reduced using two main techniques Dechamps, (1999): a TET reduction, leading to a pressure ratio reduction and hence to a power output reduction and an efficiency reduction; a reduction of mass flow in the cycle. This can be achieved by closing the variable inlet guide vanes (VIGVs) in the compressor. Closing the VIGVs will reduce the mass flow to 70 to 80% of its design value and hence the power output. While they are being closed, the TET is kept to its design value; this allows the power to be reduced without reduction of the cycle maximum temperature. However, when the VIGVs are fully closed, it must be followed by a reduction of TET to be able to reduce the power further.

Comparing the two methods of reduction in power, the efficiency drops at part load are larger with the VIGV control (Figure 6). A reduction of 35% of shaft power, for example, will lead to a drop of efficiency of 8% with the VIGV control and to a drop of only 3.7% with the TET reduction. Hence the performance, in terms of SFC, is better at part load with a reduction in TET. On the contrary, it can be noticed that the TET reduction implies more rapid exhaust temperature degradation than the VIGV control. This is a very important parameter for the combined cycle, but not for the gas turbine itself. The reduction of efficiency and exhaust temperature is lower for the natural gas configuration, again this is due to the difference of compressor map used.
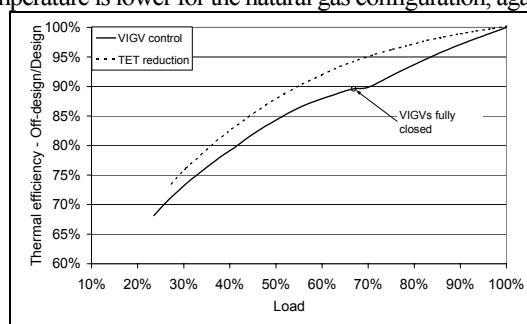

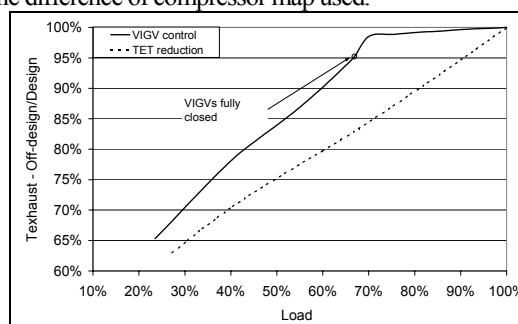Figure 6 – Efficiency comparison at part load between VIGVs control and TET control for the EFGT cycle

Figure 7 – Exhaust gas temperature comparison at part load between VIGVs control and TET control for the EFGT cycle

The behaviour of the EFGT presents differences from the conventional cycle, especially in terms of efficiency and exhaust temperature due to its regenerative configuration. Consequently, the behaviour of the EFCC will also present some differences from the conventional combined cycle.

Combined cycle plants generally can achieve efficiencies of the order of 60%. This analysis aims at finding the new gas and steam turbine characteristics that will give the highest efficiency for the design point.

The TET of the GT is limited by the heat exchanger thermal capability and was set to 1077°C. A gas turbine pressure ratio sensitivity was then carried out for the EFCC and the natural gas combined cycle configuration. GateCycle® asks for the value of the steam pressure of the steam turbine, when operating at design point. A macro was built for this purpose. For the same steam

temperature and condenser pressure, as the steam pressure is increased, the efficiency and power output are increased but the steam quality is reduced. The macro was set to find the highest steam pressure before the steam quality falls below its limit.

In the conventional configuration, as the pressure ratio is increased, the efficiency and power output of the GT are increased but the exhaust temperature is decreased. This leads to a reduction in the steam pressure that can be used in the steam turbine. Indeed, a reduction in exhaust gas turbine temperature is accompanied by a reduction of steam temperature (since the HRSG characteristics are kept constant). As a consequence, the steam pressure has to decrease to keep the steam quality value above its limit. The gas turbine pressure ratio was increased up to 27, after which the required decrease of steam pressure is too important and can not be achieved to keep the steam quality above its limit (Figure 8).

Contrary to the conventional cycle, the EFGT exhibits an increase of exhaust temperature when the gas turbine pressure ratio is increased. Consequently the steam temperature is increased, and the steam pressure can be increased otherwise the steam quality will stay very high, resulting in efficiency and power output losses. The gas turbine pressure ratio was increased up to 23, after which, the corresponding increase of steam temperature was too high and the stack temperature was falling below its limit.

This difference of steam pressure evolution, when the gas turbine pressure ratio is increased at design will lead to a completely different behaviour of steam cycle performance.
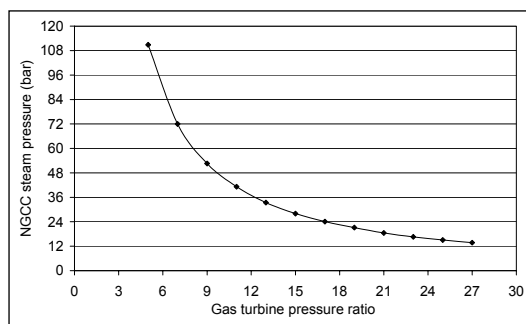

Figure 8 – Live steam pressure versus gas turbine pressure ratio for the natural gas combined cycle
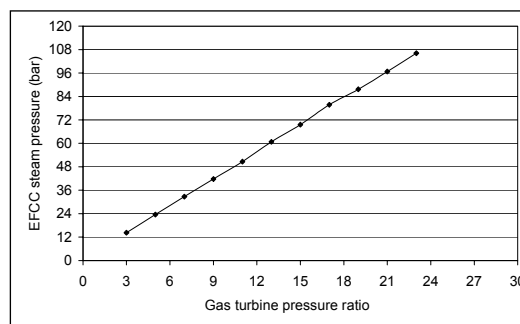

Figure 9 – Live steam pressure versus gas turbine pressure ratio for the EFCC cycle

Concerning the power output for the EFCC, the gas turbine power output is increasing up to its maximum value and then decreasing as in the conventional configuration. As the steam pressure and temperature are increasing with the gas turbine pressure ratio, the steam power output exhibits a continuous increase contrary to the conventional configuration, where the steam power output is reducing. Consequently, the combined cycle power output shows a continuous increase, less sharp than the steam turbine increase, due to the decrease of gas turbine power output (Figure 10). Contrary to the conventional cycle, the EFCC power output increases instead of decreasing as pressure ratio increases.
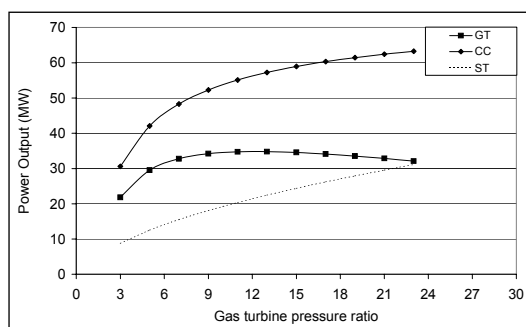

Figure 10 – Power output versus gas turbine pressure ratio for the EFCC cycle
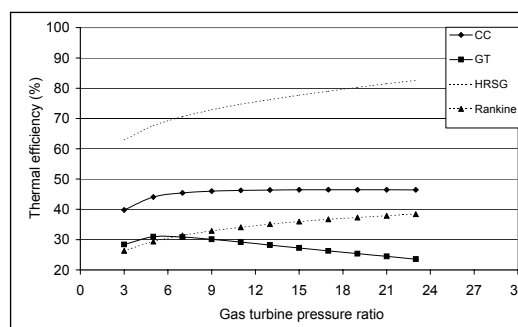

Figure 11 – Thermal efficiency versus gas turbine pressure ratio for the EFCC cycle

As far as the efficiency is concerned, the EFGT efficiency is behaving differently because of the presence of the heat exchanger; it is reducing as the gas turbine pressure ratio is increasing. The steam cycle efficiency is also behaving differently due to the increase of live steam pressure and temperature; it is increasing continuously with the gas turbine pressure ratio. Since the exhaust gas temperature is increasing and the stack gas temperature decreasing, the HRSG efficiency is increasing with the gas turbine pressure ratio. As a consequence, the combined cycle efficiency exhibits a continuous slight increase.There is no maximum value, so the design point pressure ratio chosen was 13, because after that the steam turbine entry temperature was higher than its thermal limit of

540°C. A combined cycle power output of 57.2MW and a combined cycle efficiency of 46.4% correspond to this pressure ratio. The equivalent steam pressure is around 61 bars, which is much higher than the design point steam pressure of the conventional natural gas configuration.
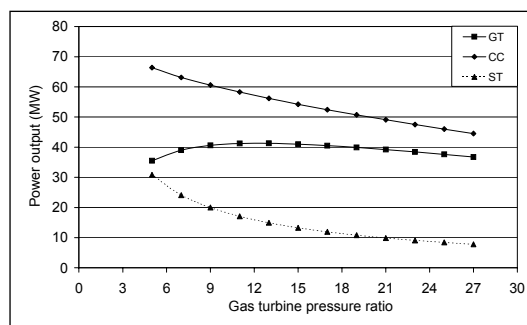

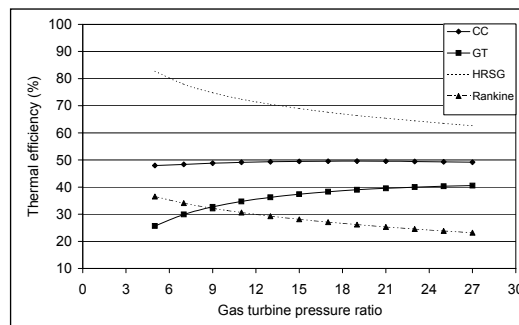Figure 12 – Power output versus gas turbine pressure ratio for the natural gas combined cycle

Figure 13 – Thermal efficiency versus gas turbine pressure ratio for the natural gas combined cycle

It is important to notice that the optimum pressure ratio for the EFCC is higher than the optimum one of the simple EFGT. A change in gas turbine for more advanced technology will be required for the EFCC, which is not advantageous. It is important to recall that the simulations were run with the same TET, mass flow, HRSG, condenser, gas and steam turbine characteristics. However, when comparing the power output of the conventional combined cycle and of the EFCC for the same gas turbine pressure ratio, it is striking to realise that the power output obtained with the EFCC is higher than the one obtained with the conventional configuration (by approximately 1.7%). The behaviour of the steam turbine power output is the reason for this difference. Concerning the efficiency, the EFCC efficiency is still lower than the one obtained with the conventional configuration by about 6%. This is due, partly to the extra losses in the combustor and heat exchanger.

4. Conclusions

The fuel flexibility, clean gas operation, and simplicity of the EFGT are its principal attributes. Commercially available gas turbines can be employed with relatively minor modifications to accommodate the heat exchanger in lieu of the gas turbine combustor system.

From a thermodynamic point of view, the EFGT operates like a regenerative cycle. Contrary to the conventional cycle, the efficiency of the EFGT is reducing for higher pressure ratio. The design point analysis has underlined that a maximum efficiency of 32% is obtained for a quite low pressure ratio, 6.3. The efficiency achieved is lower due to the regenerative aspect of the cycle and because of additional pressure losses.

The specific power is lower than that obtained with the conventional cycle fuelled with natural gas, notably because of the pressure losses introduced by the heat exchanger and because more exhaust pressure is required to take into account the pressure drops in the combustor and heat exchanger before the exhaust.

Another difference presented by the EFGT, is the increase of exhaust temperature as the pressure ratio increases at design. This will lead to a completely different behaviour of the combined cycle.

The part load performance of the EFGT was simulated with two different control modes of power reduction. It has been shown that a TET reduction will give the best performance in terms of efficiency and fuel economy. However, the VIGV control mode leads to a lower degradation of exhaust cycle temperature, which is interesting for the combined cycle applications.

The EFCC analysis reflects that the regenerative characteristic of the EFGT leads to a quite different combined cycle behaviour. The conventional gas turbine has two contrary effects within the combined cycle: an augmentation of gas turbine pressure ratio will increase the gas turbine efficiency and power output but it will also reduce the heat available in the exhaust for the combined cycle and hence reduced the power output and efficiency of the steam cycle. On the contrary, since an increase of pressure ratio means an increase of exhaust gas temperature in the EFGT, the performance of the combined cycle will increase continuously, whereas the efficiency of the GT will fall. This has led to the choice of an optimum PR of 13 for the externally fired combined cycle which gives the maximum efficiency allowed before the steam turbine inlet temperature reaches its limit.

The externally fired cycle presents some disadvantages in terms of gas turbine performance, due to the limitation in gas turbine inlet temperature and due to the presence of additional pressure losses. However, due to its regenerative aspect, the externally fired offers good steam cycle performance. The low gas turbine pressure ratio required at design leads to a high exhaust gas temperature, heat input to the steam cycle. The externally fired combined cycle promises to be competitive with other cycles even fuelled with low heating value fuels.

The most effective technology advance to promote externally fired cycles is a better heat exchanger with high temperature capability and long life cycle.

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