THE USE OF BIOMASS FUELS IN GAS TURBINE COMBINED CYCLES: GASIFICATION vs. EXTERNALLY FIRED CYCLE

Sandro Barros Ferreira*
Instituto Tecnológico – ITUC, PUC – Rio de Janeiro – Brazil
Tel.: +55 (0) 3114 1857, Fax: +55 (0) 3114 1543
sandro@ituc.puc-rio.br

Pericles Pilidis
School of Engineering, Cranfield University
Tel.: +44 (0) 1234 754 646, Fax.: +44 (0) 1234 750 728
P.Pilidis@Cranfield.ac.uk

Marco A. R. do Nascimento
NEST – DEM/IEM – EFEI, Itajubá – MG – Brazil
Tel.: +55 (0) 35 3629 1266, Fax: +55 (0) 35 3629 1355
marco@iem.efei.br

Abstract – The use of biomass as gas turbine combined cycle fuels is broadly seen as one of the alternatives to diminish greenhouse gas emissions, mainly CO_2, due to the efficiency delivered by such systems and the renewable characteristic of biomass itself. Integrated gasification cycles, BIGGT, are the current technology available, however the gasification system severely penalizes the power plant in terms of efficiency and demands modifications in the engine to accommodate the large fuel mass flow. This gives an opportunity to improvements in the current technologies and implementation of new ones. This paper intends to analyze new alternatives to the use of solid fuels in gas turbines through the use of external combustion, EFGT, discussing its advantages and limitations over the current technology. A study of the use of intercooled and recuperated cycles in conjunction with integrated gasification and external combustion is also carried out.

The results show that the intercooled and recuperated cycles are the best systems in terms of efficiency. However due to their complexity and given the already high costs of BIGGT and EFGT cycles their intercooled/recuperated counterparts are not likely to come into operation in the near future. On the other hand, the inherently recuperative characteristic of the EFGT gas turbine engine makes it well suited to the biomass market. The thermal efficiency of this cycle is much higher than the BIGGT system. Furthermore, its fuel flexibility is another advantage that makes it an interesting option for the Brazilian market

1. Introduction

Brazil is a vast country and its potential for the use of renewable energy is undoubtedly immense. Biomass is one of the most promising renewable fuels within the actual scenario, and the wise use of this potential can contribute to a diversified and reliable energy matrix.

Taking the State of São Paulo as an example, its sugar cane production represents 62% of the total production of this crop in the country. There are more than 2,200,000 ha, yielding an average of 104 ton/ha of cane and 14.42 ton/ha of straw Magalhães and Braunbeck, 1999, if just 50% of the straw were used for power generation, it would be equivalent to 25% of the oil imported by the country.

* Corresponding author

The by-product used for power generation in the sugar mills is the fibrous residue that comes out of the process called bagasse. This fuel is generally used in steam cycles, allowing low efficiencies. The average production is about 20-30 kWh/ton of milled cane Nogueira and Walter, 1995, as mechanical and electrical power, being the former used to move the mills. This situation could be changed with the use of more efficient conversion processes, raising that average to 105 kWh/ton of milled cane Ingham, 1998.

Amongst the most promising technologies for the use of biomass as fuel are the gas turbine cycles. Due to the high sensitivity of these engines, the conventional approach is to gasify the solid fuel before its combustion in the engine, the well known biomass integrated gasification/gas turbine cycle, BIGGT. Direct burn of biomass in gas turbines has proven difficult, with a high rate of carbon deposition in the blades, erosion due to the presence of particulate, and corrosion due to the presence of alkali metals in the hot gas stream Ragland; Misra; Aerts; and Palmer, 1995, Wright; Leyens; and Pint, 2000, and Yuri; Hisamatsu; Etori; and Yamamoto, 2000. Of course the impact of such effects in the turbine life considerably increases the operation and maintenance costs. Gasification allows a higher quality fuel, though the cleaning process is still a costly part of the system.

A different approach is proposed in this paper, the so called externally fired gas turbine cycle, EFGT. This system is considerably advantageous when compared to the BIGGT system, as will be seen later on in this paper. The principle of an EFGT cycle is as follows (Figure 10): clean filtered air is compressed in a compressor, after what it passes through a heat exchanger, receiving heat from the hot gases leaving the combustor, the hot air is then expanded in a turbine, and the air from the turbine is directed into the combustor.

This paper will carry out a design point performance analysis of both the BIGGT and the EFGT cycles. One variant for each cycle is also studied, the intercooled externally fired cycle, ICEFGT, and the intercooled/recuperated integrated gasification/gas turbine cycle, BIGICR.

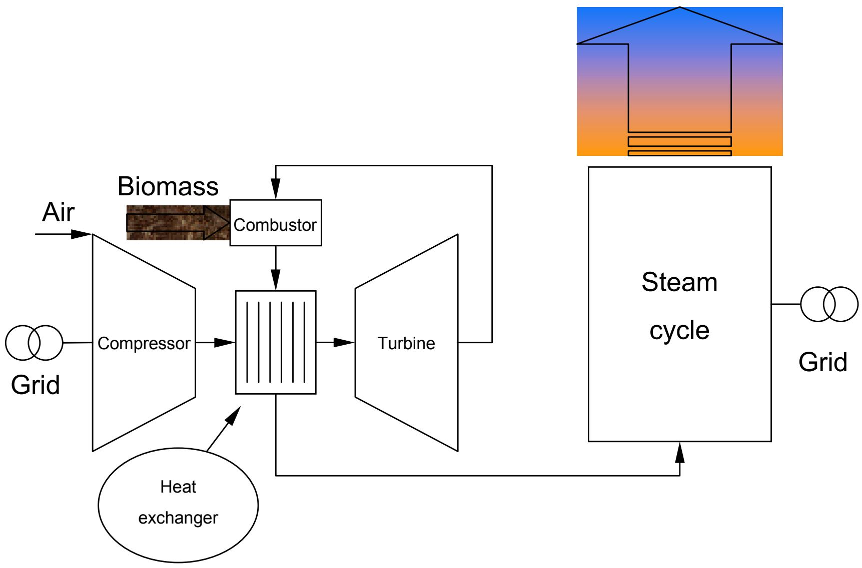

Figure 10 - Schematic of an EFGT cycle

2. The BIGGT cycle

There are two ways of burning biomass in a gas turbine engine: direct and indirect firing. The first can be subdivided into two subclasses, in one of them the solid fuel, biomass, is burnt in a combustor and the combustion products go to the expander. In this kind of cycle the high ash content of the combustion products will seriously damage the turbine blades, consequently turbine life is drastically shortened.

The second subclass is the BIGGT cycle, which is the reference case analysed in this paper. In this cycle the biomass is converted into a gas and then sent to the combustor. This system suffers from some major drawbacks. The gasification and cleaning system severely penalise the overall cycle efficiency. The gasifier introduces a large efficiency loss and the contaminants in the gas leaving the gasifier must be removed, otherwise those contaminants (ash, tar, alkali metals, particulate, etc.) will damage blades, cause blockage of injectors, valves, and filters.

The gas cleaning can be carried out at either low or high temperature, each with its drawbacks. These issues have already been well discussed by many authors including Larson and Williams, 1988, Bridgwater, 1995, and Consonni and Larson, 1996.

Furthermore the calorific value of the fuel can deteriorate the performance of the engine, when it has not been designed to this kind of fuel. Due to the very high fuel flow, the components operate within regions of lower efficiencies in respect to their design characteristics (Mathieu and Pilidis, 1991). This may reduce the overall efficiency of the engine unless an expensive redesign is undertaken. There is also a handling risk; because the change in the mass balance between the compressor and turbine causes the pressure ratio to rise and the compressor to surge. This leads to the use of bleed valves and/or VIGVs (variable inlet guide vanes) or alternatively to the need of redesigning the turbine nozzle guide vanes.

3. The BIGICR cycle

This cycle consists of a two stage intercooled compression process, a heat exchanger to recuperate the heat from the gas turbine exhaust, a combustion chamber, an expander, and a power turbine. In this paper the fuel for the engine comes from a gasification island, as in the BIGGT case.

This kind of cycle has the drawback of being bulk, but in the present analysis this is not considered a problem since the gasification island is itself much bigger than the gas turbine engine. The main advantage of the BIGICR cycle is its high efficiency due to the heat recovered from the exhaust gases. Although this high efficiency is penalised by the gasification system, when compared to the BIGGT cycle, it proves to be more efficient for pressure ratios of up to 32. The BIGICR reaches the highest efficiency when the overall pressure ratio is equally split between the two compressors, as demonstrated for the simple cycle in Cohen; Rogers, and Saravanamuttoo, 1996.

4. The EFGT cycle

The EFGT cycle, as aforementioned, consists of a compressor, a ceramic heat exchanger, a gas generator turbine, a power turbine, and a combustor (Figure 10). The air delivered by the compressor receives heat energy from combustion gases in the heat exchanger, then the air is expanded in the gas generator turbine to provide work to the compressor, goes to the power turbine to generate useful work, after what is directed into the combustor. Finally, the combustion products go to the ceramic heat exchanger to provide heat to the compressed air.

The main drawback of this approach, when compared to the conventional cycle fuelled by fossil fuels, is the low efficiency when a high temperature heat exchanger is not used (Ferreira; Pilidis; and Nascimento, 2001). Due to the constraints of the heat exchanger, the turbine entry temperature is much lower than that of the conventional cycle (Larson and Williams, 1988). The development of heat exchangers that can withstand higher temperatures at reasonable costs will change this situation, allowing efficiencies as good as those of a conventional cycle. A controllable problem (Ranasinghe and Reistad, 1987) is related to the deterioration of the heat exchanger performance due to slagging - deposition of solid particulates in the ceramic heat exchanger tubes.

On the other hand there are several advantages in using the EFGT cycles fuelled by biomass. The first is that the gasification system is no longer needed. As already said this device is very costly and introduces large losses in the overall cycle efficiency, not to talk about its bulk. The second advantage is the versatility of the combustion chamber, i.e., many different fuels can be burnt, allowing the use of that which is the cheapest or readily available. The pre-treatment needed is modest when compared to other cycles. These points are of great relevance when considering biomass fuels. Finally, the working fluid is clean air, what means that the maintenance costs will drop and the engine lifetime will be augmented (Ferreira and Pilidis, 2001).

Once the turbine inlet temperature, TIT, is imposed by the heat exchanger capacity, this device must be capable of withstanding very high temperatures for the sake of the cycle efficiency. Many researchers and research centres have been trying to develop a heat exchanger specifically to be used in EFGT cycles (Lahaye and Zabolotny, 1989, Ranasinghe; Aceves-Saborio; and Reistad, 1989, Solomon; Serio; Cosgrove; Pines; Zhao; Buggeln; and Shamroth, 1996, and Jolly; O'Doherty; and Bates, 1998).

Conventional materials used in high temperature heat exchangers, such as super-alloys, are not suitable for highly efficient EFGT cycles, the maximum temperature these materials can withstand is around 900^{o}C. The use of ceramics has been seen as the most promising solution to the problem. Although some researchers say that state-of-the-art ceramics are still unsuitable for use in EFGT engines (Solomon et al., 1996), there are records of ceramic heat exchangers working with products from coal combustion for approximately 500 hours at temperatures of up to 1535°C (1808K), without distress (LaHaye and Zabolotny, 1989). This EFGT cycle presented thermal efficiency of 35%, with pressure ratio of 9, and TIT of 1150°C (1423K). With the development of ceramic materials and techniques to minimise the damage caused by fuel contaminants it is reasonable to expect that in the near future high efficiency EFGTs may be commercially available. It is worth to point out that the experimental time of operation above is still unsatisfactory to have a clear view of the ceramic heat exchanger performance, though the same authors mention the successful use of ceramic heat exchangers in other industries.

Jolly and others, 1998, proposed the use of bayonet tube arrangements in the ceramic heat exchangers for externally fired gas turbines. A bayonet element is made of two concentric tubes, the outer tube is plugged at one end and the inner tube is open at both ends. The air enters the inner tube and reverses its flow at the end to travel through the annulus (or vice-versa). The combustion products flow across the bayonet elements in the shell side of the heat exchanger. The use of such elements is justified by their capacity to expand or contract under the influence of very large temperature differences, minimising thermal stress.

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 in this paper the heat exchanger outlet temperature is constrained at 1350 K, meaning temperatures of 1435 K in the hot side inlet of the heat exchanger for the EFGT simple cycle and 1446 K for the same stream in the ICEFGT.

The EFGT variant assessed in this work, ICEFGT, is basically the EFGT cycle with the compression process equally divided in two steps, being an intercooler placed between them. Again, as in the BIGICR cycle, the cycle is optimised for an equally split pressure ratio between the two compressors.

The very high exhaust temperatures also suggest that this cycle is suitable for combined gas/steam cycles. Actually, the steam cycle can be optimized independently of the gas turbine cycle (and vice-versa), once the HRSG inlet gas temperature can be adjusted without interfering with the gas turbine engine (Lahaye and Zabolotny, 1989).

Intercooling and recuperation are hardly used in power generation cycles due to its high costs and bulk. Intercooling itself is a quite complicated addition to the system due to the large quantities of water needed (Cohen and others, 1996). However, on average water is not a problem in most regions of Brazil what makes the analysis of these cycles still valuable.

5. Results

Four cycles have been assessed, the BIGGT, BIGICR, EFGT, and ICEFGT. Figures 2 to 5 show the design point performance of each one. Table 12 shows the design point data chosen from the optimisation process. The turbine inlet temperature for the gasification cycles was chosen to be 1450 K, and for the externally fired cycles 1350 K. These figures have been chosen due to limitations in flame temperatures for the BIG cycles and material constraints in the EF cycles. The values for pressure ratios were chosen as the ones that give the highest thermal efficiencies under the given TIT.

Table 12 - Design parameters chosen for emission calculations

	BIGGT	BIGICR	EFGT	ICEFGT
OPR	18	15	6	9
η_c	0.88	0.88	0.88	0.88
$\left(\Delta P/P\right)_{cc}$	4%	4%	2%	2%
η_{gg}	0.89	0.89	0.89	0.89
η_{pt}	0.89	0.89	0.89	0.89
ε_{IC}	---	0.90	---	0.90
ε_R	---	0.90	---	---
ε_{CerHx}	---	---	0.90	0.90

The fuel chosen is a pre-compacted sugar cane bagasse called Bagatex (Codeceira-Neto and Pilidis, 1999), and its composition and respective gasification product is presented in Table 13.

Table 13 - Solid fuel and flue gas characteristics

Compounds (solid fuel)	Weight %	Flue gas	Volume %
C	23.50	N_2	48.40
H	3.25	CO	21.00
O	22.00	CO_2	9.70
W	50.00	H_2	14.50
A	1.25	CH_4	1.60
		H_2O	4.80
LHV [MJ/kg]	18.44	LHV [MJ/kg]	4.46

Figure 11 and Figure 12 provide a comparison between both the BIGGT and the BIGICR cycle. The thermodynamic advantages of the intercooling and recuperation are clearly apparent, providing a substantial increase in the thermal efficiency, η_{th}, and specific work of the BIGICR cycle, principally for low overall pressure ratios, OPRs. In the BIGGT case, the higher the pressure ratio - within the assumed boundaries for OPR and TIT - the higher the thermal efficiency for a given TIT. This is not the case for the BIGICR cycle, the presence of heat exchangers considerably changes the optimum OPR due to the pressure losses these devices pose to the system. However, it is worthy to point out again the high costs involved in the intercooled/recuperated cycles. So far, these systems have been used only in naval applications, where water is abundant and high performance is paramount. With the increase of fuel prices and severe restrictions on emissions, these cycles can become an economically viable alternative for electricity generation. Another characteristic of the BIGICR cycle is the capacity of keeping a small difference between the higher and the lower OPR, what can not be said the same of the BIGGT cycle.

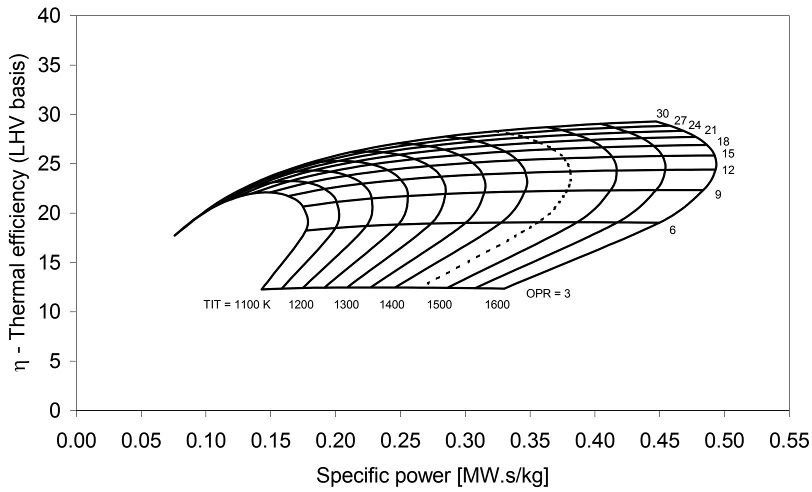

Figure 11 - Design point performance for the BIGGT cycle

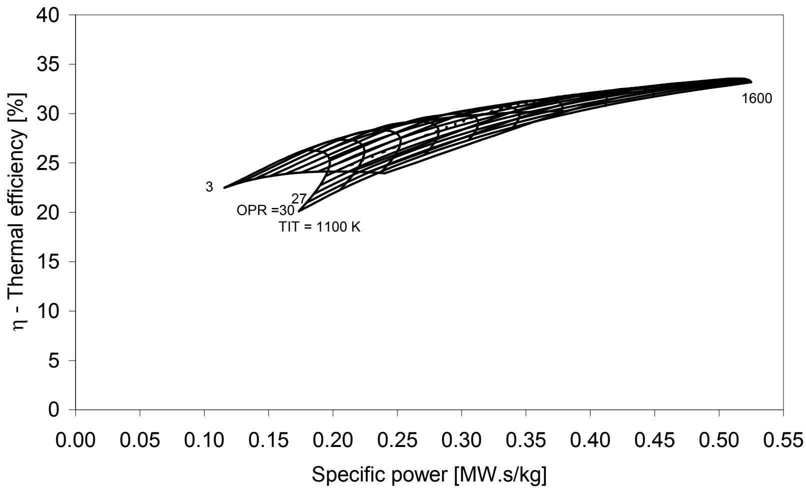

Figure 12 - Design point performance for the BIGICR cycle

Figure 13 and Figure 14 provide a comparison between the EFGT and the ICEFGT cycles. Again, the thermodynamic superiority of the intercooled cycle is clearly seen when compared to the simple cycle. The same characteristics of enhanced thermal efficiency and specific work are noticed in the ICEFGT cycle, though, as said before, such a system would be costly and demanding in terms of water needed in the intercooling heat exchanger. Also, the efficiency variation from one operating point to another is not as large as in the EFGT cycle.

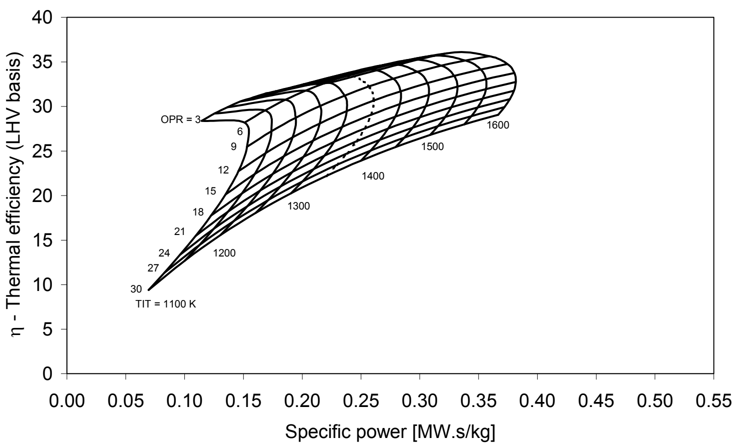

Figure 13 - Design point performance for the EFGT cycle

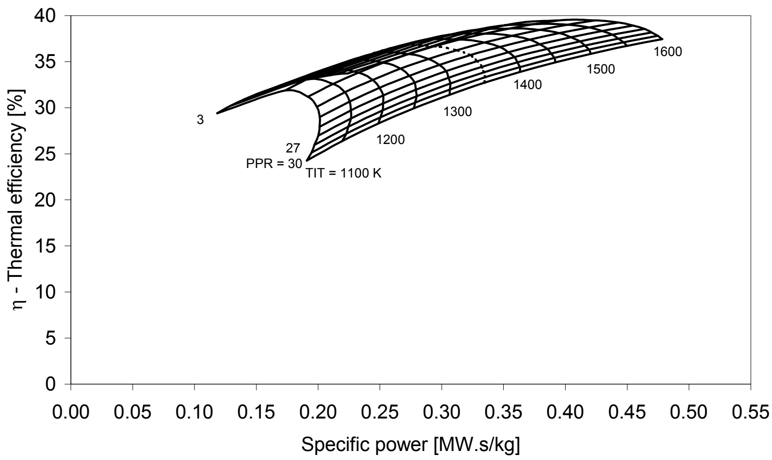

Figure 14 - Design point performance for the ICEFGT cycle

When comparing the two simple cycles, BIGGT and EFGT, the last presents a higher efficiency for low TITs than the BIGGT cycle. Due to the presence of the heat exchanger working as a recuperator, this cycle is more efficient at low OPRs. Although the heat exchanger will not be a compact device, the gasification island will be a quite bulk system compared to the gas turbine engine, making the EFGT system an attractive system for power generation in places where fuel supply can vary along the year. Both BIGICR and ICEFGT present superior performance at design point than the simple cycles, being the ICEFGT the one that reaches higher efficiencies. The dotted lines in each chart show the limitation in TIT imposed by technological issues such as flame temperature and materials.

6. Conclusion

Four gas turbine cycles and their performance at design point were presented. The modified cycles presented the best performance, being the ICEFGT cycle the one with the highest thermal efficiencies of all four.

In the simple cycle cases the EFGT is the one with the best performance. This brings a new system into the power generation scenario using clean biomass fuels. The EFGT allows fuel versatility and low costs, still contributing to reducing the greenhouse gases emissions.

With the advance of ceramics, the increase in fossil fuel prices, and emissions taxation it is possible to say that in the near future the EFGT, ICEFGT, and BIGICR will play a major role in the generation market.

7. Acknowledgements

The authors wish to thank CAPES/Brazil for its strong financial support to this work.

8. References

1. Bridgwater, A. V.; 1995; *The Technical and Economic Feasibility of Biomass Gasification for Power Generation*; Fuel; 74; (5); 631-653.
2. Codeceira-Neto, A. and Pilidis, P.; 1999; *A Comparative Exergy Analysis of Advanced Power Cycles Using Biomass Fuel*; Indianapolis, Indiana. USA.
3. Cohen, H.; Rogers, G. F. C., and Saravanamuttoo, H. I. H.; 1996; *Gas Turbine Theory*; Addison Wesley Longman Limited.
4. Consonni, S. and Larson, E. D.; 1996; *Biomass-Gasifier/Aeroderivative Gas Turbine Combined Cycles: Part A - Technologies and Performance Modeling*; Journal of Engineering for Gas Turbines and Power; 118; (July); 507-515.
5. Ferreira, S. B. and Pilidis, P.; 2001; *Comparison of Externally Fired and Internal Combustion Gas Turbine Using Biomass Fuel*; Journal of Energy Resources Technology - ASME; (December).
6. Ferreira, S. B.; Pilidis, P., and Nascimento, M. A. R.; 2001; *A Comparison of Different Gas Turbine Concepts Using Biomass Fuel*; New Orleans, LA, USA.
7. Ingham, J. M. *Energia da Biomassa* [Web Page]. 1998. Available at: www.bgtechnologies.net. Notes: in Portuguese
8. Jolly, A. J.; O'Doherty, T., and Bates, C. J.; 1998; *COHEX: a computer model for solving the thermal energy exchange in an ultra high temperature heat exchanger. Part A: computational theory*; Applied Thermal Engineering; 18; 1263-1276.
9. Lahaye, P. G. and Zabolotny, E. R.; 1989; *Externally Fired Combined Cycle (EFCC)*; Nice, France.
10. Larson, E. D. and Williams, R. H.; 1988; *Biomass-Fired Steam-Injected Gas Turbine Cogeneration*New York.
11. Magalhães, P. S. G. and Braunbeck, O. A. *Colheita de Cana de Açúcar Verde* [Web Page]. 1999. Available at: www.agrobrasil.com.br., Notes: in Portuguese
12. Mathieu, P. and Pilidis, P.; 1991; *The use of Gaseous Fuels on Aero-Derivative Gas Turbine Engines*; Orlando, FL.
13. Nogueira, L. A. H. and Walter, A. C. S.; 1995; *Experiências de Geração de Energia Elétrica a Partir de Biomassa no Brasil: aspectos técnicos e econômicos*; Montevideo, Uruguai.
14. Ragland, K. W.; Misra, M. K.; Aerts, D. J., and Palmer, C. A.; 1995; *Ash Deposition in a Wood-Fired Gas Turbine*; Journal of Engineering for Gas Turbines and Power; 117; (July); 509-512.
15. Ranasinghe, J.; Aceves-Saborio, S., and Reistad, G. M.; 1989; *Irreversibility and Thermoeconomics Based Design Optimization of a Ceramic Heat Exchanger*; Journal of Engineering for Gas Turbines and Power; 111; 719-727.
16. Ranasinghe, J. and Reistad, G. M.; 1987; *Use of Ceramic Heat Exchangers in Wood-Fueled Power Plants*.
17. Solomon, P. R.; Serio, M. A.; Cosgrove, J. E.; Pines, D. S.; Zhao, Y.; Buggeln, R. C., and Shamroth, S. J.; 1996; *A Coal-Fired Heat Exchanger for an Externally Fired Gas Turbine*; Journal of Engineering for Gas Turbines and Power; Vol. 118; (January); 23-31.
18. Wright, I. G.; Leyens, C, and Pint, B. A.; 2000; *An Analysis of the Potential for Deposition, Erosion, or Corrosion in Gas Turbines Fueled by the Products of Biomass Gasification or Combustion*; Munich, Germany.
19. Yuri, I.; Hisamatsu, T.; Etori, Y., and Yamamoto, T.; 2000; *Degradation of Silicon Carbide in Combustion Gas Flow at High Temperature and Speed*; Munich, Germany.