PROGRESS OF AN EXTERNALLY FIRED EVAPORATIVE GAS TURBINE CYCLE FOR SMALL SCALE BIOMASS GASIFICATION

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ABSTRACT
The present paper reports on a demonstration project supported by the THERMIE program of the European Commission and by the VLIET program of the Flemish Government. A CHP gas turbine plant fueled by product gas from a biomass fluidized bed gasifier has been constructed. The demonstration scale is 150 kW for production of power and heat for the university campus district heating. At the present stage 150 kW power has been delivered to the grid. Problems encountered and results achieved during the first startups of the power plant will be discussed. In the future some natural gas topping combustion will be included to overcome the temperature limitation of materials used in the metallic high temperature air heater. Water injection in this air heater will be included to enhance power output and to allow flexible power to heat ratios. The target commercial scale is 2 to 5 MWe using atmospheric gasification and external firing through a high temperature metallic heater.

KEYWORDS
externally fired cycle, gas turbine, gasifier, gasification, biomass

INTRODUCTION
Different routes for power production from biomass are now in development. Biomass can be burned with conventional techniques in boilers to produce steam for power and heat. Liquid and gaseous fuels can be produced from biomass through pyrolysis. Pressurized combustion of pulverized wood for use in a gas turbine is under consideration. For small to medium power ranges, alternatives are proposed where indirect firing of a gas turbine is considered. For smaller scales the combustion of biomass product gas in IC engines has been tried for many years, but reliable operation under market conditions is not attained so far.

The indirect gas turbine firing [1][2][3][4] is an intermediate solution between external firing of steam turbines and internal firing in gas turbines or reciprocating engines. The main incentive to proceed in this way is to reduce the gas cleaning problems encountered in internal firing and to take advantage of the potential higher efficiency of a gas turbine cycle.

The present project targets commercial application in the 2 to 5 MWe scale. A gas turbine cycle is proposed where the firing occurs indirectly by firing the product gas in the exhaust of the engine and recovering the heat through a metallic air heater. Using the latest available alloys (Haynes steel 120) tube wall temperatures up to 850°C can be attained (at 10 bar pressure), and it is expected that temperatures may raise above 900°C in the coming years. To overcome the insufficient turbine inlet temperature some topping natural gas combustion will be added at a later time to improve the plant economics. The use of a ceramic air heater [5][6] has been considered but appeared to be immature. Even if available, the cost of such a ceramic exchanger would probably be too high.

In previous papers [7][8] ASPEN simulations were carried out to predict the performance and output of the considered cycle. Evaporation and topping combustion were included in these simulations.

GENERAL DESCRIPTION OF THE INSTALLATION

Power plant layout
Figure 1 gives an overview of the power plant. The different parts and their characteristics will be discussed in the following paragraphs.
Gasifier

The gasification plant is drawn on Fig. 2. The wood is fed from a one day capacity silo to the fluidized bed gasifier at a flow rate of some 400 kg/h. The equivalence ratio \( \phi \) ranges between 0.25 and 0.3 producing a gas with a caloric value of 3.5-4 MJ/kg (excluding tars). The bed temperature is about 700°C and drops to about 625°C in the freeboard. The freeboard has an expanded volume in relation to the bed in order to increase the residence time of the fly ash by which the carbon loss is minimized.

The principal design considerations for the fluidized bed are related to efficient fluidization and design flowrate of the biomass (Table 1). Experience has shown that in practice the ratio between gas and minimum fluidization velocities, which offers a measure of solid mixing, should be about 10. Especially with feedstocks of low density and small particle diameter such as biomass residues which tend to float at the surface of the bed it is very important to maintain a mixing as good as possible to avoid agglomeration of the bed material. In addition the biomass feeding point is located close to the distributor in order to allow sufficient residence time for the particles.

Table 1: Fluidization parameters

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Unit</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Mean particle diameter</td>
<td>mm</td>
<td>0.50</td>
</tr>
<tr>
<td>Min. fluidization velocity</td>
<td>m/s</td>
<td>0.08</td>
</tr>
<tr>
<td>Gas velocity</td>
<td>m/s</td>
<td>0.80</td>
</tr>
<tr>
<td>Particle terminal velocity</td>
<td>m/s</td>
<td>5.00</td>
</tr>
<tr>
<td>Bubble diameter max</td>
<td>m</td>
<td>0.50</td>
</tr>
<tr>
<td>Bubble diameter min</td>
<td>m</td>
<td>0.25</td>
</tr>
<tr>
<td>Bed height</td>
<td>m</td>
<td>0.60</td>
</tr>
<tr>
<td>Bed diameter</td>
<td>m</td>
<td>0.80</td>
</tr>
</tbody>
</table>
The dust in the producer gas is removed by means of a single cyclone. A dust content lower than 500 mg/Nm³ has been specified. Alkali and chlorine contents are depending on the nature of the biomass but should be minimal. The gas pressure is 400 mm H₂O. The expected producer gas temperature is 600°C which is on one side low enough to avoid further cracking of the tars and on the other side high enough to avoid tar condensation. Deposition and corrosion will be evaluated to determine if stricter gas cleanup measures are necessary. Figure 3 shows the gasifier.

**Gas turbine cycle**

The direct firing of the product gas from biomass in a small gas turbine is still difficult to achieve and external firing is considered as an alternative (Fig. 4). The external firing reheats the turbine exhaust gas (air) and the energy is recovered in front of the turbine by a high temperature heat exchanger. Although large in size, the cost of the metallic heat exchanger itself does not exceed 10% of the total project cost.

![Figure 4: Indirect firing of a gas turbine using an air heater](image)

Figure 4 shows a close-up of the gas turbine. The main design parameters of the gas turbine cycle are shown in Table 2.

![Figure 5: The gas turbine](image)

**Table 2: Main design parameters**

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Turbine back pressure</td>
<td>60 mbar (gauge press.)</td>
</tr>
<tr>
<td>Turbine inlet temperature</td>
<td>800 to 1000°C</td>
</tr>
<tr>
<td>Approx. design pressure ratio</td>
<td>8</td>
</tr>
<tr>
<td>Cold side pressure loss</td>
<td>0.5 bar</td>
</tr>
<tr>
<td>Hot side pressure loss</td>
<td>15 mbar</td>
</tr>
<tr>
<td>Hot side temperature difference</td>
<td>100°C</td>
</tr>
<tr>
<td>Maximum material temperature</td>
<td>850°C</td>
</tr>
<tr>
<td>Water heater</td>
<td></td>
</tr>
<tr>
<td>Hot side pressure loss</td>
<td>15 mbar</td>
</tr>
<tr>
<td>Stack temperature</td>
<td>180°C</td>
</tr>
</tbody>
</table>

**The external combustor**

The external combustor consists of a standard natural gas pilot burner and injection of the product gas in the comburant air, as shown in Fig. 6.

![Figure 6: External combustor](image)

The standard burner is able to operate the plant with natural gas, which facilitates startup and operation in case of gasifier outages. The natural gas burner also ensures a stable flame and allows the temperature control.
Low and high temperature heat exchangers

For the air heater, a temperature difference of 100°C between the inlet of the combustion gases and the outlet of the air was postulated during the design. The front tubes of this air heater are made of Haynes 120 alloy (Ni-3Co-33Fe-25Cr-2.5Mo-2.5W). The remaining tubes are finned and made of Hastelloy 800 HT and Avesta 253MA. The hot collector and hot duct to the turbine are made of two concentric tubes. The inner tube is made of sleeving pieces of 321 Steel. Since the temperature of the combustion gases at the exit of the air heater is high enough, a water/gas heat exchanger is installed for the low temperature recuperation. The upper part of this heat exchanger will be used in the evaporation unit. The lower part is utilized for district heating purposes. The stack temperature is kept relatively high to avoid acid corrosion problems.

The external combustor, low and high temperature heat exchangers are arranged into one piece of equipment as shown in Fig. 7.

Evaporation unit

Both efficiency and power of the gas turbine cycle can be improved in combined-cycle plants, STIG and evaporation cycles such as HAT, CHAT and REVAP cycles. In the present system, which is too small for combined cycles and where no steam is produced, injection and evaporation of preheated liquid water is considered for boosting power and efficiency. It is to be observed that the choice of injecting water in the air heater offers the best option for maximum heat recovery [9]. Direct injection of water in the cold side tubes of the air heater is designed, but not yet realized in the present stage.

Topping combustion

In the original project (Fig. 1), a topping combustor burning some natural gas was considered but has not yet been realized due to excessive cost. This topping combustor will nevertheless be included in the near future for different reasons which are easier startup, easier temperature control and high marginal efficiency by raising the turbine inlet temperature.

PLANT COST AND COST STRUCTURE

The total cost of the project is about 3 Mecu. The cost structure of the project is shown in Table 3. The cost and the cost structure for a similar power plant scaled up to 2.5MWe has been studied in [7].

Table 3: Cost structure of the power plant

<table>
<thead>
<tr>
<th>Component</th>
<th>Cost (%)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Gas turbine</td>
<td>25%</td>
</tr>
<tr>
<td>Gasifier and peripherals</td>
<td>27%</td>
</tr>
<tr>
<td>HT heat exchanger</td>
<td>10%</td>
</tr>
<tr>
<td>Ductwork</td>
<td>6%</td>
</tr>
<tr>
<td>Combustor</td>
<td>6%</td>
</tr>
<tr>
<td>LT heat exchanger</td>
<td>4%</td>
</tr>
<tr>
<td>Civils, electric and other</td>
<td>22%</td>
</tr>
</tbody>
</table>

CONTROL AND MEASUREMENT

The gasification and turbine plants can operate independently of each other. The gasifier starts by preheating the gasification air and venting the exhaust to the flare. Once the gasifier bed is preheated the biomass feeding is automatically enabled. When normal operation is achieved, the air flow is controlled by a valve to keep the temperature constant. The air flow is measured by two venturi’s: the main air flow and extra air flow which is injected below the lock to prevent the gas from escaping through the injection screw. The biomass flow is controlled manually by varying the speed of the dosing screw. The calibration of the biomass flow must be done manually for each type of biomass fuel. A flange for catching and weighting the biomass is present in front of the lock. The gasifier is stopped by cutting the biomass flow feeder.

The turbine starts on natural gas. A hydraulic starter spins the turbine up to 30% speed. In order to reduce the thermal stresses in the air heater, the heating rate is limited. At 100% the speed is kept constant by a bleed valve on the compressor discharge, allowing the grid connection to be realized. Once on the grid the gas flow is regulated by the temperature demand in the external combustor. The air flow in the gas turbine cycle is measured in the stack by means of a diaphragm flow meter.

The biomass can be admitted to the combustor by a manual command on the gasifier control panel, provided the turbine control gives the permission to do so.
TEST RESULTS

History

Table 4 gives some milestones of the tests done up to the time of writing.

Table 4: Milestones

<table>
<thead>
<tr>
<th>Date</th>
<th>Event Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>20/04/96</td>
<td>First startup gas turbine (natural gas)</td>
</tr>
<tr>
<td>12/12/96</td>
<td>First successful operation of the gasifier</td>
</tr>
<tr>
<td>07/08/97</td>
<td>First grid connection (5 minutes)</td>
</tr>
<tr>
<td>26/09/97</td>
<td>Second short grid connection</td>
</tr>
<tr>
<td>06/10/97</td>
<td>First successful admission of biogas</td>
</tr>
<tr>
<td>09/10/97</td>
<td>First performance evaluations at no load</td>
</tr>
<tr>
<td>08/12/97</td>
<td>Full automatic run of the gasifier (5 hours)</td>
</tr>
<tr>
<td>20/02/98</td>
<td>First official demonstration</td>
</tr>
<tr>
<td>20/03/98</td>
<td>First 150 kW on biogas (2 hours)</td>
</tr>
</tbody>
</table>

Results gasifier

Figure 8 gives an overview of the measurements made at the 23rd of March 1998. The heating rate is about 2°C/min. Above a bed temperature of 400°C the heating rate accelerates since charcoal was still present in the bed from the previous runs. This charcoal burns thus preheating the bed up to 650°C. On Fig. 8 we can see that after 3.5 hours the preheating of the air is cut off at a bed temperature of approximately 650°C. At a bed temperature of +/- 750°C normal operation is achieved and the control system keeps the bed temperature constant by regulating the air flow. After 5 hours the gasifier was cut off by stopping the wood feed. The air feed stops when the bed temperature goes below +/- 650°C. In this way enough charcoal is kept in the bed for the next startup. The cut off of the air feed can be seen on Fig. 8 by observing the behaviour of the pressure drop over the gasifier bed.
Results gas turbine cycle

The measurements made at the 20th of March 1998 are shown in Fig. 9. The turbine was started on natural gas and accelerated to 30% speed by means of the hydraulic starter. From then on the heating rate was limited to 20°C/min leading to a startup time of 45 minutes. After circa one and a half hour (95% speed) biogas from the gasifier was admitted and grid connection was made. The temperature of the tubes of the air heater increased strongly and it was decided to shut down the biogas flow. The plant ran further on natural gas until closing.

During the test heat was recovered in the low temperature heat exchanger to heat up water. The water flow equals to circa 20 m³/h. The temperature profiles of the water at inlet and outlet can be seen on Fig. 9. Stack temperature was circa 180°C as designed, while ambient temperature was about 10°C.

The turbine inlet temperature reached a maximum temperature of approximately 700°C which means that the turbine is in off design. During the test 150 kW electricity was delivered to the grid for approximately 2 hours.

TECHNICAL PROBLEMS AND SOLUTIONS

Gasifier

On the overall, the gasification plant has operated without major problems. A critical point is the feeding system. One should provide a good accessibility for maintenance. The feeding point inside the gasifier was cantilevered causing premature wearing. A solution for this problem is under consideration. Manpower and safety requirements for the ash collection have been underestimated and some extra investment is planned to improve this situation. The solid effluent consists of 50% carbon and 50% mineral ashes, which is not straightforward to dispose. An in situ post combustion through a new type of cyclonic combustor is planned.

Turbine

Heat and mass transfer (leakage) is found from compressor discharge to turbine inlet, inside of the gas turbine casing. This heat transfer results in a temperature drop from the exit of the air heater to the very inlet of the expansion wheels, and heat thus bypasses the air heater. During the design, it was assumed that this heat transfer would result in a 10°C drop, but the analysis indicates a drop up to 30°C. This results in a significant drop in efficiency and power. Possible solutions are to include a good insulation in the inner ductwork, but this is probably difficult to achieve in the type of aero-engine used in the project. To solve the problem in future application, engines should be selected where compressor and turbine ductwork are completely separated, and where the turbine inlet ductwork is cooled as little as possible. Recent turbines equipped with recuperators are candidates for this (e.g. Mercury engine from Solar [10]). Another possible solution is to use low cost turbochargers which are now available in the right power ranges and are becoming cheaper and getting higher performances.
Indirect firing

The first burner tests failed because the air flow was too low at start running speed. This problem was solved by adding a startup fan which yields 100% flow during startup and which is shut around 80% speed. Also a non-symmetric temperature distribution was found on the metallic high temperature heat exchanger due to the bends upstream the external combustor. The problem was solved by making CFD calculations which led to the installation of vanes and grids in the bends.

Special attention has to be paid to the hot outlet collector of the high temperature heat exchanger (Fig. 10). It consists of two concentric cylinders which are insulated (partially air, partially ceramic wool) in respect to each other. The tubes of the heat exchanger are connected with the inner cylinder of the collector and are insulated from the outer cylinder. The thickness of the insulation between the cylinders was determined by the means of a finite element analysis in order to get a uniform temperature field on the outer cylinder. During startup this piece of equipment can be subjected to strong non-axisymmetric temperature distributions. Figure 10 shows the steady state temperature distribution in a typical section of this collector based on the nominal operation conditions. As a result of this temperature distribution, thermal stresses will be induced. The transient temperature distribution and their induced stresses is a topic which is studied at this moment.

![Figure 10: Temperature distribution in the hot collector](image)

PERFORMANCE AND OPERATING COSTS

Performance

Since operation at design point of the turbine (1000°C) was not yet achieved, it is not possible to give final data about global performance. The project has however demonstrated some problems related to the external firing which affect the performance, and which are under consideration in the continuation of the project. The consumption of the peripherals is also known.

So far only the composition of the product gas and the ashes have been measured, with exclusion of the tar content in the product gas. Only a small percentage of the carbon is not converted and found in the ash bin, which means that the conversion rate of the exceeds 95%. The gas analysis is found to correspond to the typical values in literature. Since the tar content is not yet measured, it is not possible to mention the exact calorific value of the gas. Since the gas is kept hot and tars are burnt in the turbine cycle, most of the energy from the gasifier is however recovered and the losses mainly result from the non-converted carbon (a few %), the heat losses (unknown, probably also a few %) and the peripheral consumption.

The performances calculated with ASPEN are summarized in Fig. 11 and 12, where efficiencies are shown versus amount of water injection in the case of no top firing (Fig. 11) and maximum top firing (Fig. 12). Details of these calculations are available in [7]. At present, no water injection or top firing is installed and the first electric performance target to be achieved corresponds to the left side of Fig. 10, which amounts to about 13% efficiency.

The first performances were analysed at zero load 100% speed. An abnormal high fuel consumption was observed, which to a large extent is due to non-uniform temperatures in the front of the air heater, and to some extent to excessive heat losses. The performance is also affected by mass and heat transfer between compressor discharge and turbine inlet, which thus bypass the turbine.

![Figure 11: Target performance in absence of top firing](image)

![Figure 12: Target performance with maximum top firing](image)
Operating costs

The yearly operating costs could not yet be determined. Some indications can however be given:

- Attempts to get wood fuel on a regular base were disappointing. The cost of the wood fuel appears to be much higher than expected, and the candidate fuel which was saw dust has a high demand for other applications (particle boards, beds and others). Potential negotiations failed on the base of the high risk incurred, the VUB not being able to guarantee the continuous acquisition of the fuel.

- The costs of transporting and handling both the wood fuel and the ashes for such a small installation were underestimated. Much attention should be given to the easy delivery and minimum handling of both fuel and ashes, and the scale should be increased to reduce these costs.

- It is not clear yet to what extent the plant can be operated automatically, in absence of a permanent operator. In the case of a permanent staff member the labor operating cost is dominating and probably uneconomic. The scale should be increased and the operator should be shared with other tasks to have a viable operating cost.

ECONOMIC VIABILITY

There is not enough operating experience so far to give a reasonable figure of the economic viability of the externally fired system. It is clear however that the specific investment cost should be decreased significantly by increasing the scale (2 - 5 MWe), and by using cheaper turbomachinery. The original idea to use wood fuel from local suppliers clearly makes no chance and the project should be applied in companies producing their own fuel, thus avoiding problems of supply and transport costs.

Running the plant in the near future will occur with biofuels imported from the south of Europe: residues from olive and grape pressing, almond shells and rice husk. These fuels can be obtained at a cost which is below 5 US cents per kg dry matter, excluding the transport cost.

CONCLUSION

Different problems affecting the technical success of the project have been detailed in the previous sections. On the overall, the gasifier is operated successfully and observed problems can be solved. The indirect fired cycle is however still subject to further development: problems of bypass in the turbine and uniformity of the flow in the air heater seem to be the most important issues which affect the performance significantly.

Top firing and water injection have not been achieved yet, and will be tackled only when the plant will operate properly under present conditions.

NOMENCLATURE

Letter symbols

- CHP Combined Heat and Power
- HAT Humid Air Turbine
- CHAT Cascaded Humidified Advanced Turbine
- REVAP Regeneration EVAPoration
- AF air-fuel ratio (mass flow)
- $\phi$ equivalence ratio defined as follows: $AF/(AF)_{st}$

Subscripts

- st stoichiometric
- e electric

BIBLIOGRAPHY