DEVELOPMENT AND PERFORMANCE OF TURBOCHARGER BASED MICRO GAS TURBINE SYSTEM USING BIOMASS FUEL

by

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LIST OF SYMBOLS

		Units
A	Surface area	m^2
A_s	Surface area for shell side	m^2
A_t	Surface area for tube side	m^2
A_p	Cross section area before and after the pipe bundle	m^2
C	Heat capacity rate	W/K
C*	Heat capacity ratio	_
Cp	Specific heat capacity	J/kg.K
d_i	Pipe inner diameter	m
d_p	Particle diameter (cyclone separator design)	µm
d_o	Pipe external diameter	m
D_h	Hydraulic diameter	m
D_s	Shell pipe diameter	m
e	Surface roughness magnitude	mm
f	Friction factor (pressure drop calculations)	_
GMW	Gram molecular weight of the pollutant	gram
G	Mass flow rate per unit area	$kg/m^2.s$
Gr	Grashof number	_
Gs	Graetz number	_
h	Enthalpy	J/kg
H	Cyclone separator inlet height	m
h_s	Heat transfer coefficient for the tube side	$W/m^2.K$
h_t	Heat transfer coefficient for the shell side	$W/m^2.K$
K_a	Thermal conductivity of the air	W/m.K
k_c	Compression pressure drop factor in the pipe bank inlet	_
k_e	Expansion pressure drop factor in the pipe bank outlet	_
K_g	Thermal conductivity of the combustion gases	W/m.K
K_w	Thermal conductivity of the pipe wall	W/m.K
L	Total length of the heat exchanger	m
$\dot{m}$	Mass flow rate	kg/s
$\dot{m}_a$	Air mass flow rate	kg/s
$\dot{m}_g$	Mass flow rate of the combustion gases	kg/s
$\dot{m}_{output}$	Output air mass flow rate	kg/s
$\dot{m}_{pg}$	Producer gas mass flow rate	kg/s
Mp	Measured weight of the pollutant	mg

N_e	Number of effective turns	_
N_H	Number of velocity inlet heads (cyclone separator calculations)	_
N_t	Total number of pipes (heat exchanger design)	_
Nu	Nusselt number	_
P	Pressure	bar
P_r	Relative pressure (ideal brayton cycle calculations)	_
Pr	Prandtl number (heat exchanger design)	_
Pt	Distance between two pipes (heat exchanger design)	mm
Q	Power	W
r	Thermal conductivity resistance	$m^2.K/W$
R	Gas constant	kJ/kg.K
Re	Reynolds Number (heat exchanger design)	_
Rfa	Fouling resistant for air	$m^2.K/W$
Rfg	Fouling resistant for combustion gases	$m^2.K/W$
S	Entropy	J/kg.K
U	Over all heat transfer coefficient	$W/m^2.K$
T	Temperature	°C
V_i	Gas inlet velocity (cyclone separator design)	m/s
W	Shell side width	m
w	Wall thickness	m
W	Cyclone separator inlet width	m
ρ_g	Gas density	kg/m^3
ρ_p	Particle density	kg/m^3
ρ_{in}	Air density at the inlet of heat exchanger (pressure drop calculations)	kg/m^3
ρ_m	Air mean density (pressure drop calculations)	kg/m^3
ρ_{out}	Air density at the exit of heat exchanger (pressure drop calculations)	kg/m^3
μ_a	Viscosity of air inside heat exchanger	Pa.s
μ_g	Viscosity of the gas	Pa.s
σ	Reduction (or increment) ratio	_
ΔP_s	Pressure drop in the shell side (pressure drop calculations)	Pa
ΔP_t	Pressure drop in the tube side (pressure drop calculations)	Pa
Δt	Gas residence time inside the cyclone separator	s
ΔT	Temperature difference	°C
ΔT_{max}	Maximum temperature difference	°C
η	Efficiency	%
η_j	Particle collection efficiency (cyclone separator design)	%

LIST OF ABBREVIATION

A/F	Air Fuel Ratio
BAT	Biomass Air Turbine
BFBC	Bubbling Fluidized Bed Combustor
BFBG	Bubbling Fluidized Bed Gasifier
BIGCC	Biomass Integrated Gasifier Combined Cycle
BIGGT	Biomass Integrated Gasifier-Gas Turbine Cycle
BIGMGT	Biomass Integrated Gasifier-Micro Gas Turbine Cycle
BIGICR	Intercooled/ Recuperated Biomass Integrated Gasifier-Gas Turbine Cycle
BIFRCC	Biomass Integrated fired Recuperated Combined Cycle
BIPCC	Biomass Integrated Post Combustion Combined Cycle
CFD	Computational Fluid Dynamics
CHP	Combined Heat and Power
DG	Distributed Generation
DFGT	Direct Fired Gas Turbine
DFMGT	Direct Fired Micro Gas Turbine
EFGT	Externally Fired Gas Turbine
EFMGT	Externally Fired Micro Gas Turbine
EvGT	Evaporative Gas Turbine
FBC	Fluidized Bed Combustor
FBG	Fluidized Bed Gasifier
GTCC	Gas Turbine Combined Cycle
g/m^3	Gram of Pollutant per Cubic Meter (Measured at standard Pressure and Temperature)
HAT	Humid Air Turbine
HHV	Higher Heating Value
HRU	Heat Recovery Unit
IC	Internal Combustion Engine
ICECC	Internal Combustion Engine Combined Cycle
ICEFGT	Intercooled Externally Fired Gas Turbine
IGCC	Integrated Gasifier Combined Cycle
kW_e	Kilowatt Electrical
kW_{th}	Kilowatt Thermal
HV	Heating Value (used as lower heating value for producer gas)

LHV	Low Heating Value (used for wood)
LPG	Liquid Petroleum Gas
MC	Moisture Content
MGT	Micro Gas Turbine
NTU	Number of Transferred Units
PCC	Pressurized Cyclone Combustor
PG	Producer Gas
PFBC	Pressurized Fluidized Bed Combustor
PFBG	Pressurized Fluidized Bed Gasifier
ppm	Parts Per Million
rpm	Revolution Per Minute
SOFC	Solid Oxide Fuel Cell
SS	Stainless Steel

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PEMBANGUNAN DAN PRESTASI SISTEM TURBIN GAS MIKRO BERDASARKAN PENCAS TURBO MENGGUNAKAN BAHAN API BIOJISIM

ABSTRAK

Bahan api alternatif adalah keutamaan dalam bidang penyelidikan tenaga, kerana masalah dari kekurangan bahan api fosil dan pencemaran persekitaran. Biojisim merupakan salah satu tenaga boleh diperbaharui yang penting untuk termal dan penjanaan kuasa, terutamanya di Malaysia di mana sisa biojisim adalah banyak. Bagi penjanaan kuasa elektrik, minat terpusat di dalam penjanaan teragih (DG) dengan kebaikan berbanding dengan penjanaan, baru-baru ini meningkat di beberapa negara. Enjin pembakaran dalam, turbin gas mikro (MGT) dan turbin angin adalah calon utama untuk teknologi DG. Biojisim penggas alir bawah atau lapisan terbendalir dengan enjin diesel atau enjin gas salingan telah menunjukkan keputusan yang menggalakkan. Tetapi, masalah utama dengan sistem ini adalah kos penyelenggaraan, kerana gas yang dihasilkan dari biojisim mesti dibersihkan, disejukkan dan dikeringkan sebelum digunakan dalam enjin pembakaran dalam.

Penyelidikan ini merupakan pembangunan dan pencirian sistem turbin gas mikro untuk termal dan penjanaan kuasa (CHP) menggunakan gas yang dihasilkan dari biojisim (PG) sebagai bahan api. PG dibersihkan di dalam unit pembersihan panas berkos rendah yang terdiri daripada pemisah pusar tertebat. Haba deria dari PG panas dikekalkan sebagai haba tambahan untuk sistem, dan juga untuk mengekalkan tar di dalam PG di dalam keadaan wap. PG kemudian dibakar sepenuhnya dalam ruang pembakaran pusar bertekanan (PCC). Gas ekzos pembakaran kemudian dimasukkan ke dalam MGT peringkat duaan. MGT ini telah dibangunkan berdasarkan pada dua pencas turbo kenderaan, pengurangan kelajuan unit kapi dan penjara elektrik kelajuan rendah. Pencas turbo dengan saiz yang berbeza diuji semasa pembangunan MGT untuk mencapai prestasi terbaik. Gas ekzos kemudian

dipulihkan menggunakan unit pemulihan haba dua laluan (HRU) untuk pengeluaran udara panas. HRU ini direka berdasarkan aliran berlawanan dua laluan berasingan annulus paip tatarajah untuk mencapai efisiensi termal yang tinggi.

PCC telah dioptimumkan untuk pembakaran PG dengan menggunakan perisian simulasi CFD Fluent. Sistem telah diuji secara eksperimen dalam tiga mod operasi. Mod pertama adalah dengan sistem CHP didorong oleh 100% PG. Penggas alir bawah adalah ditekan hingga 1.1barg. MGT dua tahap ini dengan HRU telah mencapai 1kW$_e$ bekalan elektrik dan 35kW$_{th}$ kuasa haba, dengan kecekapan sistem keseluruhan 44.7%. Dalam mod operasi kedua, gas petroleum cair (LPG) digunakan dengan PG pada keadaan atmosfera dalam operasi bahan api duaan. MGT dua tahap ini dengan HRU telah mencapai 0.5kW$_e$ bekalan elektrik dan 34kW$_{th}$ kuasa haba, dengan kecekapan sistem keseluruhan 18%. Dalam mod operasi ketiga, MGT satu tahap digunakan untuk pengeluaran udara panas menggunakan PG pada keadaan atmosfera. Sistem telah mencapai 34kW$_{th}$ kuasa haba, dengan kecekapan sistem keseluruhan 37.5%. Sistem ini telah mencapai pencemar CO dan NOx rendah di bawah 115 dan 245ppm untuk semua mode operasi.

DEVELOPMENT AND PERFORMANCE OF TURBOCHARGER BASED MICRO GAS TURBINE SYSTEM USING BIOMASS FUEL

ABSTRACT

Alternative fuels are a priority in energy research field, due to issues of fossil fuel depletion and environmental degradation. Biomass is an important renewable energy fuel source for thermal and power applications, especially in countries like Malaysia where abundant biomass waste available. As for electrical power generation, interest has recently increased in small scale distributed generation (DG) due to its advantages over centralized power generation. Internal combustion (IC) engines, micro gas turbines (MGT) and wind turbines are the main candidates for DG technology. Biomass gasifiers with IC engines have shown success for power generation. However, one of the problems with these systems is the maintenance requirement, since producer gas has to be cleaned, cooled and dried before it can be used in IC engines.

This research developed and characterized a small scale combined heat and power (CHP) producer gas (PG) fueled micro gas turbine system. The PG was cleaned in a low-cost hot cleaning unit consisting of an insulated cyclone separator. Sensible heat of the hot PG was preserved as additional thermal power for the system and also to maintain PG tar contamination in vapor form. The PG was then fully combusted in a pressurized cyclone combustor (PCC). Combustion flue gas was then introduced into a two-stage MGT. The MGT was developed based on two vehicular turbochargers, a speed reduction pulley unit and low speed generator. Different size turbochargers were tested during MGT development phase to achieve the best performance. Exhaust flue gas was then recovered using two-pass heat recovery unit (HRU) for hot air production. The HRU was designed based on two-pass counter-flow separate annular tube heat exchanger to achieve high thermal efficiency.

The PCC was optimized for PG combustion using Fluent CFD simulation software. The system was tested experimentally in three operation modes. The first mode was with 100% PG fueled CHP system. The downdraft gasifier in this mode was pressurised up to 1.1barg. The two-stage MGT with HRU achieved 1kW$_e$ and 35kW$_{th}$ electrical and thermal powers, respectively, with overall system efficiency of 44.7 %. In the second mode, liquefied petroleum gas (LPG) was used with atmospheric PG in dual fuel operation. The two-stage MGT with HRU achieved 0.5kW$_e$ and 34kW$_{th}$ electrical and thermal powers, respectively, with overall system efficiency of 18 %. In the third mode, a single-stage MGT was used with atmospheric PG fuel for hot air production. 34kW$_{th}$ thermal power was achieved with overall efficiency of 37.5 %. Low CO and NOx emissions below 115 and 245ppm respectively were achieved for all modes of operation.

CHAPTER 1

INTRODUCTION

1.0 Background

Excessive fossil fuel utilization has led to fuel depletion, global warming and pollution. Thus, the last decade has witnessed significant increment in renewable fuels research and development for new techniques to utilize them. Renewable energy sources such as hydropower, wind, biomass, geothermal and solar are the preferable and most promising fossil fuel alternatives. The global renewable electrical generation (excluding hydropower) has tripled in the period of 2000 to 2009 as shown in Figure 1.1. Renewable energy contribution to the global electrical generation in 2009 was 21% and 3.8% with and without hydropower, respectively (REDB, 2010).

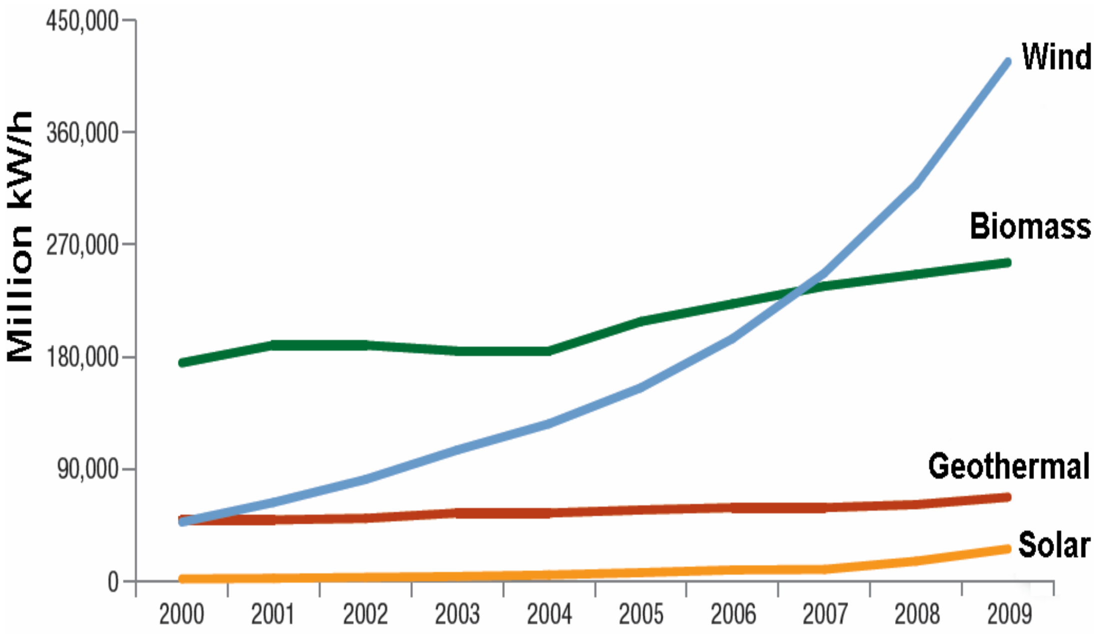

Figure 1.1: Renewable electrical generation (excluding hydropower) in the last decade (REDB, 2010)

Biomass is an important type of renewable energy fuel source in Malaysia. It provides more reliable electrical and thermal power source throughout the year with wider distribution compared to solar and wind power sources. Biomass fuel refers to any organic substance from plant materials or animal wastes used as fuels. Biomass includes for example, food crops, grassy and woody plants, agricultural or forestry residues and urban wastes.

Biomass fuel combustion does not increase the net carbon dioxide emissions in the atmosphere through the biomass growth cycle where carbon dioxide is removed through photosynthesis process (NREL, 2011). Biomass can be used for liquid or gaseous fuel production, direct power production and bioproducts. Main methods of converting biomass into a useful form of energy are summarized in Table 1.1.

Table 1.1: Biomass main conversion technologies

Technology	Conversion Process Type	Major Biomass Feedstock	Energy or Fuel Produced
Direct Combustion	Thermochemical	Wood, agricultural waste, municipal solid waste, residential fuels	Heat, steam, electricity
Gasification	Thermochemical	Wood, agricultural waste, municipal solid waste	low or medium-Btu producer gas
Pyrolysis	Thermochemical	Wood, agricultural waste, municipal solid waste	synthetic fuel, oil (biocrude), charcoal
Anaerobic Digestion	Biochemical (anaerobic)	animal manure, agricultural waste, landfills wastewater	medium Btu gas (methane)
Ethanol Production	Biochemical (aerobic)	sugar or starch crops, wood waste, pulp, sludge, grass straw	ethanol
Biodiesel Production	Chemical	Rapeseed, soy beans, waste vegetable oil, animal fats	biodiesel
Methanol Production	Thermochemical	Wood, agricultural waste, municipal solid waste	methanol

Source: Oregon, 2009.

Distributed generation (DG) was the earliest type of electrical generation to provide the power requirements for local areas. However, the attractive scale-up economical value has shaped the power generation trend and the power system development a philosophy of centralized generation (CG). In the last decade, there was a renewed interest in DG in many countries with its important role in minimizing power losses in power distribution systems (Banerjee, 2006, Sadrul Islam et al., 2006). There are large variations in the DG definitions used in literature in terms of DG size range, purpose, location, etc. One of the simple DG definitions is: electric power generation within distribution networks or on the customer side of the network

(Ackermann, 2001). DG can be used as standalone power units for the site as the main or backup power source. The other option is to connect DG to the power grid to reduce the impact of electricity price fluctuations, strengthen energy security, and provide greater stability to the electricity grid. Moreover, medium size DG can be used to meet base-load power, peaking power, backup power, remote power, power quality, as well as the CHP requirements for a particular onsite application (Oregon, 2009).

The main candidates for this technology are: internal combustion (IC) engines, micro gas turbines (MGT), fuel cells, wind turbines and photovoltaics (PV). The first three can be either used with renewable or non-renewable fuels unlike the latter two technologies that are purely renewable.

1.1 Biomass for thermal and power outputs

In Malaysia, with 3.9 million hectares of oil palm plantation and more than 360 palm oil mills, biotechnology development was emphasized in the Malaysian 9th economical plan with RM2 billion funding for biotechnology (EPU, 2009). However, besides bioproducts and biofuel production, significant amounts of oil palm industry wastes are abundant and not fully utilized. These biomass wastes can be thermo-chemically converted by gasification into combustible gas fuel known as producer gas (PG).

PG fuel can be used for thermal applications, electrical generation or combined thermal and electrical power outputs. For thermal applications, one of the most important applications in the industry sector is the drying process, such as timber drying and food processing. However, the big challenge is to get a cheap and clean heat source, knowing that the most used methods are electrical heaters or steam-based dryers. Drying is usually highly energy-intensive process and most of industrial sectors require this process to some extent. For some applications such as food processing, drying process requires special quality for the drying medium with minimal undesirable contaminations. Thus, hot filtered air is used for such process. Therefore, PG fueled hot air production unit can reduce drying process cost significantly.

For small scale power applications, small scale DG units in the range of 20-400kW$_e$ using downdraft and fluidized bed gasifiers with IC engines have shown promising success, especially for rural areas with the lack of fossil fuels supply. However, the main problem with these systems is the high maintenance requirement. Since the reciprocating engines are sensitive to the amount of tar, temperature and humidity in PG, additional cleaning, cooling and drying systems are required after the gasifiers. Further more, the engines working life becomes shorter, and in rural areas it is difficult to provide the villagers with the required technical knowledge to perform all the operation and maintenance duties correctly, so the system could fail due to poor maintenance.

Another option for small scale biomass fueled DG is to use micro gas turbine. MGT can provide a significantly lower pollution compared to IC engines with much higher thermal output making it more suitable for CHP applications. However, compressing PG after the gasifier to be injected to the MGT combustor requires intensive PG cleaning and cooling that will increase maintenance for the system.

Using a pressurized gasifier eliminates the necessity for PG cooling and enables the use of low-maintenance hot cleaning unit. Furthermore, preserving the additional thermal power of hot PG increases the system efficiency.

Using PG fueled MGT requires a pressurized PG combustor. Currently, for large scale power plants, PG is co-fired with other fossil fuels to avoid major modification on gas turbine combustors. However, for small scale MGT based DG, there is a lack of practice and studies on PG pressurized combustors. Special combustor design is required to provide high air-gas mixing quality with long residence time for PG to complete the combustion.

MGT can also use atmospheric PG combustion in the case of the externally fired micro gas turbine (EFMGT). However this method suffers from the higher capital cost of the system with lower overall efficiency as will be discussed in the next section.

1.2 Gas turbine firing methods

There are two main methods for gas turbine firing, the directly fired turbine (DFGT) and the externally fired turbine (EFGT). The direct firing of gas turbine refers to the conventional gas turbine firing where combustion products expand directly in the turbine. Whereas the externally or indirectly fired gas turbine means that the combustion chamber is not directly connected to the gas turbine. Therefore, the combustion product gases are not in direct contact with the turbine's impeller. The combustion process heats up a compressed fluid (commonly air) using high temperature heat exchanger. The hot compressed fluid then expands in the turbine producing high speed shaft power.

The indirectly and directly fired gas turbine, are both similar in concept and explained thermodynamically by the Brayton cycle. The ideal Brayton cycle temperature-entropy (T-S) diagram is shown in Figure 1.2 for the two methods. For DFGT (on the right), air is drawn by the compressor (1) and compressed (2). The pressurized combustion process (2-3) is assumed to be under a constant pressure. Hot pressurized combustion products (3) are then expanded through the turbine (4) and released to the environment.

For EFGT (on the left), combustion process (a-b) is done externally and is usually atmospheric. The working fluid is drawn by the compressor (1), compressed (2) and then passed through a heat exchanger for heating up (3). Combustion thermal power (Q) is subjected on the high temperature heat exchanger resulting in lower thermal power (q) gained by the working fluid at (3). The compressed hot fluid then expands through the turbine (4), and either discharged directly to the environment or returned back to the compressor after cooling process. As can be noticed from the figure, gas turbine inlet temperature (TIT) at (3) for DFGT is higher than TIT for EFGT, resulting in lower cycle surface area for the latter and lower efficiency.

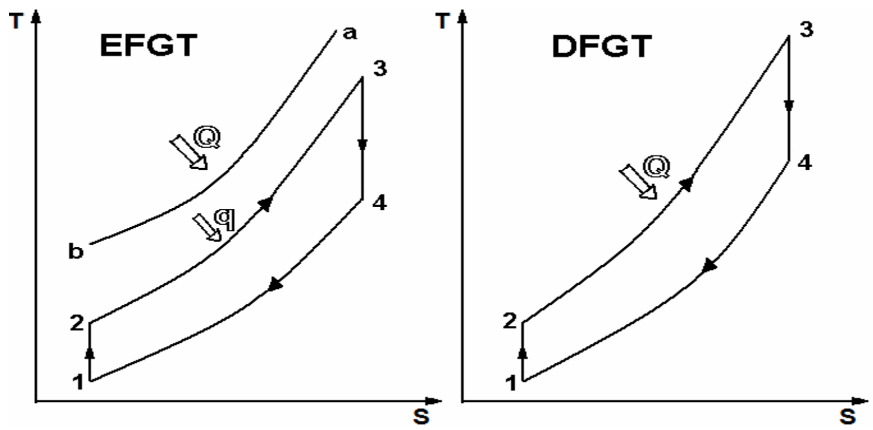

Figure 1.2: Ideal Brayton cycle T-S diagram for EFGT and DFGT

1.3 Problem statements

MGT is one of the main DG candidates and it has a large potential as CHP system especially with biomass fuel since the system can be located near the biomass sources. However, there are many difficulties in utilizing the biomass derived PG gas fuel for MGT firing. These difficulties can be summarized as following:

1. The combustion difficulties for the PG fuel.
2. The instability in MGT operation when fueled by 100% PG due to the large difference in volume/heating value between PG and high HV gas fuels.
3. The extensive MGT modifications to operate on PG.
4. The high maintenance requirment of the PG cold cleaning process.

1.4 Objectives of the study

The main objectives for this study can be summarized as following:

1. To develop and characterize a small scale pressurised cyclone combustor suitable for PG fuel combustion for MGT applications.
2. To develop and characterize a two-stage turbocharger based MGT system along with a low speed electrical generator.

3. To develop and characterize a gas-to-gas MGT heat recovery unit for hot air production.
4. To determine the performance of a biomass MGT system for power and thermal outputs using different PG fuel configurations.

1.5 Scope and limitations of the study

The scope of this research work and the equipment limitations are summarized as following:

1. Design of the PCC for PG combustion using Fluent 6 CFD program.
2. Utilizing the available $100kW_{th}$ and $150kW_{th}$ downdraft gasifiers to supply the hot PG fuel for the MGT firing.
3. Investigating the DFMGT concept using vehicular turbochargers and low-speed electrical generators.
4. Design of a suitable heat recovery unit that can be used for hot air production.
5. Characterizing the CHP-MGT system based on experimental work.
6. The downdraft gasifiers used in this study can preferably use large wood blocks as fuel for stable operation. Biomass fuel is limited to off-cut furniture wood available from local furniture industries.
7. Gasifier compression equipment were limited with maximum PG pressure of 1.1barg.
8. Flow rate and moisture content of the MGT flue gases were not available for mass balance calculations.

1.6 Overview of the study

Introducing biomass fuel in to the DFGT and cyclone combustors technologies was getting more attention lately. Some of the studies on gas turbines running on solid fuels or low HV gas fuels are presented in Chapter 2. The Chapter also presents a variety of technologies and methods on the low HV gas combustion. Chapter 3 presents the theoretical frame work of the study including the following technologies: gasification, low HV gas combustion, micro gas turbine, low and high speed electrical generation and MGT heat recovery. In Chapter 4, theories and methods those were implemented during the research are discussed elaborately, including: PCC design and simulation, MGT system design and development, electrical generation system development, heat recovery unit design and also the experimental and measurement rig during the different stages of the study.

In Chapter 5, the findings during the MGT development phase are discussed followed by the performance of the different parts of the system. Different system configuration with single and double stages MGT, and different operation modes with single and dual fuel are compared. The final part of the Chapter includes the system performance comparison for the three main operation modes with pressurized PG CHP system, dual fuel CHP system and atmospheric PG hot air production system.

The performance of the different system parts and operation modes are concluded in Chapter 6. This Chapter also includes different recommendations for further development of the system.

CHAPTER 2

LITERATURE REVIEW

2.0 Introduction

In this Chapter, some of the studies on gas turbines using biomass fuel are presented under two main categories: Large/medium scale and small scale systems. Low HV fuel combustion technologies review with the different combustors designs are presented, followed by the type of gasifiers currently used for gas turbine applications. Humidified Gas turbine technology review is presented after that. Finally, literature summary and the study contributions are presented.

2.1 Large and medium scales biomass fueled gas turbine systems

The utilization of biomass fuel for medium and large scale (above $1MW_e$) gas turbine power systems has been widely studied. Biomass fuel can be used as a single fired fuel or co-fired with other higher heating value fuels to run gas turbine engines. The first issue to be taken into consideration is the choice of a suitable biomass combustion method since biomass can be combusted directly as solid fuel, or converted into liquid or gas fuel and then combusted. Secondly, the turbine firing method can be direct firing, indirect firing or a combination between the two methods. Lastly, the overall system efficiency can be increased by using different system configurations involving other technologies such as the co-generation with steam turbines or IC engines, etc.

One of the main concerns for the large scale gas turbine power plants is the low HV fuel combustion. Hence, such fuels have relatively higher gas flow associated with lower burn velocity and heat generation compared to higher HV gas fuels, additional to the high quality air/fuel mixing requirement. All that require replacement or major design modifications for the combustor and size modification for the turbine. Therefore, co-firing technology has been presented as economical solution for this issue. Since the high and low HV fuels can be both used in the existing power plants with a co-firing ratio that requires minor modifications on the combustors.

2.1.1 Co-firing biomass with other fuels for gas turbine systems

A study on coal/biomass co-firing was investigated by Huang et al. (2006). Pressurized fluidised bed combustion (PFBC) system was used in this study. The system was based on a commercially available P800 module developed by ABB Carbon as shown in Figure 2.1.

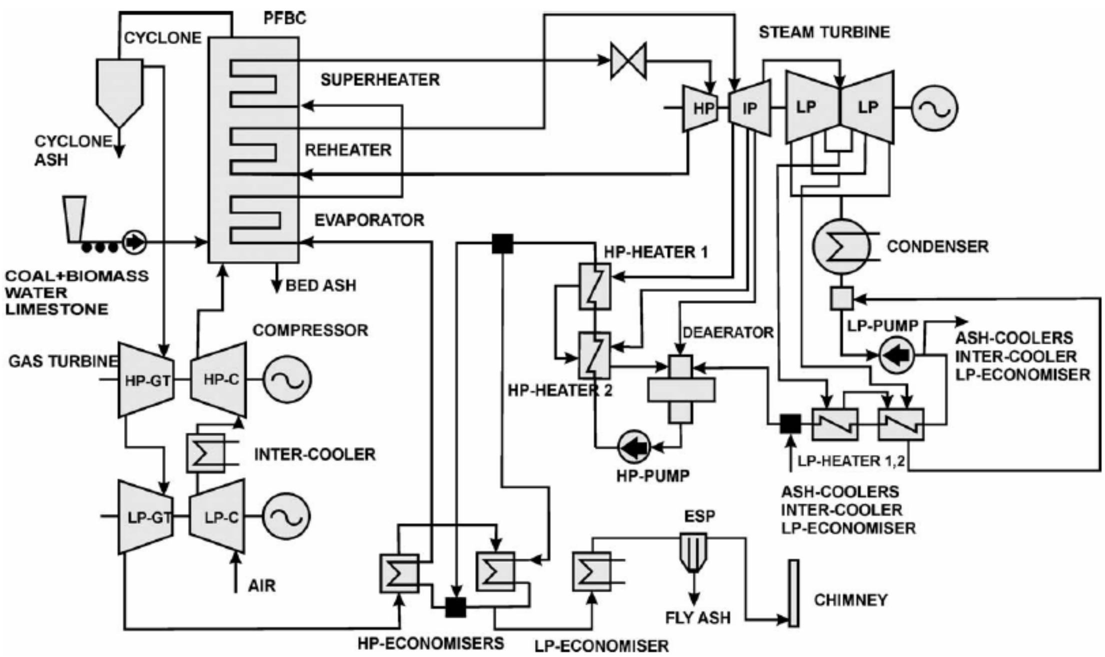

Figure 2.1: Schematic drawing of the PFBC combined cycle power plant

(Huang et al., 2006)

In this study, computational simulation was carried out for various fuel feedstock mixtures of up to 40% biomass maximum to avoid major modifications in this coal fired system. The bed temperature inside the combustor was low of about 855°C to prevent melting of the ash and to reduce NOx emissions. In this system, only one converting step was used to convert the solid fuels into combustion products that can be expanded directly in the gas turbine. This can be acceptable in the fluidized bed systems due to the long combustion residence time. Hot flue gases out of the PFBC were passed through parallel sets of two-stage cyclones before expanding in a two-stage gas turbine that is coupled with a two-stage compressor with intercooler. The compressor provides about 16bar pressurized air at 300°C for the combustor. The combustor also provided thermal power for electrical generation using steam turbine power plant. The overall electric power output of the PFBC combined cycle was expected to be about 360MW_e. The selected types of biomass and biomass were: straw, willow chips, switch grass, miscanthus and olive pits. The moisture contents varied from 7.17% to 33.51%. The results showed that the steam cycle output reacts more sensitive to the fuel configurations comparing with the gas turbine cycle. Also, the increased fraction of biomass reduces net CO_2 and SOx significantly. However, NOx emissions tended to rise for all biomass types, except the high moisture content willow chips. Although the increment of biomass co-firing ratio has caused a reduction in steam cycle thermal power, flue gas flow has increased, resulting in a larger fraction of gas turbine output. For example, willow chips co-firing ratio of 40% has increased the gas turbine output by 17.93MW_e and decreased the steam turbine output by 37.51MW_e compared to 100% coal. Thus, although the turbine inlet temperature decreases with biomass, higher flue gas flow through the turbine provides more output power. This is encouraging for the future development of biomass fueled gas turbine systems.

In a similar study, coal/biomass co-gasification has been investigated in an integrated gasifier combined cycle (IGCC) system (Jong et al., 1999). The study was under the multinational EU JOULE project and it included a 1.5MWth air/steam pressurized bubbling fluidised bed gasifier (PFBG) at Delft University (Figure 2.2). The gasifier was planned to be used in axial gas turbine with modified combustor and steam turbine combined cycle. PG exits the gasifier at 10bar and 900ºC maximum pressure and temperature, respectively. PG was cleaned in a hot gas cleaning system consisting of online-cleaned ceramic candle filters. The paper described the performance of the gasifier with coal/miscanthus, coal/straw blends and brown coal/miscanthus eventual study at different mixing ratios with limestone as an additive. A modified pressurised ALSTOM Typhoon gas turbine combustor was used for PG combustion. Parallel kinetics-based model simulation of the system using ASPEN PLUS was also performed. However, the system was not tested with gas and steam turbines.

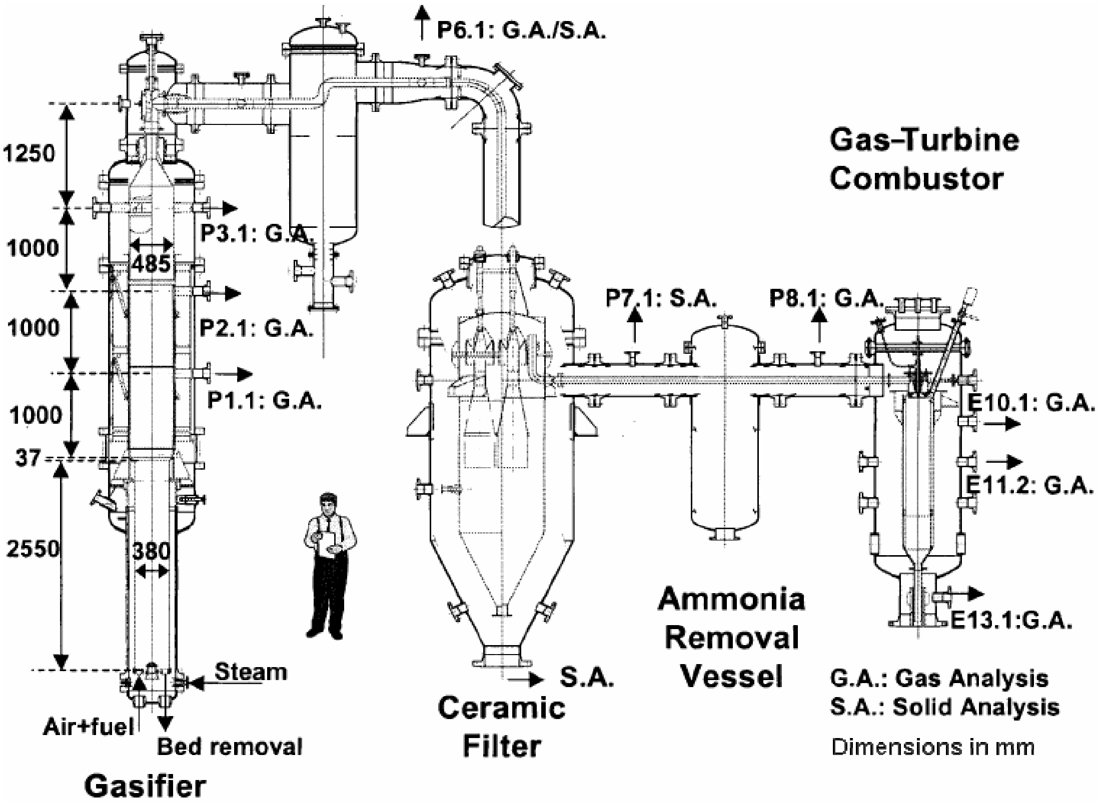

Figure 2.2: Schematic drawing of the 1.5MW$_{th}$ Delft PFBG test rig

(Jong et al., 2003)

The addition of hot gas filtration using ceramic channel filters with smaller pressurized fluidised bed gasifier (PFBG) has also been investigated (Jong et al., 2003). The 50kW$_{th}$ PFBG test rig was tested at Stuttgart University (DWSA) as shown in Figure 2.3. The PFBG reactor was electrically heated to maintain constant temperature over the bed. PG was cleaned through hot gas cleaning system consisting of a cyclone separator and ceramic SiC candle filter at 500ºC. The combustor was specially designed for PG combustion. The combustor design was based on ceramic chamber with annular swirl-diffusion chamber with primary and secondary swirl air inlets. And the combustor was contained in water-cooled pressurized vessel.

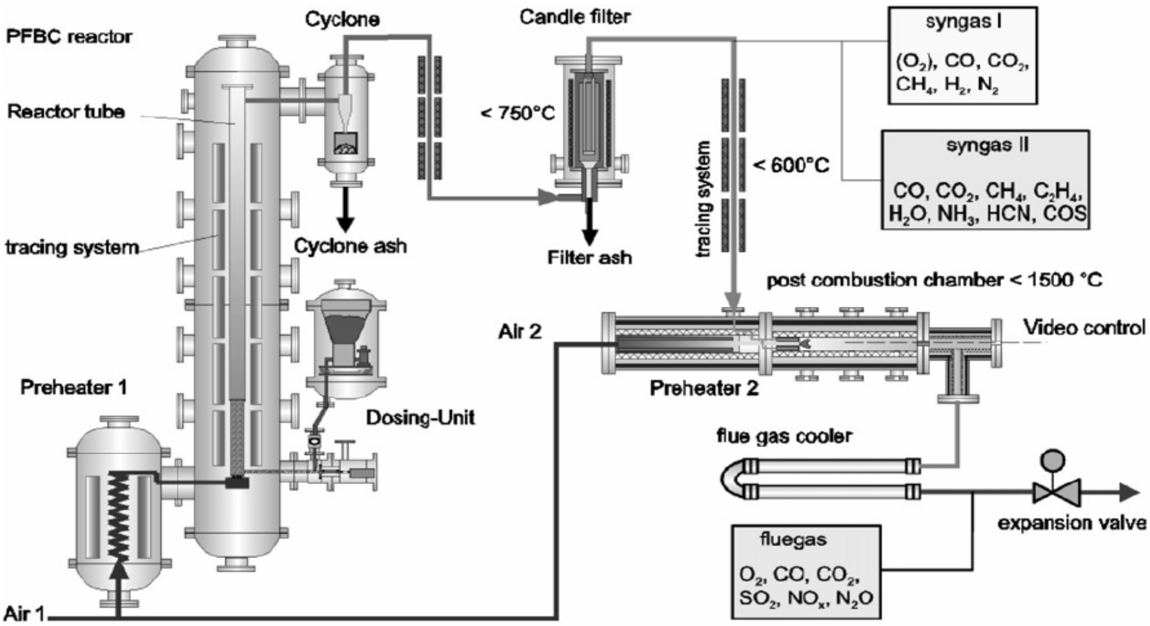

Figure 2.3: Schematic drawing of the $50kW_{th}$ PFBG test rig (Jong et al., 2003)

Utilizing biomass fuel in its solid state requires pre-treatment for the fuel to be reduced in size to be suitable for cyclonic or fluidized bed single-stage combustors. However, this type of combustion is not preferable for direct gas turbine firing due to the high particulate matter content in the combustion products that require intensive cleaning before it can be used in turbine engines. A study on coal/biomass co-firing technology was investigated (Tillman, 2000). The study reviewed three different techniques for the co-firing:

- Blending the biomass and coal in the fuel handling system and feeding that blend to the boiler.
- Preparing the biomass fuel separately from coal, and injecting it into the boiler without impacting the conventional coal delivery system.
- Gasifying the biomass with subsequent combustion of the producer gas in either a boiler or a combined cycle combustion turbine (CCCT) generating plant.

For first and second techniques, biomass fuel was used in the solid state and combusted in a boiler for power generation using steam turbine system. However, for gas turbine systems, it is preferable to convert biomass fuel into gaseous or liquid form before the direct firing into gas turbine as in the third technique. For coal fueled power stations, biomass co-firing can cause reduction in system efficiency. However, the environmental benefits by reducing NOx, SO_2, CO_2 and metal traces such as mercury emissions makes this technology favorable, especially with the use of biomass as a renewable source of energy that adds more credibility for such stations.

Natural gas-PG Co-firing in biomass integrated gasification/ combined cycle (BIG-GT) systems has also been investigated (Rodrigues et al., 2003-A). Economic analyses were also performed for same system (Rodrigues et al., 2003-B). PG used in the simulation was based on sugar-cane residues gasification with 6MJ/m^3 LHV. The study included economic and efficiency analysis with different co-firing ratios. The use of PG to run the gas turbine at the rated power results in a very high flow rate through the combustor and expander. Therefore, a major modification or a total replacement was required for the combustor to provide enough residence time for the complete combustion of the high flow producer gas. Moreover, some modifications were required for the expander as well to cope with the higher pressure and flow rates and to avoid turbine over speeding. The addition of natural gas to the fuel mixture increased the heating value of the gas for stable gas turbine operation and also to avoid the gas turbine power de-rating and the high drop in system efficiency. High natural gas ratio above 50% allowed a normal operation without modifications on the BIG-GT plant.

Exergy loss based economic analysis for the natural gas/biomass co-fired combined cycle power plants has been studied (Franco and Giannini, 2005). Two plant configurations, biomass integrated post combustion combined cycle (BIPCC) and biomass integrated fired recuperated combined cycle (BIFRCC) have been analyzed. For both proposed cycles, unlike the previous studies, PG was not mixed with natural gas to run the gas turbine in order to avoid any modifications on the gas turbine system. However, only thermal power was utilized from producer gas by burning the gas in an atmospheric burner. For BIPCC, commercial gas turbine GE LM6000PD was used. Biomass thermal power was used to increase thermal power of the turbine flue gas to increase the steam cycle power. Maximum efficiency of this cycle was found to be around 60% with about 23% biomass thermal input. For BIFRCC, another commercial gas turbine GE MS6001FD was found to be more suitable for this cycle with higher discharge temperature after turbine. Biomass thermal power was used to preheat air for the gas turbine as a recuperator. Maximum efficiency of this cycle was found to be around 57% with about 20% biomass thermal input. A schematic drawing of both cycles is shown in Figure 2.4.

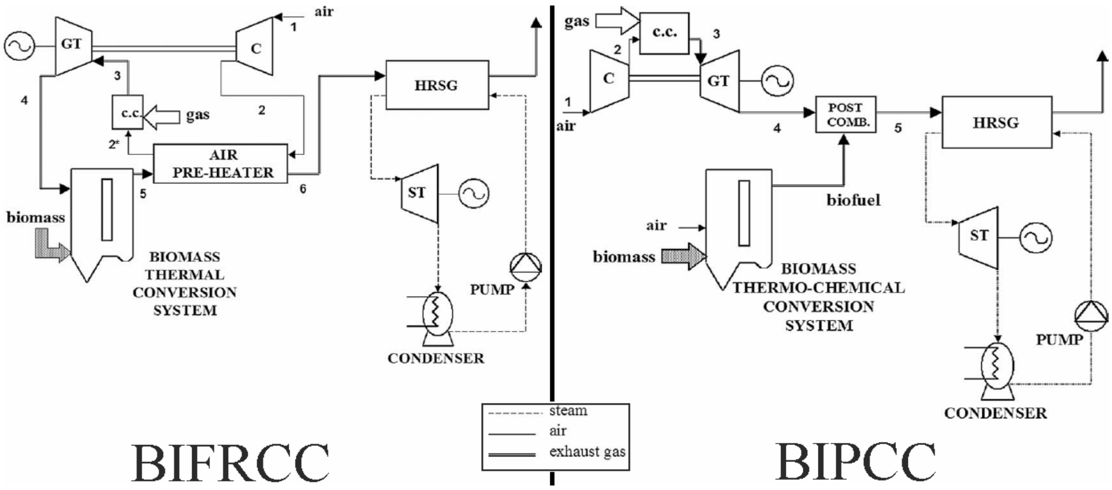

Figure 2.4: Schematic drawings for the BIFRCC & BIPCC cycles

(Franco and Giannini, 2005)

A simulation study was carried out on an existing GE5 gas turbine power plant to evaluate the natural gas/ biomass co-firing option from the economical point of view (Fiaschi and Carta, 2007). Increasing PG amount has caused a reduction in gas turbine efficiency especially for the compressor side and required a modification on the turbine engine geometry. However, 30% producer gas co-firing ratio was found to be suitable from the economical point of view to avoid turbine modifications but with output power drop of about 8-10%. Recycling the PG cleaning water as injected steam was also studied to enhance gasifier performance and reduce water treatment cost.

The study also aimed at CO_2 emission reduction from 10% to 50% in the existing IGCC gas turbine based power plants with simple and low cost modifications. The idea was to return some of the gas turbine hot flue gases back to the gasifier as gasification agent since it contained some amounts of oxygen, with some additional steam. Part of the flue gases thermal power was used in this case to reduce the biomass fuel consumption. On the other hand, CO_2 amounts in the flue gases can enhance carbon conversion into CO as in the following reaction: $C + CO_2 = 2\ CO$.

Feasibility analyses have been done on producer gas and natural gas co-fired in biomass gasification integrated to combined cycle (BIG-CC) (Walter and Liagostera, 2007). Simulation was based on $145MW_e$ gas turbine with sugarcane residues as biomass and $5.16MJ/m^3$ LHV PG. The study showed a promising economical potential for the 100% biomass fueled combined cycle BIG-CC. However, high economical risk due to the lack of experience in such units urges for economically safer solutions such as co-firing to achieve learning factor for the short term.

Sondreal et al. (2001) have reviewed the biomass co-firing with variety of higher HV fuels and the different gas and steam turbines technologies. Three main systems were compared: supercritical steam boiler with advanced emission controls, EFGT combined cycle and hybrid gasifier pressurized fluidized bed combustor (PFBC) system. First two systems are well known; however, third one combines different technologies as shown in Figure 2.5 for the basic coal-fired system. In this system, PG was combusted in a topping combustor along with the hot flue gases from the PFBC to rise the temperature up to 1260ºC for gas turbine firing in a combined cycle.

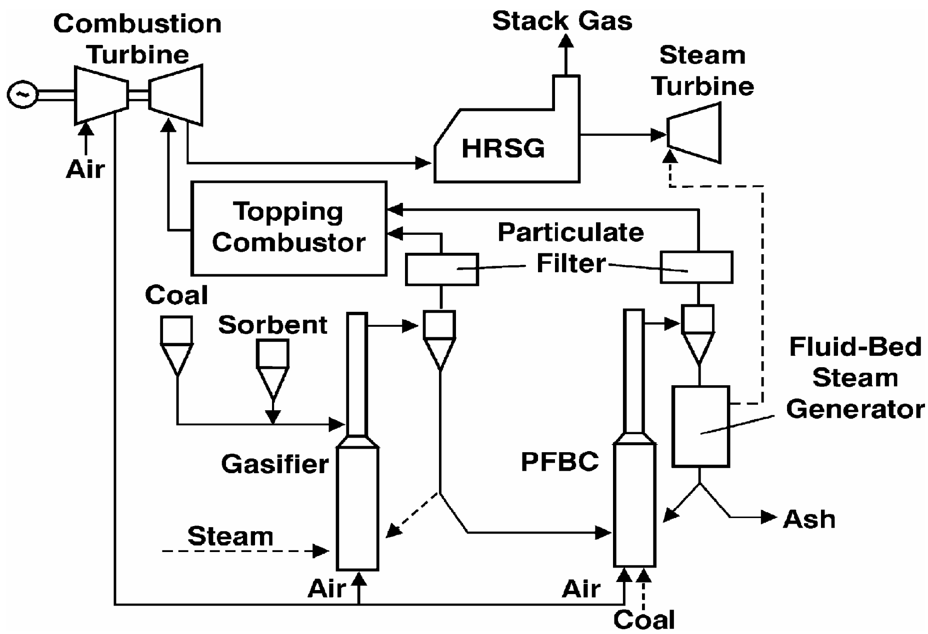

Figure 2.5: Hybrid gasifier pressurized fluidized bed combustor (PFBC) system (Sondreal et al., 2001)

From the economical point of view, Hughes (2000) has studied the potential and the required policies for biomass co-firing in the existing power plants in the USA.

2.1.2 Biomass fueled combined cycle systems

A 100% biomass fueled systems without co-firing with higher HV fuels have been widely studied. Some of the studies on the biomass integrated gasifier combined cycle (BIGCC) are presented below.

Rodrigues et al. (2007) have studied the utilization of Sugar cane residue using atmospheric circulating fluidized bed gasifier in BIGCC system. Three commercial gas turbines were simulated: small capacity LM2500 (22MW$_e$), medium GE PG6101 (70MW$_e$) and large GE PG7001 (159MW$_e$). The main issue to be considered when operating gas turbine on PG is the expander-compressor matching.

This is because gas turbine is designed to operate on similar flow rate on the expander and compressor, since high HV fuel flow is very low unlike the PG operation. Different techniques were compared:

- Increasing compression ratio.
- De-rating. This technique was imposed as compressor surge control by reducing turbine inlet temperature.
- Compressor air bleeding. High flow PG operation resulted in a compressor over pressure. Bleeding was imposed to keep the compressor pressure at maximum rated value as with natural gas operation.
- Retrofitting the expander. Modifications on expander geometry mainly on the gas nozzle critical area at turbine inlet and nozzle angle were made, while maintaining the compressor nominal operation. Retrofitting the expander provided a cycle efficiency enhancement similar to the compression ratio increment technique as shown in Figure 2.6.

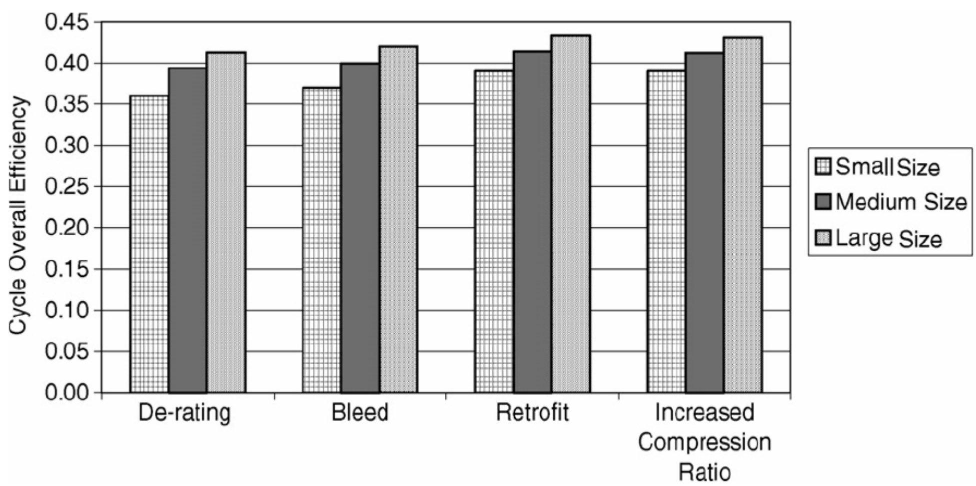

Figure 2.6: BIGCC efficiencies for different turbine sized with different operation techniques (Rodrigues et al., 2007)

A 205MW$_e$ BIGCC plant (see Figure 2.7) with CO_2 chemical absorption was simulated using Aspen Plus program (Corti and Lombardi, 2004). Dry poplar (biomass fuel) was converted using atmospheric gasification into PG with 6.2MJ/kg LHV. However, after CO_2 absorption with aqueous amines solutions, PG heating value increased up to 11.3MJ/kg. The calculated efficiency was in the range of 35% and 36% comparing with efficiency values previously found for IGCC fed with coal with upstream CO2 chemical absorption (38–39%).

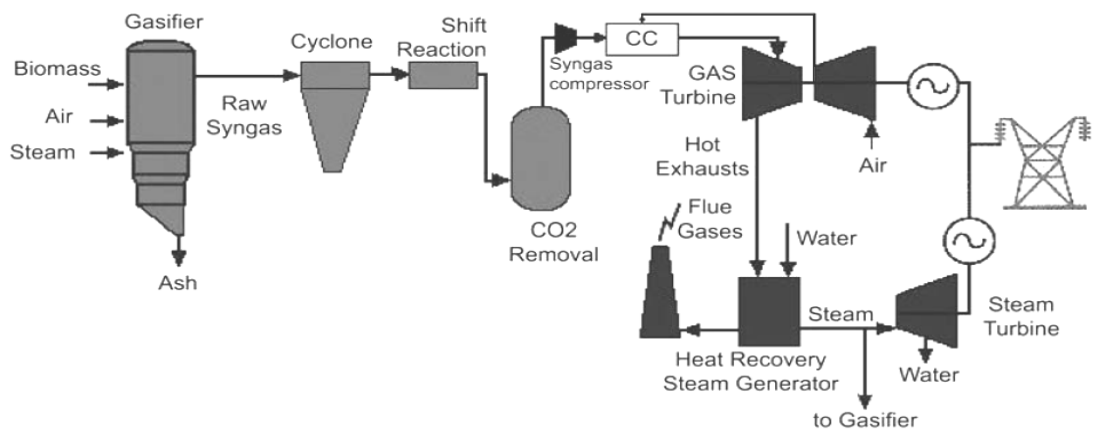

Figure 2.7: Schematic drawing of the BIGCC with chemical Co2 removal

(Corti and Lombardi, 2004)

Another 20MW BIGCC system fueled by the olive tree by-product biomass waste was simulated using MATLAB software (Jurado et al., 2003). Simulations were performed on. Biomass was converted into PG of 5MJ/kg HHV using pressurized fluidized bed gasifier at 10bar pressure with hot gas cleaning to preserve PG thermal power. The study concentrated mainly on modeling a real time control for turbine temperature and speed.

Brown et al. (2007) have used a parametric stoichiometric equilibrium model to compare between two biomass fueled cycles: internal combustion engine combined cycle (ICE-CC) and gas turbine combined cycle (GT-CC). First cycle used a fluidised bed gasifier (FBG) with cold PG cooling and cleaning unit. Second cycle used pressurized FBG with hot PG cleaning unit. Three gasification agents were compared: air, oxygen and steam. Steam was found to be the optimum gasification agent for both cycles from the economical point of view. GT-CC has achieved lower electrical specific cost compared to the ICE-CC.

Economic and emissions evaluations have been carried out for different combined cycle configurations (Lazaro et al., 2006). The study compared between three power generation options: biomass fueled gas turbine cogeneration plant (combined cycle), industrial steam boiler fueled by natural gas and biomass fueled steam boiler for power generation. The comparison considering electrical efficiency, economical value and CO_2 emission allowance prices based on European Directive 2003/87/CE as the main factors. Combined cycle provides higher efficiency and electrical intensity; however, biomass boiler becomes beneficial for higher CO_2 emission allowance prices due to its low emissions.

The performance of a direct fired gas turbine combined cycle was compared to the externally fired gas turbine cycle (Ferreira et al., 2003). In this study, the following gas turbine cycles were compared:

i. Biomass integrated gasification/ gas turbine cycle (BIGGT), with direct turbine firing.
ii. Externally fired gas turbine (EFGT), with ceramic heat exchanger.