PERFORMANCE OF A BIOMASS FUELLED MICRO GAS TURBINE POWER UNIT

ASSOC.PROF.DR.ZAINAL ALIMUDDIN ZAINAL ALAUDDIN

UNIVERSITI SAINS MALAYSIA
KAMPUS KEJURUTERAAN
2008

Laporan Akhir Projek Penyelidikan Jangka Pendek

Performance of a Biomass Fuelled Micro Gas Turbine Power Unit

by

Assoc. Prof. Dr. Zainal Alimuddin Zainal Alauddin
Muhammad Azman Miskam
Mohd Yusof Idroas
Khalid Mohamad Ali-Attab

UNIVERSITI SAINS MALAYSIA

LAPORAN AKHIR PROJEK PENYELIDIKAN JANGKA *PENDEK*

FINAL REPORT OF SHORT TERM RESEARCH PROJECT

Sila kemukakan laporan akhir ini melalui Jawatankuasa Penyelidikan di Pusat Pengajian dan Dekan/Pengarah/Ketua Jabatan kepada Pejabat Pelantar Penyelidikan

1. **Nama Ketua Penyelidik:** ZAINAL ALIMUDDIN BIN ZAINAL ALAUDDIN
   Name of Research Leader

   ☑ Profesor Madya/ *Assoc. Prof.* ☑ Dr./ *Dr.* ☐ Encik/Puan/Cik *Mr/Mrs/Ms*

2. **Pusat Tanggungjawab (PTJ):**
   School/Department

   PUSAT PENGAJIAN KEJURUTERAAN MEKANIK

3. **Nama Penyelidik Bersama:** **Muhammad Azman Miskam, Mohd Yusof Idroas dan Khalid Mohamad Ali-Attab**

4. **Tajuk Projek:** **PERFORFORMANCE OF A BIOMASS FUELLED MICRO GAS TURBINE POWER UNIT.**
   Title of Project

5. **Ringkasan Penilaian/*Summary of Assessment*:**

	Tidak Mencukupi *Inadequate* 1	2	Boleh Diterima *Acceptable* 3	Sangat Baik *Very Good* 4	5
i) **Pencapaian objektif projek:** *Achievement of project objectives*	☐	☐	☑	☐	☐
ii) **Kualiti output:** *Quality of outputs*	☐	☐	☑	☐	☐
iii) **Kualiti impak:** *Quality of impacts*	☐	☐	☑	☐	☐
iv) **Pemindahan teknologi/potensi pengkomersialan:** *Technology transfer/commercialization potential*	☐	☐	☑	☐	☐
v) **Kualiti dan usahasama :** *Quality and intensity of collaboration*	☐	☐	☑	☐	☐
vi) **Penilaian kepentingan secara keseluruhan:** *Overall assessment of benefits*	☐	☐	☑	☐	☐

6. **Abstrak Penyelidikan**
(Perlu disediakan di antara 100 - 200 perkataan di dalam **Bahasa Malaysia dan juga Bahasa Inggeris.** Abstrak ini akan dimuatkan dalam Laporan Tahunan Bahagian Penyelidikan & Inovasi sebagai satu cara untuk menyampaikan dapatan projek tuan/puan kepada pihak Universiti & masyarakat luar).

Abstract of Research
(An abstract of between 100 and 200 words must be prepared in Bahasa Malaysia and in English). This abstract will be included in the Annual Report of the Research and Innovation Section at a later date as a means of presenting the project findings of the researcher/s to the University and the community at large)

PLEASE REFER TO APPENDIX 1

7. **Sila sediakan laporan teknikal lengkap yang menerangkan keseluruhan projek ini. [Sila gunakan kertas berasingan]**
Applicant are required to prepare a Comprehensive Technical Report explaning the project. (This report must be appended separately)

PLEASE REFER TO APPENDIX A

Senaraikan kata kunci yang mencerminkan penyelidikan anda:
List the key words that reflects your research:

Bahasa Malaysia	*Bahasa Inggeris*
Gas turbin mikro	Micro Gas turbıne
Penggasan Biojisim	Biomass gasification
Teknik Pengeringan	Drying Technique

8. **Output dan Faedah Projek**
Output and Benefits of Project

(a) * **Penerbitan Jurnal** **REFER TO APPENDIX B**
Publication of Journals
(Sila nyatakan jenis, tajuk, pengarang/editor, tahun terbitan dan di mana telah diterbit/diserahkan)
(State type, title, author/editor, publication year and where it has been published/submitted)

(b) **Faedah-faedah lain seperti perkembangan produk, pengkomersialan produk/pendaftaran paten atau impak kepada dasar dan masyarakat.**
State other benefits such as product development, product commercialisation/patent registration or impact on source and society.

The project is a product development and upon further extensive test can be commercialized with high potential application in industries requiring drying.

* Sila berikan salinan/*Kindly provide copies*

(c) **Latihan Sumber Manusia**
Training in Human Resources

i) Pelajar Sarjana:
Graduates Students
(Perincikan nama, ijazah dan status)
(Provide names, degrees and status)

KHALID MOHAMMAD ALI AL-ATTAB, OBTAINED MASTER IN SCIENCE (MECHANICAL ENGINEERING)

ii) Lain-lain:
Others

____

____

____

9. Peralatan yang Telah Dibeli:
Equipment that has been purchased

TENMAR BLOWER 7.5 KW

5/3/08

Tandatangan Penyelidik
Signature of Researcher

Tarikh
Date

Komen Jawatankuasa Penyelidikan Pusat Pengajian/Pusat
Comments by the Research Committees of Schools/Centres

Penyelidekan dijalankan didalam bidang Tenaga Diperbaharui dan memajukan sistem sedia ada dgn. 'externally fired gas turbine'. Kajian pulu juga membangunkan model matematik yg. boleh meng-optimumkan proses tersebut.
Perlulah disebut bil. of lah pelajar MSc/PhD yg. diselia berserta terbitan jurnal yg berkaitan.

5/3

TANDATANGAN PENGERUSI
JAWATANKUASA PENYELIDIKAN
PUSAT PENGAJIAN/PUSAT
Signature of Chairman
[Research Committee of School/Centre]

Tarikh
Date

ABSTRAK PENYELIDIKAN

Alternative fuels have been considered as a priority in the research field, especially for the renewable energy, due to the issue of fossil fuel depletion. Biomass is an important type of renewable energy fuel sources for thermal or electrical power production, especially in Malaysia since it has a large potential of the biomass fuel sources. As for the thermal power, one of the most important applications in the industry sector is the drying process. 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. As for the electrical power generation, downdraft or fluidized bed gasifiers with diesel or gas reciprocating engines have shown a promising success but with high cost additional cleaning systems. However, by the direct combustion of the producer gas, no additional systems are required. Moreover, the combustion thermal power can be used for power generation using the externally fired gas turbine system (EFGT) or as a thermal power for any industrial drying process.

This research is to develop and characterize a small scale biomass hot air production system using biomass fuels, by employing a turbocharger-based EFGT method, and by using air blowers instead of the turbocharger engine. A gasifier-combustor has been used to convert off cut furniture wood into thermal power. Thereafter, the heat is transferred to a clean air as the system product, through a stainless steel heat exchanger.

An EFGT system with turbocharger was designed, fabricated and operated. However, the operation of the EFGT was not achieved due to the startup failure. The maximum air pressure inside the heat exchanger was about 0.65bar and the maximum turbine inlet temperature was 693°C. The heat exchanger effectiveness was about 60.8%. The hot air production system was then characterized without the turbocharger engine. The output hot air power, temperature and flow rate were in the range of 11.8 - 13 kWt, 400 - 560 °C and 0.02 -0.03 kg/s respectively, with low CO emission of 0-164.22 mg/Nm^3. The maximum NOx emission was about 220.34 mg/Nm^3. The heat exchanger effectiveness was 70% and the overall efficiency of the system was 18%.

Bahan api merupakan subjek utama dalam bidang pengajian, terutamanya untuk tenaga boleh diperbaharui, disebabkan oleh kekurangan bahan api fosil. Biojisim merupakan salah satu tenaga boleh diperbaharui yang penting untuk penjanaan kuasa, terutamanya di Malaysia di mana banyak biojisim boleh didapati dan ia berpotensi yang besar untuk digunakan sebagai bahan api. Pengeringan merupakan salah satu proses yang penting di dalam kebanyakan sektor industri yang Menghadapi satu masalah untuk mendapatkan bekalan tenaga haba yang bersih dan murah. Cara yang biasa digunakan ialah pengering eletrikal dan stim. Untuk penjanaan kuasa elektrik, penggas alir bebas bawah atau lapisan terbendalir dengan enjin diesel atau enjin gas salingan telah menunjukkan keputusan yang menggalakkan, tetapi kos untuk sistem pembersihan adalah tinggi. Dengan pembakaran terus biojisim, tiada sistem tambahan diperlukan. Tenaga haba dari pembakaran juga boleh digunakan untuk penjanaan kuasa dengan menggunakan turbin gas pembakaran luar atau sebagai tenaga haba untuk proses pengeringan industri.

Pengkajian ini merupakan pembangunan dan pencirian sistem penjanaan udara panas skala kecil dengan biojisim, menggunakan turbin gas pembakaran luar berdasarkan sebuah "turbocharger" dan penghembus udara. Sebuah pembakar penggas telah digunakan untuk menukar sisa kayu perabot kepada tenaga haba. Tenaga haba ini kemudian dipindah kepada udara bersih sebagai produk sistem melalui penukar haba keluli.

Satu sistem turbin gas pembakaran luar telah direkabentuk, difabrikasi dan digunakan. Operasi turbin gas pembakaran luar tidak berjaya disebabkan oleh kegagalan semasa proses permulaan. Tekanan udara maksimum di dalam penukar haba lebih kurang 0.65bar dan suhu salur masuk turbin maksimum adalah 693°C. Kecekapan penukar haba adalah lebih kurang 60.8%. Kemudian sistem pengeluaran udara panas telah dicirikan tanpa enjin turbocharger. Tenaga pengeluaran udara panas, suhu dan kadar aliran masing-masing berada dalam nilai 11.8 -13kWt, 400-500°C dan 0.02-0.03kg/s. Pengeluaran CO adalah kurang dengan nilai 0-164.22 mg/Nm^3. Pengeluaran NOx maksimum adalah lebih kurang 220.34 mg/Nm^3. Kecekapan penukar haba adalah 70% dan kecekapan keseluruhan sistem adalah 18%.

APPENDIX B

1. Alattab K. A., Zainal Z. A., 2006. Externally Fired Gas Turbine for Small Scale Power Unit / Hot Air Production Unit for Drying Processes Using Biomass Fuel. International Conference on Energy and Environment, August 28-29, 2006. University Tenaga Nasional, Kuala Lumpur, Malaysia.

2. Alattab K. A., Zainal Z. A., 2007. Performance of High Temperature Heat Exchanger in Externally Fired Gas Turbine System Using Biomass Gasifier-Combustor with Full Air Recuperation. Submitted on 1st Oct 2007, for review in the Energy International Journal, Elsevier Science Limited.

3. Alattab K. A., Zainal Z. A., 2007. Design of high temperature heat exchanger for the indirectly firing of a micro gas turbine using biomass fuels. Conference on Design, Simulation, Product Development and Optimization, December 10-11, 2007. Universiti Sains Malaysia, Park Royal Hotels, Batu ferringhi, Peang, Malaysia.

UNIT KUMPULAN WANG AMANAH
UNIVERSITI SAINS MALAYSIA
KAMPUS KEJURUTERAAN
SERI AMPANGAN

PENYATA KUMPULAN WANG

TEMPOH BERAKHIR 31 OKTOBER 2007

PROF MADYA ZAINAL ALIMUDDIN

304.PMEKANIK.6035211

JUMLAH GERAN :- 18,768.00

NO PROJEK :-

PANEL :- J/PENDEK

PENAJA :- JANGKA PENDEK

Tempoh Projek:31/08/2006 - 01/08/2008

PERFORMANCE OF A BIOMASS FUELED MICRO GAS TURBINE POWER UNIT

Vot	Peruntukan (a)	Perbelanjaan sehingga 31/12/2006 (b)	Tanggungan semasa 2007 (c)	Perbelanjaan Semasa 2007 (d)	Jumlah Perbelanjaan 2007 (c + d)	Jumlah Perbelanjaan Terkumpul (b+c+d)	Baki Peruntukan Semasa 2007 (a-(b+c+d)
11000: GAJI KAKITANGAN AWAM	3,000.00	0.00	0.00	3,730.95	3,730.95	3,730.95	(730.95)
21000: PERBELANJAAN PERJALANAN DAN SARAHI	4,000.00	0.00	0.00	1,634.00	1,634.00	1,634.00	2,366.00
23000: PERHUBUNGAN DAN UTILITI	384.00	0.00	0.00	18.30	18.30	18.30	365.70
26000: BAHAN MENTAH & BAHAN UNTUK PENYELE	0.00	0.00	0.00	392.40	392.40	392.40	(392.40)
27000: BEKALAN DAN ALAT PAKAI HABIS	8,000.00	2,655.90	360.00	8,679.10	9,039.10	11,695.00	(3,695.00)
28000: PENYELENGGARAAN & PEMBAIKAN KECIL	0.00	0.00	0.00	0.00	0.00	0.00	0.00
29000: PERKHIDMATAN IKTISAS & HOSPITALITI	3,384.00	165.00	0.00	1,918.00	1,918.00	2,083.00	1,301.00
	18,768.00	2,820.90	360.00	16,372.75	16,732.75	19,553.65	(785.65)
Jumlah Besar	18,768.00	2,820.90	360.00	16,372.75	16,732.75	19,553.65	(785.65)

UNIVERSITI SAINS MALAYSIA

PEJABAT PENGURUSAN & KREATIVITI PENYELIDIKAN
RESEARCH CREATIVITY AND MANAGEMENT OFFICE [RCMO]

LAPORAN AKHIR PROJEK PENYELIDIKAN JANGKA *PENDEK*
FINAL REPORT OF SHORT TERM RESEARCH PROJECTS

6035211 – no. akaun

1) **Nama Ketua Penyelidik** :
Name of Research Leader :

Ketua Penyelidik *Research Leader*	PTJ *School/Centre*
ZAINAL ALIMUDDIN BIN ZAINAL ALAUDDIN	MECHANICAL ENGINEERING

Nama Penyelidik Bersama
(Jika berkaitan) :
Name/s of Co-Researcher/s
(if applicable)

Penyelidik Bersama *Co-Researcher*	PTJ *School/Centre*
MUHAMAD AZAMAN BIN MISKAM	MECHANICAL ENGINEERING
MOHD YUSOF IDROAS	MECHANICAL ENGINEERING

2) **Tajuk Projek** :
Title of Project:

PERFORMANCE OF A BIOMASS FUELED MICRO GAS TURBINE POWER UNIT

3) **Abstrak untuk penyelidikan anda**

(Perlu disediakan di antara 100 – 200 perkataan di dalam **Bahasa Malaysia dan Bahasa Inggeris.** Ini kemudiannya akan dimuatkan ke dalam Laporan Tahunan Bahagian Penyelidikan & Inovasi sebagai satu cara untuk menyampaikan dapatan projek tuan/puan kepada pihak Universiti & luar).

Abstract of Research

(Must be prepared in 100 – 200 words in Bahasa Malaysia as well as in English. This abstract will later be included in the Annual Report of the Research and Innovation Section as a means of presenting the project findings of the researcher/s to the university and the outside community)

Some agricultural products and timber require drying prior to application in specific industries. Drying is primarily achieved by supplying hot air or superheated steam. The production for steam and hot air is conventionally done by electricity. The consumption of electricity is significantly high. Otherwise drying can also be done by solar drying but the process is dependent on the climate. Steam is also normally produced in steam plants using diesel or biomass as fuel in direct combustion systems. In this project a biomass gasifier/combustor system and a high temperature heat exchanger were developed. A micro gas turbine unit was installed to produce hot air via the heat exchanger. The system employs an indirect fired (externally fired) concept. The shaft power from the turbine runs the compressor and induced air into the heat exchanger. Subsequently the air is heated by the combustor and runs the turbine which in turn runs the compressor. The thermal output of the system in the form of hot air from the turbine exhaust is about 20kWt. The flow of hot air was found to be 0.05kg/s with a temperature of about 400°C. The overall efficiency of the system is about 35%. The heat exchanger effectiveness was found to be about 70%.

Pelbagai produk pertanian dan kayu memerlukan pengeringan sebelum digunakan di industri yang tertentu. Pengeringan dihasilkan menggunakan udara panas atau stim panas lampau. Penghasilan stim dan udara panas lazimnya daripada tenaga elektrik. Penggunaan elektrik adalah sangat tinggi. Sebalik itu pengeringan boleh dihasilkan dengan mengguna tenaga suria tetapi prosesnya bergantung kepada keadaan cuaca. Stim lazimnya dihasilkan di loji stim menggunakan bahanapi diesel atau biojisim. Dalam projek ini sebuah penggas/pembakar dan penukar haba suhu tinggi telah dihasilkan. Sebuah unit gas turbin mikro is pasang untuk menghasilkan udara panas melalui penukar haba. Sistem tersebut menggunakan konsep pembakaran luaran. Kuasa aci daripada turbin memutar pemampat dan menyedut udara kedalam penukar haba. Selepas itu udara dipanaskan oleh pembakar and memutarkan turbin yang kemudiannya memutar pemampat. Kuasa terma keluaran dalam bentuk udara panas adalah 20kWt. Kadar alir udara panas adalah 0.05 kg/s dengan suhu udara 400°C. Kecekapan keseluruhan adalah 35%. Keberkesanan penukar haba adalah 70%

4) Sila sediakan Laporan teknikal lengkap yang menerangkan keseluruhan projek ini. [Sila gunakan kertas berasingan]
Kindly prepare a comprehensive technical report explaining the project (Prepare report separately as attachment)

PLEASE SEE APPENDIX A

Senaraikan Kata Kunci yang boleh menggambarkan penyelidikan anda :
List a glosssary that explains or reflects your research:

Bahasa Malaysia	Bahasa Inggeris
Micro gas turbin	Micro gas turbine
Penggasan biojisim	Biomass gasification
Teknik pengeringan	Drying techniques

5) **Output Dan Faedah Projek**
Output and Benefits of Project

(a) * Penerbitan *(termasuk laporan/kertas seminar)*
Publications (including reports/seminar papers)
(Sila nyatakan jenis, tajuk, pengarang, tahun terbitan dan di mana telah diterbit/dibentangkan).
(Kindly state each type, title, author/editor, publication year and journal/s containing publication)

1) Al-Attab K. and Zainal, Z.A Externally Fired Gas Turbine for Small Scale Power Unit/Hot Air Production for Drying Processes Using Biomass fuel. International Conference on Energy and Environment. Bangi,28-29 August 2006
2)

(b) **Faedah-Faedah Lain Seperti Perkembangan Produk, Prospek Komersialisasi Dan Pendaftaran Paten atau impak kepada dasar dan masyakarat.**
Other benefits such as product development, product commercialisation/patent registration or impact on source and society

The project is a product development and upon further extensive test can be commercialized with high potential application in industries requiring drying.

* Sila berikan salinan
* *Kindly provide copies*

(c) **Latihan Gunatenaga Manusia**
Training in Human Resources

i) Pelajar Siswazah :
Postgraduate students:
(perincikan nama, ijazah dan status)
(Provide names, degrees and status)

Khalid Ali Al-Attab
M.Sc 2007
PhD start 2007

ii) Pelajar Prasiswazah :
Undergraduate students:
(Nyatakan bilangan)
(Provide number)

Numbers of undergraduate students involved in the project: 2

iii) Lain-Lain :
Others:

6. **Peralatan Yang Telah Dibeli :**
Equipment that has been purchased:

Main equipment purchased:
1. Blowers- 5 and 7.5 kW
2. Micro gas turbine unit
3. Thermocouples
4. Gasifier/combustor was designed by the school but fabricated by a company

KOMEN JAWATANKUASA PENYELIDIKAN PUSAT PENGAJIAN
Comments of the Research Committees of Schools/Centres

Projek ini telah dijalankan dengan jayanya. Peralatan yang dibina dapat digunakan dalam penyelidikan seterusnya dimasa akan datang.

b/p

TANDATANGAN PENGERUSI
JAWATANKUASA PENYELIDIKAN PUSAT PENGAJIAN
Signature of Chairman
[Research Committee of School/Centre]

18-12-2007
TARIKH
Date

Prof. Madya Dr. Zaidi Mohd Ripin
Dekan
Pusat Pengajian Kejuruteraan Mekanik
Kampus Kejuruteraan
Universiti Sains Malaysia
Seri Ampangan
14300 Nibong Tebal
Pulau Pinang

Updated : 16MAC2006

Externally Fired Gas Turbine for Small Scale Power Unit / Hot Air Production Unit for Drying Processes Using Biomass Fuel

K.A. Al-attab[a], Z.A. Zainal[b]

Universiti Sains Malaysia, School of Mechanical Engineering, Engineering Campus
14300 Nibong Tebal, Seberang Perai Selatan, Penang, Malaysia.
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Keywords: Biomass, hot air for drying, gasifier-combustor, turbocharger, externally fired gas turbine.

Abstract

Our externally fired "turbo-charger based" micro gas turbine using biomass fuel is small scale flexible unit which could be used in the simple form without the electrical generator to fulfil the industrial demands of a cheap and clean hot air for different drying processes. Comparing with the other sources of hot air of the same air temperature and flow rates, it is much cheaper and occupies a small area of ($1.1*0.7$ m²).

The unit could also be used as small scale power unit for the rural areas. It could be a better option than the small scale power units using biomass downdraft gasifiers with diesel or gas engines for the rural applications because of the complication in the gasification system due to the sensitivity of the piston engines to the tar, humidity and high temperature levels in the producer gas, so cleaning, drying and cooling systems are required, which lead to more periodic maintenance of these systems and also for the engines. All of that increases the coast of the operation, and also requires a significant training for the villagers, which is not practical in some rural places.

The biomass fuel is used as a heat soars using a gasifier-combustor. The heat power is transferred through a high temperature heat exchanger to the turbine's working fluid which is air in this project. For power unit, air is returned back as an oxygen source to the gasifier combustor, which is also a 100% recuperating process for the unit. For hot air production unit, a part of hot air after the turbine could be used for the drying process, although it reduces the amount of air that accelerates the gasification and combustion processes in the batch feeding system gasifier-combustor, but the turbine hear is not used for electrical power generation, so the turbine should only produce enough power for the compressor to sustain the high speed.

The objective of the research is to employ the turbocharger for the externally fired gas turbine cycle using off cut furniture wood as a biomass fuel, to build the system and to gain the knowledge of experimenting this promising concept.

The gasifier-combustor has been designed and fabricated in a previous research in University Sains Malaysia, and then it was modified to be suitable for this project. The high temperature heat exchanger has been designed and fabricated, and the unit has been assembled, it is now under testing and improving.

1 Introduction

Biomass is a very promising source of renewable energy in Malaysia either for electrical or thermal applications as we can see in the following table about the renewable energy source potential in Malaysia:

Renewable Energy	Energy Value (RM million per annum)
Forest residues	11,984
Oil palm biomass	6,379
Solar thermal	3,023
Mill residues	836
Hydro	506
Solar PV	378
Municipal Waste	190
Rice Husk	77
Landfill gas	4
Source: Jafaar et al. Greener energy solutions for a sustainable future Malaysia Energy Center 2003.	

Table 1: Renewable energy source potential in Malaysia[6].

For thermal applications, the drying process is one of the biggest challenges for many industries to get a cheap and clean heat source, for example, timber drying, food processing. There are more than 9000 food processing factories in Malaysia as reported in the Ministry of

International Trade and Industry 1993 (GSFTRS, 1995),and most of these industries are using the drying process as an important part of their process.

Mostly, the drying processes are using the temperature from electrical heaters or steam based drying, the second option is the most popular type for drying because it's cheaper than the electrical heaters especially if the boilers are using biomass fuels, but even though if biomass fuels are used, boilers are depending on electrical power to operate the air blowers, water pumps...etc. which makes our unit for hot air production depending on biomass fuel with almost no electrical power source during the operation (except a small turbine lubricating system) much cheaper.

For small scale electrical power applications in the range of 20-400kWe, downdraft and fluidised bed gasifiers with diesel or gas reciprocating engines have shown a promising success, but the main problem with these systems is the coast of the maintenance with more frequent regular maintenance times due to the amount of tar contained in the producer gas. And since the reciprocating engines are sensitive to the amount of tar, temperature and humidity in the producer gas which is supplied to the engine's air intake, an additional cleaning, cooling and drying systems are required to be added to the gasifiers, but even though the engines working life becomes shorter.
For the rural places with the difficulties of giving the villagers a significant training to perform all the operation and maintenance duties in a correct way, so the system could fail due to the poor maintenance.

In our unit, since the heat exchanger has much less sensitivity to the effects of the producer gas comparing with the reciprocating engines, and most of the producer gas is burned totally before reaching the heat exchanger, so no additional systems are required. Also the turbocharger engine is much simpler than the reciprocating engine and it deals with a pure air with maximum 700 °C turbine inlet temperature which leads to less maintenance requirements, so the unit is expected to be more suitable for the rural applications.

2 Externally fired gas turbine preparation

The externally or indirectly firing of the gas turbine means that the combustion chamber is not directly connected to the gas turbine therefore the combustion exhausted gases are not inserted directly to the turbine and not in direct contact with the turbine's impeller.
The combustion process is used hear for heating up a compressed fluid (commonly air) using a high temperature heat exchanger, then the fluid is expanded in the turbine producing a high rotary speed shaft power.

The indirectly and directly fired gas turbine, both are similar in concept and both are explained thermodynamically by the Brayton cycle. In the indirect fired open Brayton cycle (see fig. 1), the working fluid is drawn by the compressor (point 1) and compressed (2), passes through a heat exchanger for heating up the working fluid (3), expands through the turbine (4), and either discharged directly to the environment from the turbine or returned back to the compressor after A cooling process.
The combustion process (a-b) is assumed to be under a constant pressure. Thermal power (Q) is transferred to the working fluid by the heat exchanger in process (2-3).

This cycle is shown in the temperature-entropy (T-S) diagram (fig. 1). Isentropic processes (1-2 and 3-4) assumed that the compressor and turbine are 100% efficient.

The actual efficiency of the components combined with the heat exchanger efficiency results in to a lower overall efficiency.

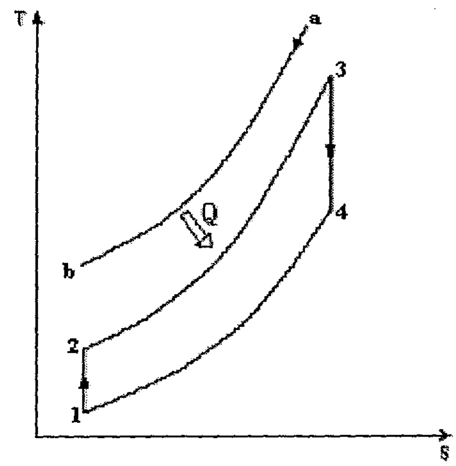

Fig. 1: Temperature and entropy relation for an externally fired open Brayton cycle.

The externally fired gas turbine (EFGT) has the advantage of freedom in choosing the fuel source either liquid, gas or even solid type of fuels like coal and biomass fuels, this advantage is also available in the Rankine steam cycle, but it has a lower thermodynamic efficiency comparing with the Brayton cycle which has a higher temperature of the working fluid in the inlet of the turbine with a lower pressure comparing with steam temperature and pressure in Rankine cycle.

The direct fired gas turbine (DFGT) has a higher thermodynamic efficiency comparing with the (EFGT) because of the higher temperature of the combustion gases in the turbine's inlet, but in the other hand it can only deal with the clean liquid or gas fuels along with the fuel compressing and injecting equipments which is not necessary in the (EFGT). The (DFGT) can use the solid fuels like biomass or coal after gasification process only after an intensive cleaning process for the producer gas. (M. Anheden, 2000)

The externally fired gas turbine cycle can be divided in to two types:

- The open cycle externally fired gas turbine.
- The close cycle externally fired gas turbine.

In the open cycle, the working fluid is discharged to the combustion chamber as a heat and oxygen supply for the combustion process "as in our case", or discharged to the environment after decreasing its temperature in a heating or drying process.
In the closed cycle, the working fluid is returned back to the compressor after a cooling process.

3 Combustor

The combustor is a gasifier- combustor at University Sains Malaysia (USM). The gasifier is a small scale multi-chamber combustor with two stages of chamber namely, gasifier chamber and combustor chamber. Both parts are made of a refractory cement layer and covered with a mild steel sheet. It is a batch feeding system gasifier combustor, biomass feeding port is placed on the top of the gasifier chamber to feed the wood into the gasifier, and it has a fit clamped and isolated cover.

Air is supplied to the secondary chamber at almost atmospheric pressure after the expansion in the turbine. Air is passing through a swirling flow generator to provide a good mixing with the gases from the gasifier chamber. A complete combustion in the combustion chamber creates a very hot zone which pushes the gases upward and creates a stack effect which creates a vacuum pressure in the gasifier chamber and pulls the air through a controlled air intake in the down side and in to the gasifier chamber to complete the gasification process.

Assuming the gasifier is producing a thermal power of 100 kW, and assuming that the efficiency of the gasifier is about 70%, the energy content for wood (20% moisture content) Cv = 15 MJ/kg, so we can calculate the biomass fuel consumption using Eq. 1:

$$Q = \dot{m}_w . Cv \quad . \qquad (1)$$

Wood mass flow rate will be around $\dot{m}_w$ = 34.3 kg/hr.

To calculate theoretically the mass flow rate of the producer gas out of the gasification chamber, for calorific heating value per unit weight of about Cp = 4000 KJ/kg, from the power balance equation (Eq. 2):

$$Q = \dot{m}_g . Cp \quad . \qquad (2)$$

All air properties will be taken at 800 k, and all combustion gases properties will be taken at 1000 k.
Cp air = 1.099 KJ/ kg.K, Cp gas = 1.14 KJ/kg.K.

The mass flow rate of the producer gas is around $\dot{m}_g$ = 0.025 kg /s.

4 Heat exchanger

The high temperature heat exchanger (H.T.H.E.) is the key to the success in the (EFGT), it is required to transfer heat from the heat source like combustion chamber or furnace or even a nuclear reactor to the working fluid of the gas turbine. The higher temperature the heat exchanger can provide the higher the system efficiency will be. The main issue is how to build a heat exchanger that can withstand the stresses caused by the working conditions, considering a reasonable building cost.

With the nickel-based super alloys heat exchanger, the turbine inlet temperature could reach 800-825 °C (M. Anheden, 2000), and in many projects they are using an additional natural gas combustor to rise the temperature up to around 1100 °C to increase the cycle efficiency. However, ceramic heat exchangers can reach such a high turbine inlet temperature with a long operational life time, but the building coast of these heat exchangers are very high which takes a very long pay back time, but this option might be economical in the future with further developments for the ceramic heat exchangers technology to reduce its costs.

The high temperature heat exchanger has been designed and fabricated in the school of mechanical engineering, University Sains Malaysia (USM), and the design was based on two pass tube and shell heat exchanger type.

The combustion gases temperature is assumed to be 1000 °C, and the air temperature after the heat exchanger is assumed to be just 600 °C to keep the (H.T.H.E.) and the turbocharger in the safe side for the testing process, even though it reduces the efficiency of the (H.T.H.E.) to around 55%. We can calculate the power transferred to the air through the heat exchanger using Eq. 3:

$$Q = \dot{m}_a . Cp . (\Delta T_a) \quad . \qquad (3)$$

The power will be around Q = 54 kW.

5 Turbocharger

Turbochargers have been used for almost 80 years to increase the power of reciprocating engines by compressing the inlet air to the engine.
The compressor is taking the rotating torque from radial flow turbine which is placed in the exhaust of the engine, expanding the high speed exhaust gases.
The main construction of the turbocharger is the impeller of turbine and the impeller of compressor which are placed on the same shaft, supported with two bearings. This main construction of the turbocharger is typically the same main construction of one shaft micro gas turbine, or we can call it the hart of the micro turbine.
In this project we are going to use a truck turbocharger to build a micro gas turbine (see fig. 2) because the turbocharger is very cheap comparing with the micro turbine engine.

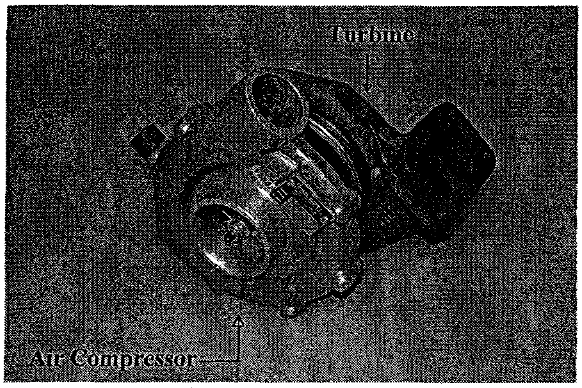

Fig. 2: Turbocharger.

The compressor air pressure is about 3 bar absolute and the air flow rate is assumed to be low because the turbocharger is not expected to run in full speed because of the low turbine inlet temperature, so air flow rate will be around $\dot{m}_a = 0.12$ kg/s.

6 Lubrication system

It is a secondary system to lubricate and cool the turbocharger's bearings, but that doesn't mean it is not important, because our turbocharger depends totally on the lubrication system, because the rotating shaft is totally floating on the oil film which is also acting like a damping system to eliminate the vibrations. The modern micro turbine engines are using air bearings with no oil, but the turbocharger is more suitable for our project for it is much cheaper.

The main components of the lubrication system are:

1. High pressure automotive oil gear pump.
2. Driving electrical motor.
3. Oil cooling unit.

5 Theoretical calculations

Here are some theoretical calculations for the system, assuming adiabatic efficiencies for the turbine and compressor of η_t = 80% and η_c = 75%. Using the thermodynamic analysing of the cycle (see T-S diagram in fig. 3), we find that the compressor power will be around 18 kW and the total power from the turbine would be around 23 kW, so the turbine net power would be around 5 kW.

For the unit as a power unit, the hot air after the turbine will be totally recuperated and around 49 kW of thermal power will be returned back to the combustion chamber and reduces the wood fuel consumption to about 17.5 kg/hr, so the overall efficiency of the unit will be around 7 %. We can increase the cycle efficiency using a bigger turbocharger that can handle more air flow rate to increase the heat exchanger efficiency up to around 80% and using a heat recovering systems for the hot chimney gases.

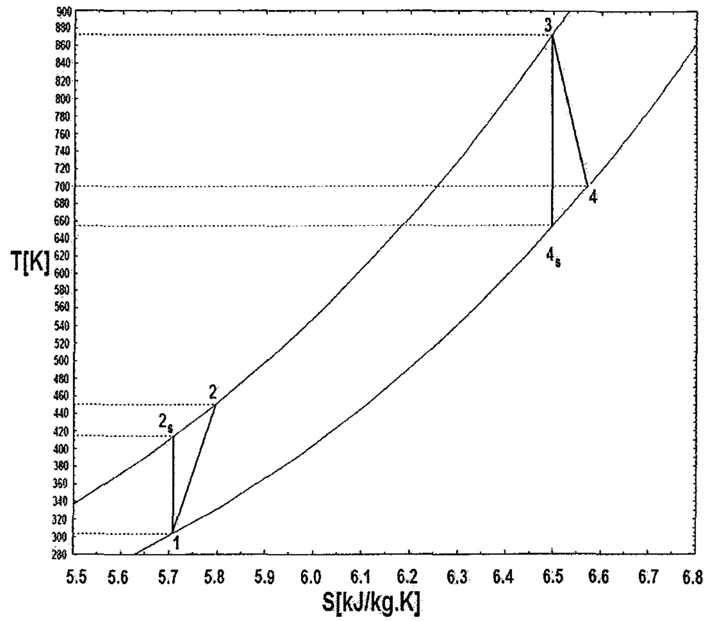

Fig. 3: T-S diagram for our cycle

The ideal cycle has 24% efficiency, however the actual cycle is: η_{cycle} = 9.3%.

For the system as a hot air production unit with almost no recuperating process, the producer gas flow rate was calculated to be around (0.025 kg/s), the air/fuel ratio is about (1), so we will use around (0.03kg/s) air to complete the combustion and the rest of air flow rate after the turbine which is around (0.09 kg/s) could be use for drying process, so we can calculate the power of the hot air using Eq. 4:

$$Q = \dot{m}_{a,d}.Cpa.(T_1 - T_2) \quad . \tag{4}$$

$\dot{m}_{a,d}$ = 0.09 kg/s, the air temperature after turbine is around T_1= 425 °C and the ambient temperature is round T_2 = 30 °C, so the air thermal power will be around Q = 39 kW.

If we calculated the cost of this amount of thermal power using electrical heaters assuming the converting efficiency from electrical power to a hot air flow is about 70%, since we are using air blower to get an air flow through the heaters, so the electrical power will be around (55.7 kW_e) and if we are using the unit for around 20 hr/day, so we can save around RM10,860 per month for 32.5 sen/kWh (Tenaga National Berhad, 2006) using a free waste biomass fuel.
The fuel consumption per month will be around (20.6 ton of wood).

6 Operation

The hot air production unit has been tested. Figure 4 shows a schematic drawing of the cycle.
We have 13 thermocouples to detect the temperature, 3 pieces for the gasifier, 6 pieces for the combustion chamber, 3 pieces for the H.T.H.E. and one piece after turbine.

First we start the gasifier-combustor using L.P.G. flame for around 10-15 minutes then we use an air blower for starting the turbocharger engine. The air temperature and pressure increase gradually until the turbine side could produce enough power for the compressor to sustain the speed, then after removing the blower, the turbocharger engine takes speed up to around $(60\text{-}70)*10^3$ R.P.M. then we can open the hot air valve for drying process.

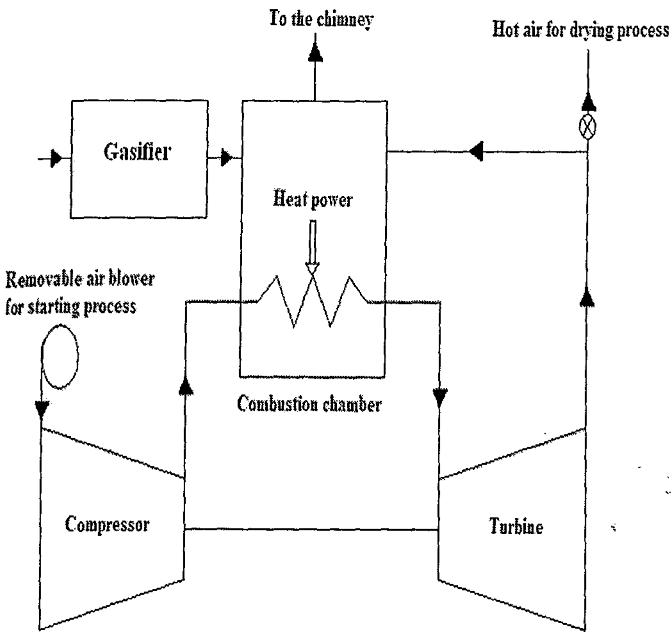

Fig. 4: Externally fired gas turbine for hot air production using biomass fuel.

The power unit cycle is shown in figure 5. The power generation set has not been installed yet.

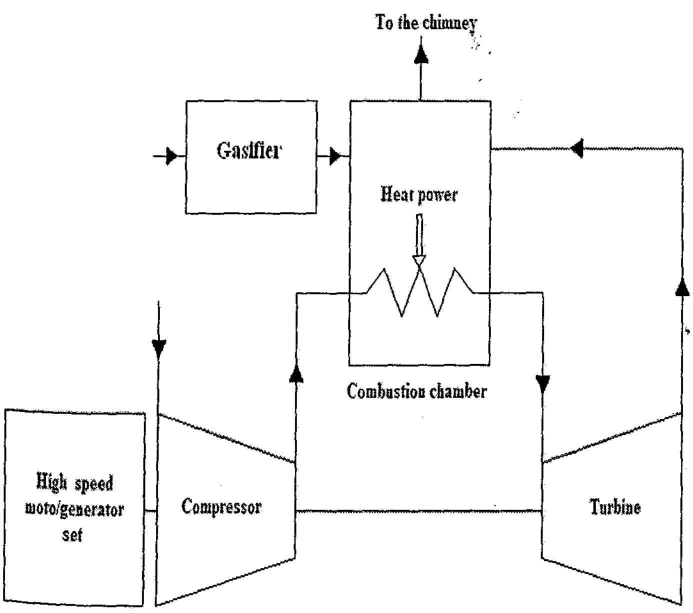

Fig. 5: Externally fired gas turbine for electrical power generation using biomass fuel.

There are two options for the power generation set:

- High speed generator with electronic converting system to 380V/50HZ, the primary coast is high but the manufacturers provide information about the long life time with low maintenance, but still it is not confirmed if the electronic system can stand the harsh conditions in Malaysian rural areas.
- Low speed generator with speed reduction gear box, it is much lower in the primary coast but it needs a periodic maintenance sine it has a lubrication system.

Starting of the turbocharger engine will be mechanically using the electrical generator as a motor.

7 Conclusion

Using the gasifier-combustor with the indirectly fired gas turbine cycle has shown a very successful and simple combination, also employing the turbocharger as a gas turbine engine has shown a promising concept with a sharp decrease of the unit building price. Design and fabrication of the high temperature heat exchanger should be under a special care, because it is the key to the success of the unit. More experimenting and developing are recommended for the unit to increase the reliability either for the industrial drying process or for the rural places electrical power generation.

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Performance of High Temperature Heat Exchanger in Externally Fired Gas Turbine System Using Biomass Fuels

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Abstract

Lately, the idea of the externally firing of micro gas turbine using biomass fuel has been studied in a wide range, but a few of these studies are using experimental work to evaluate such a system. In Malaysia such a system has not been employed so far within the biomass projects for thermal or power generation applications. The objective of this study is to determine the performance of a stainless steel high temperature heat exchanger that has been built to transfer thermal power from a biomass gasifier-combustor in to the turbine's working fluid that is a pure air in this case. The study is based on experimental work using different capacities air blowers as an air supply for this experiment. The heat exchanger has achieved 694°C turbine inlet temperature with average efficiency and effectiveness of about 50% and 62.5% respectively.

Keywords: Biomass; heat exchanger; externally fired gas turbine; gasifier-combustor; turbocharger.

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1. Introduction

In the last twenty years, the amount of studies on the externally firing of the gas turbines (EFGT) has increased dramatically especially with biomass fuels. Some of these studies [1] focused on the efficiency of the (EFGT) system and how to increase it using combined heat and power systems or steam and gas turbines systems. Other studies [2], [3] compared between different systems either with metallic or ceramic heat exchangers using some minor systems like heat recuperating units, intercoolers or auxiliary natural gas burners. The advantage of the biomass pre drying process using chimney thermal power also has been studied [4], [5], and even the organic waste fuels like sewage sludge and poultry wastes with the (EFGT) system have been investigated [6], [7]. Small (EFGT) power units for rural areas have been studied [8]. These studies have done computer simulation to get the results, but only few experimental studies [9], [10] were conducted to evaluate the (EFGT) system because of the high cost of manufacturing such a system.

The objective of this paper is to study the performance of the high temperature heat exchanger due to differant air pressures and flow rates, more specifically to study the dynamic temperature profiles, the heat exchanger effectiveness, the temperature distribution along the combustion chamber and the efficiency of the heat exchanger.

2. Externally fired gas turbine system

The externally or indirectly firing of gas turbine means that the hot combustion gases are not in direct contact with the turbine, and the turbine's working fluid is separated from these gases, so in order to increase enthalpy of the working fluid, the thermal power from the combustion gases should be transferred to the working fluid using a high temperature heat exchanger.

The directly and indirectly fired gas turbine both are explained thermodynamically by the Brayton cycle. However, the direct fired gas turbine has a higher efficiency since it has a higher turbine inlet temperature, but it can deal only with a clean gas or liquid fuels. It can also deal with solid fuels like biomass only after a gasification process and an intensive cleaning process. Externally fired cycle can deal with a wide range of fuels even solid fuels without cleaning systems or fuel compression and injection equipments, and also requires thermal power sources like combustors or furnaces.

3. System description

The waste wood from furniture industry has been used to fire a multi chamber batch system gasifier- combustor of about 45 kg wood maximum capacity. A high temperature heat exchanger is placed inside the secondary chamber of the gasifier-combustor (see fig. 1) to transfer thermal power to the micro turbine engine. A liquefied petroleum gas (LPG) burner is used for the gasifier starting process which takes about five minutes.

Producer gases completely burn directly after the throat between the two chambers. The complete combustion creates a very hot zone and a stack effect that induces the air through a controlled air opening at the bottom of the gasifier chamber to aid the gasification process. HOLSET-3LD turbocharger was used as the micro turbine engine. The compressor of turbocharger can handle a maximum air flow rate of about 0.25 kg/s.

4. Heat exchanger design

The preformance of externally fired gas turbine unit depends mostly on the design of the high temperature heat exchanger, and the higher temperature it can handle, the higher the efficiency of the unit. However, since the unit is considered to be a low cost unit, suitable for industrial drying processes or rural areas as small power generation unit, the development cost of the heat exchanger should be considered as a major factor. Ceramic heat exchangers can reach a high turbine inlet temperature, more than 1000 °C with a long operational life time, but the development cost of these heat exchangers are very high and requires a long pay back period, but this option might be economical in the future with further developments for the ceramic heat exchangers technology to reduce its costs. Nickel-based super alloys allow turbine inlet temperature to reach 800-825 °C [11].

Many type of heat exchangers have been considered in the design, the two pass cross flow with baffles shell and tube type was found to be the most suitable for such a system. Using (ε.C*.NTU) method in the design, the number of transfer units NTU = 1.14, the heat capacity ratio C* = 0.8 and the heat exchanger effectiveness of The design $\varepsilon = 0.53$. The following Nusselt number correlations (Eqs. 1&2) [12] have been used for the tube and shell side calculations respectively:

$$Nu = 0.024\, Re^{0.8} . Pr^{0.4} \tag{1}$$

$$Nu = 1.04\, Re^{0.4}\ Pr^{0.36} (Pr/ Prw)^{0.25} \tag{2}$$

Where (Re) is the Reynolds number, (Pr) is the Prandtl number and (Prw) is the Prandtl number due to the wall temperature. The air is inside the stainless steel-304 tubes (21mm inner dimeter) which are of about 2.5mm wall thickness and external tube surface area of about 8.1 m^2. The heat exchanger was designed to transfer about 57 kW thermal power to the turbine's working fluid that is pure air and the air flow rate is expected to be about 0.12 kg/s with a pressure of about 2 bar (gauge pressure). The turbine inlet temperature is expected to be about 600 °C and the efficiency is expected to be low of about 57% due to the low air temperature and flow rate.

In order to determine the thermal performance of heat exchanger, effectiveness of the heat exchanger (ε) is calculated from Eq. 3:

$$\varepsilon = \frac{C_{air}(T_{air\ out} - T_{air\ in})}{C_{min}(T_{gas\ in} - T_{air\ in})} \times 100 \tag{3}$$

Where C_{air} and C_{gas} are the heat capacity rate for air and gas respectively; C_{min} is the minimum heat capacity rate and it is equal to C_{air} in this design, it is calculated from Eq. 4:

$$C = \dot{m}.C_p \tag{4}$$

To calculate the efficiency of the heat exchanger, the mass flow rate of the hot combustion gases should be calculated from Eq. 5:

$$C_{air}\left(\Delta T_{air}\right) = C_{gas}\left(\Delta T_{gas}\right) \tag{5}$$

The total thermal power of the combustion gases is calculated from Eq. 6:

$$Q_{combustion\ gases} = C_{gas}\left(T_{gas\ in} - T_{ambient}\right) \tag{6}$$

Thermal power transferred to the hot air through heat exchanger is calculated from Eq. 7:

$$Q_{\text{air}} = C_{\text{air}}\left(\Delta T_{\text{air}}\right) \tag{7}$$

Efficiency of the heat exchanger (η_{HE}) is calculated from Eq. 8:

$$\eta_{HE} = \frac{Q_{\text{air}}}{Q_{\text{combustion}}} \times 100 \tag{8}$$

In order to understand the main factors affecting the heat exchanger efficiency, Eq. 8 was simplified in to Eq. 9:

$$\eta_{HE} = \frac{T_{\text{gas in}} - T_{\text{gas out}}}{T_{\text{gas in}} - T_{\text{ambient}}} \times 100 \tag{9}$$

5. Experiment set up

Different size air blowers were used to study the performance of the high temperature heat exchanger for different air pressure and flow rates. Figure 2 shows the system layout for this study, but in subsequent experiments, the air blowers were used for turbine startup process.

The measuring tools required in the experiment are to determine temperature, pressure and air flow rate. For temperature measurment nine type K thermocouples were connected to a 12-channel thermocouple scanner to determine the temperature profiles of the heat exchanger, turbo charger and the combustion chamber. For the heat exchanger and the turbocharger, four thermocouples were used to detect temperatures T_1 to T_4 (see fig. 3) and five thermocouples for the temperatures T_5 to T_{10} inside the combustion chamber (see fig. 4). Hot wire anemometer and air pressure gauge were used to determine the air flow rate and pressure respectively inside the heat exchanger.

6. Results and discussion

For the gasifier-combustor starting process with the LPG burner, five minutes was enough for good startup, and after the first 35-40 minutes the gasifier became stable with fuel size of about 150 mm wood blocks. The 45° inclination inside the primary chamber was insufficient for the wood blocks to drop smoothly to the combustion zone and bridging was noticed during the fuel reloading process, this caused some disturbance in the gasifier performance. The inclination was subsequently increased to 70° and bridging phenomenon was not noticed.

Different air pressures and flow rates were applied inside the heat exchanger's tubes, and since the air was totally returned back to the combustion chamber after the turbine, so the air flow rate is the main factors that led to a different temperature profiles of the heat exchanger. The batch biomass fuel feeding system did not affect the temperature profile significantly as long the amount of biomass fuel is enough inside the gasifier chamber, unlike the continuous feeding system where the temperature profiles are sesitive to the feed rate of the fuel.

Figure 5 shows the temperature plots for the ten thermocouple channels via time for one of the expirements. A sudden drop in temperature for about 1-2 minutes can be seen on plots T_4 to T_9 due to the fuel reloading process, however the temperatures inside the heat exchanger T_2 and T_3 were insensitive to this process.

The air pressure and flow rate could only be increased gradually after the turbine power was sufficient to support the compressor. Maximum air pressure and flow rate of the blowers were 0.65 bar and 0.1 kg/s respectively. The turbine inlet temperature was 694 °C that is more than the designed temperature.

The air flow rate has a direct proportion with the air flow speed, since the heat exchanger geometry is fixed. Therefore, the more the air flow rate, the higher the heat transfer coefficient. This goes for the combustion gases flow rate as well. Also, all the air flow was returned back to the chamber. Thus, any increase in air flow rate will accelerate the gasification and combustion process and increase the system temperature. Figure 6 shows the relation between the air flow rate and the air temperature in the outlet of the heat exchanger.

The heat exchanger design calculations were based on high air flow rate expected from the compressor side of the turbocharger in the self running mode. The heat capacity rate for gas (C_{gas}) was designed to be more than the heat capacity for air (C_{air}). However, in the experiments, it was noted that $C_{gas} < C_{air}$ in the beginning. But, when the air flow rate increased, C_{gas} became more than C_{air} as designed.

Figure 7 shows the changes in the effectiveness and efficiency againest the air temperature. In the beginning, another equation (Eq. 10) was used to calculate effectiveness for $C_{gas} < C_{air}$.

$$\varepsilon = \frac{(T_5 - T_{10})}{(T_5 - T_1)} \times 100 \tag{10}$$

The difference between (T_{10}) and (T_1) started to increase causing a drop in the effectiveness. However, at high temperature (T_3 = 500 °C), the heat capacity $C_{gas} > C_{air}$ and the effectiveness values started to stabilize when the rate of the air temperature rise became equal or even more than the increasing rate in gas temperature.

For the heat exchanger efficiency, the efficiency presents the ratio between air power and combustion gases power. The gas power depends on the difference between gas inlet temperature (T_5) and the ambient temperature unlike the effectiveness calculations which do not include the ambient temperature that is lower than the air inlet temperature (T_1) in our case. That produced a lower efficiency compared to the effectiveness. In the beginning, the stack temperature (T_{10}) was low causing a high efficiency. The rate of the (T_{10}) temperature rise increases with more temperature stored in the system causing more efficiency drop as can be note from the efficiency plot (fig. 7).

Figure 8 shows the effect of the air flow rate on the heat exchanger performance in the heating up period. The air and gas flow rates can affect the efficiency and effectiveness of the heat exchanger indirectly by increasing the overall heat transfer coefficient of the heat exchanger. However, the high heating rate has reduced the efficiency of heat exchanger as mentioned earlier. But, on the other hand, the turbocharger in the tests was not operating in the self running mode to provide a high flow rates that can overcome the other factors and increase the performance of the heat exchanger. The air pressure was in the range of 0.5-0.65 bar during the tests and did not show a significant effect on the effectiveness.

Figure 9 shows the temperature distribution on the combustion chamber and heat exchanger at maximum temperature, there is no crossing between the temperature lines of combustor and heat exchanger, which means, no air cooling process at any part of the heat exchanger.

The following efficiency and effectiveness results were calculated after the air temperature stabilized over 500 °C with constant flow rate around 0.1 kg/s. The average efficiency and effectiveness of heat exchanger were about 50% and 62.5% respectively. However, in the self running mode, the turbocharger can handle a flow rate of about 0.25

kg/s for maximum turbine speed and the turbine inlet temperature can rise up to 700 °C, that will increase the efficiency of the heat exchanger and the over all efficiency as well.

7. Conclusion

The externally firing of gas turbine using biomass gasification process was presented, and the performance of the high temperature heat exchanger was encouraging. The efficiency is expected to increase when turbocharger is running on full speed. The full evaluation of the heat exchanger could only be done after starting up the turbine and operating the unit in the self running mode for extended period, to determine the characteristics of the system and the behaviour of the heat exchanger materials due to continuous high thermal stresses.

The gasifier-combustor has shown good performance and fast starting up even with large wood blocks, and along with the EFGT system it showed a successful and simple combination.

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List of Figures

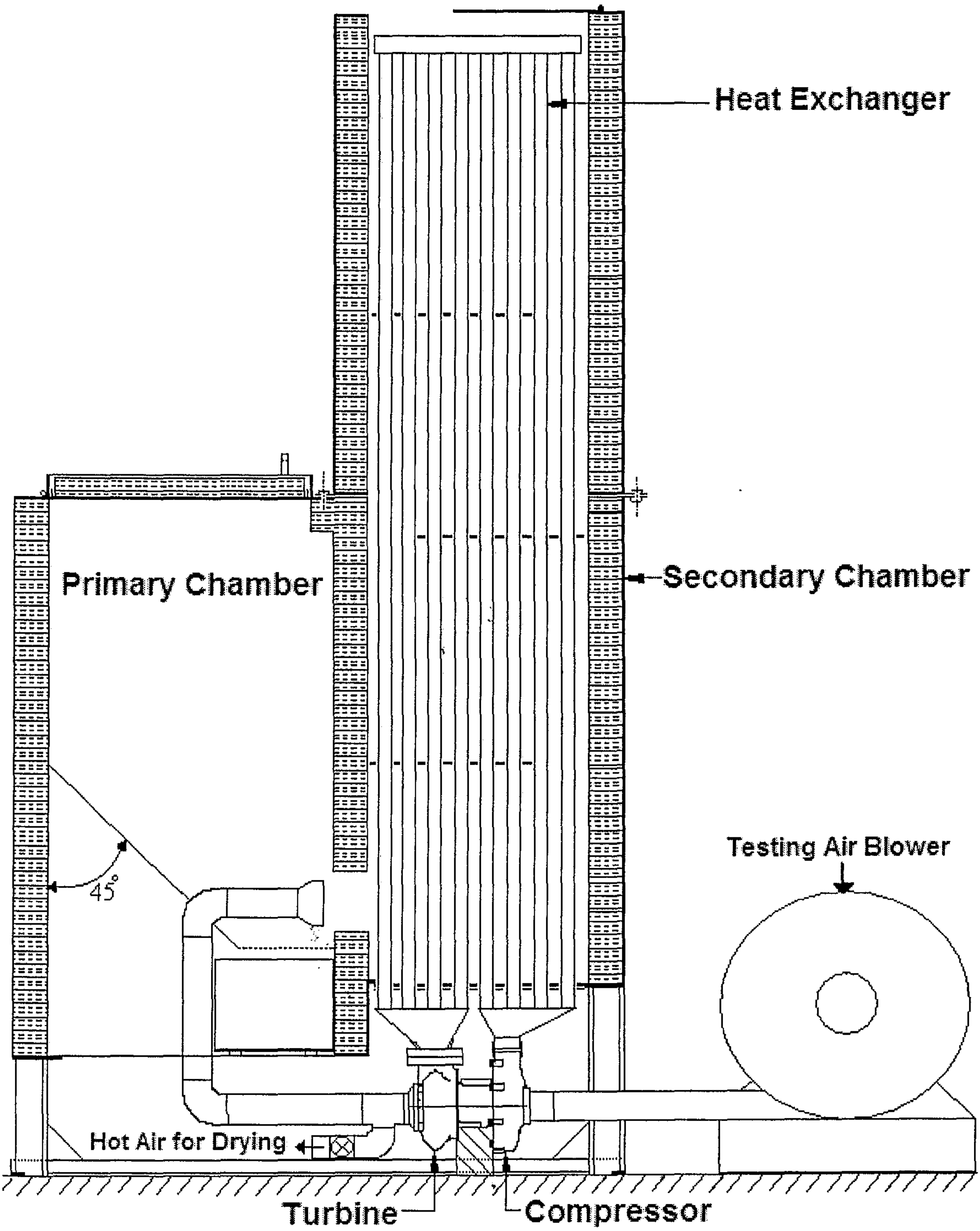

Heat Exchanger
Primary Chamber
Secondary Chamber
Testing Air Blower
45°
Hot Air for Drying
Turbine
Compressor

Figure 1

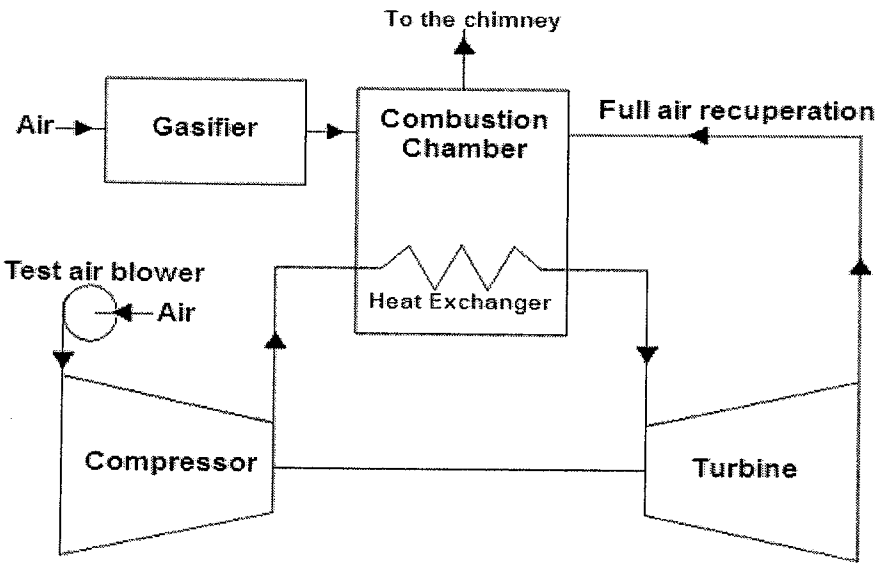


Figure 2

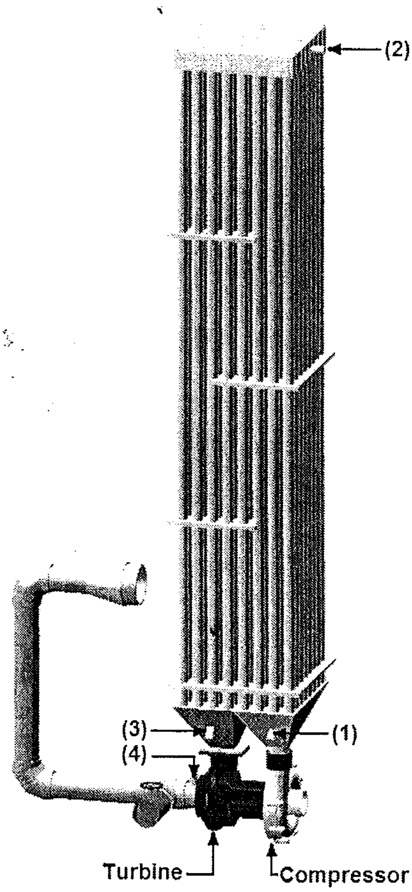


Figure 3

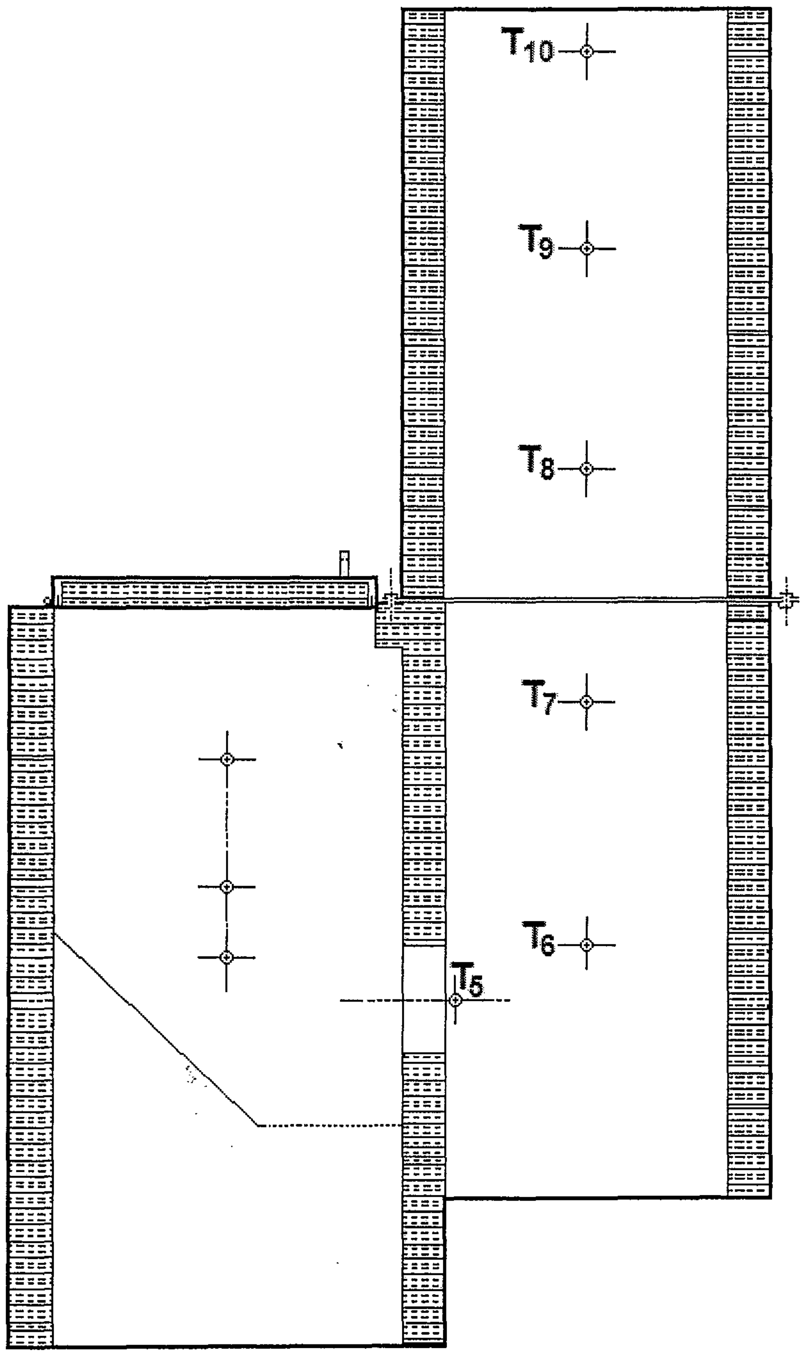


Figure 4

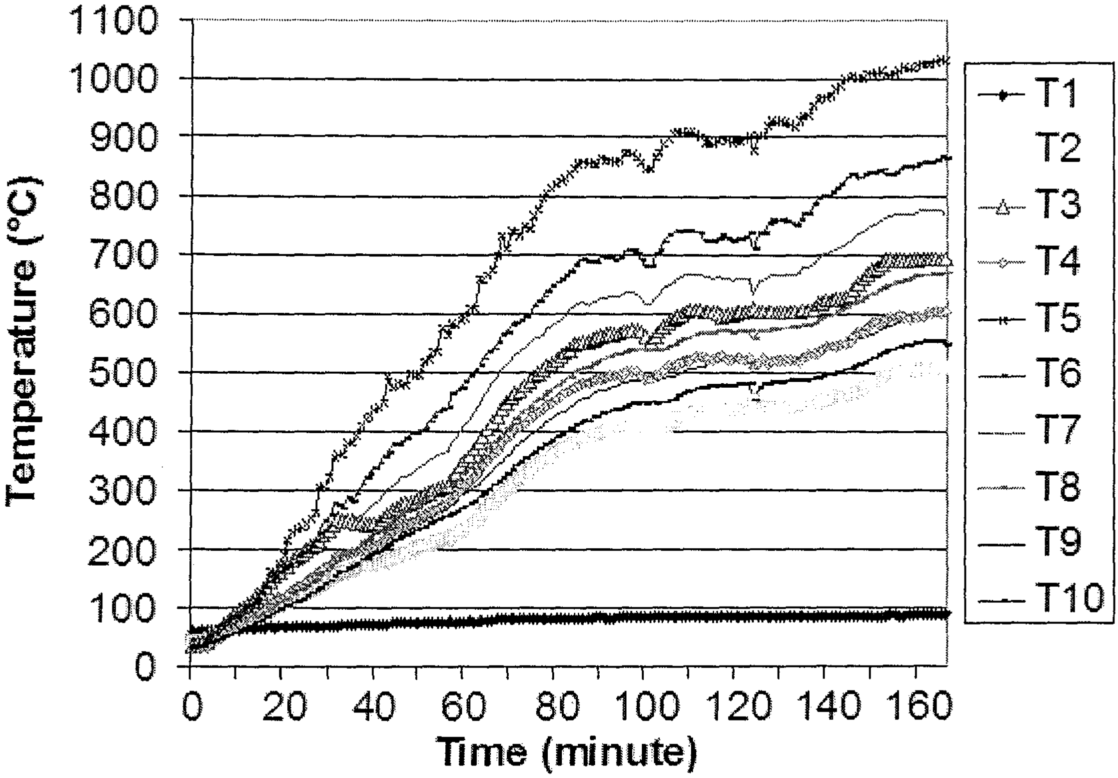


Figure 5

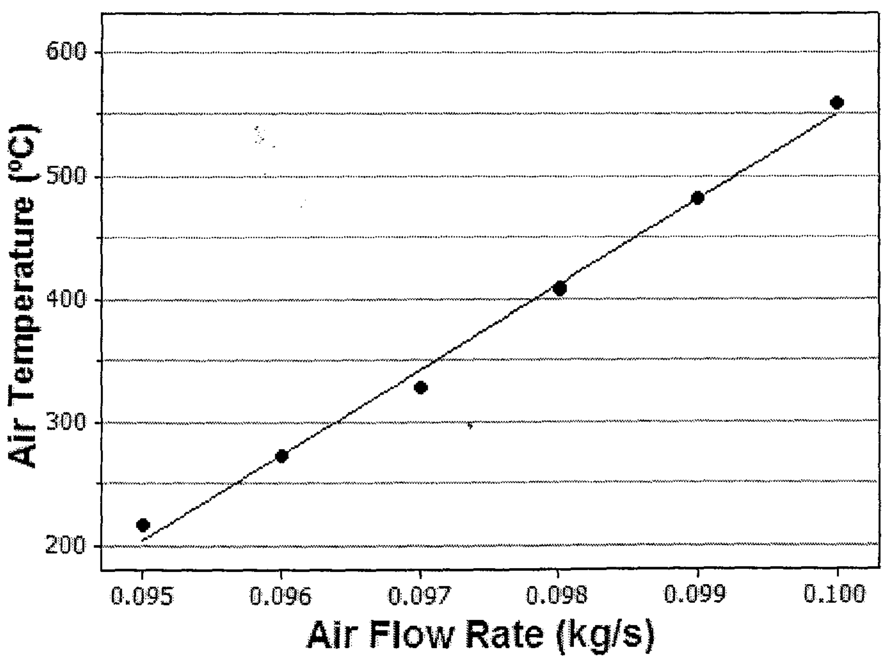


Figure 6

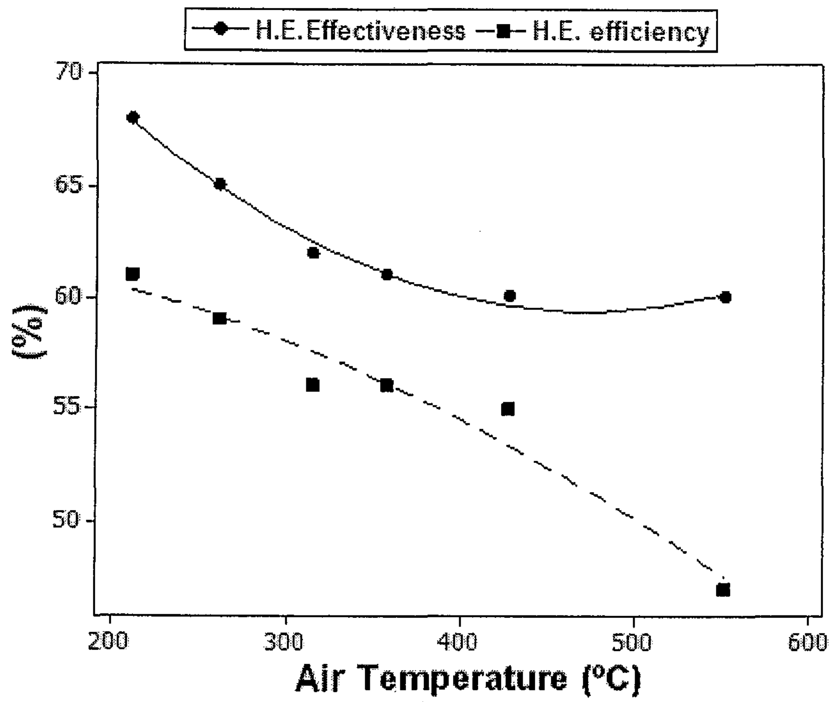


Figure 7

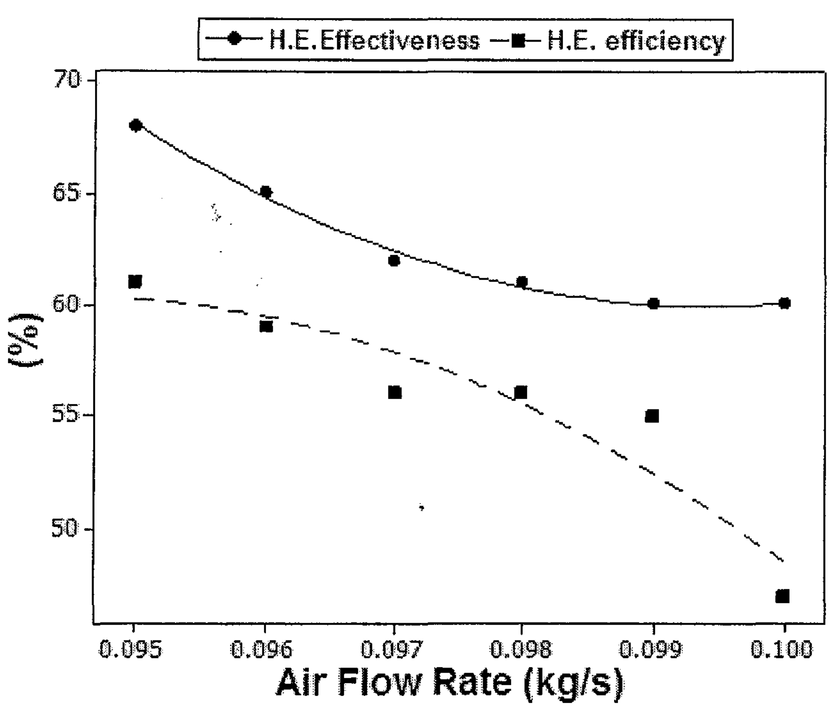
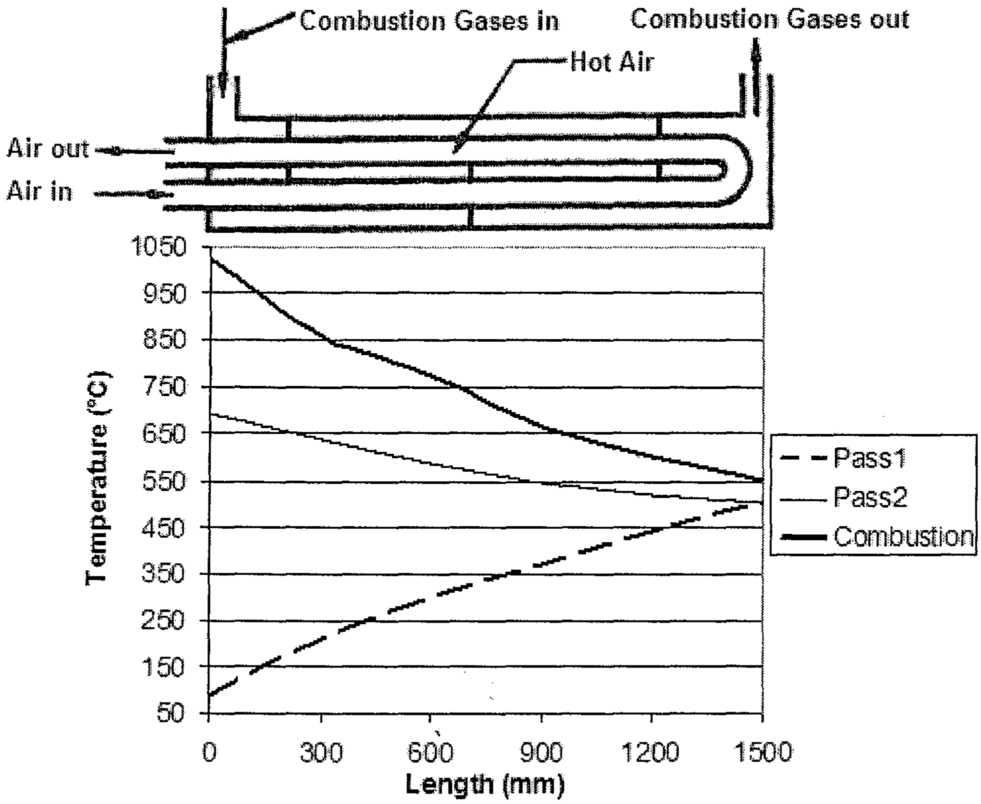


Figure 8



Combustion Gases in
Combustion Gases out
Hot Air
Air out
Air in
Temperature (°C)
1050
950
850
750
650
550
450
350
250
150
50
0
300
600
900
1200
1500
Length (mm)
Pass1
Pass2
Combustion

Figure 9

DESIGN OF HIGH TEMPERATURE HEAT EXCHANGER FOR THE INDIRECTLY FIRING OF A MICRO GAS TURBINE USING BIOMASS FUELS

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Abstract: Biomass fuels have been used lately in a wide range for power generation using a directly fired gas turbine systems. However, the high maintenance cost for these systems has led to more investigation on the indirectly fired gas turbine method. The key element in this method is the high temperature heat exchanger that is required to transfer enough power and temperature to the turbine working fluid. A small scale indirectly fired micro gas turbine system using biomass fuels have been designed, fabricated and tested at the school of mechanical engineering, Universiti Sains Malaysia. This paper summarizes the design elements and calculations of the high temperature heat exchanger, with a comparison with the experimental results. The heat exchanger has achieved 694°C turbine inlet temperature with average efficiency and effectiveness of about 50% and 62.5% respectively.

Keywords: Biomass; heat exchanger; indirectly fired gas turbine.

INTRODUCTION

In Malaysia, the big potential and variety of biomass sources makes this type of energy one of the most promising sources of energy. For small scale electrical power applications in the range of 20-400kWe, downdraft and fluidised bed gasifiers with diesel or gas reciprocating engines have shown a promising success. The direct firing of gas turbine with gasifier systems for electrical power generation has also been used even with higher power range more than 400kWe.

The main problem with these systems is the coast of the maintenance with more frequent regular maintenance times due to the amount of tar contained in the producer gas. Additional cleaning, cooling and drying systems are required to be added to the gasifiers, but even though the turbine and reciprocating engines working life becomes shorter.

The externally firing of the micro gas turbine (EFGT) can eliminate the producer gas cleaning issue by the direct combustion of the producer gas after the gasifier, with no gas compression or injection equipments. Thermal power is transferred though a high temperature heat exchanger into a pure air. The hot air produced by the heat exchanger expands in a turbine engine for power generation and then the hot air can be used as a thermal power output of the system.

In EFGT systems, Switching between different types of fuel is much easier comparing with the conventional power systems. Moreover, changing the thermal power source other than the combustion power source is also possible.

The advantages of EFGT concept have attracted many companies and research institutes to investigate such a system. However, only few experimental studies [1-3] were done on the EFGT systems due to the high development cost. Most of the studies are using theoretical analysis or simulation programming to evaluate the EFGT systems. For example, investigating the possibilities of increasing the efficiency of the EFGT systems [4],[5], or comparing between different EFGT systems either with metallic or ceramic heat exchangers using some minor systems like heat recuperating units, intercoolers or auxiliary natural gas burners[6], [7]. The advantage of the biomass pre drying process using stack thermal power also has been studied [8], [9], and even the organic waste fuels like sewage sludge and poultry wastes with the EFGT system have been investigated [10], [11]. Small EFGT power units for rural areas have been studied [12].

Different type of heat exchangers will be discussed in this paper in order to find out the most suitable type for this gasifier-combustor EFGT system. The objective of this paper is to summarize the design elements and calculations of the high temperature heat exchanger, with a comparison with the experimental results. Also, to study the performance of the heat exchanger in terms of the efficiency and effectiveness with different air flow rates.

SYSTEM DESCRIPTION

The system has been developed in a small scale based on the indirectly firing of the micro gas turbine theory. The main components of the system are:

- Gasifier-combustor: that consists of two chambers, the gasifier chamber and the combustion chamber. It is a batch fuelled unit with output thermal power of about 100kWt and the biomass fuel type used during the experiments was the off-cut furniture wood blocks.
- High temperature heat exchanger: that transfers thermal power from the combustor in to a pure air that is the working fluid of the turbine.
- Micro turbine engine: Holset truck turbocharger of about 2.5 kg/s maximum air flow rate was used as the micro gas turbine engine.

Figure 1 shows the layout of the EFGT system. The producer gas from the gasifier was totally combusted in the combustion chamber. Combustion thermal power was transferred through the high temperature heat exchanger to the turbine working fluid (air) that expands through the turbine. In the experiments, some of the hot air stream after the turbine was released to the ambient as the hot air output of the system. The rest of the hot air was returned back to the combustion chamber as a thermal power and also an oxygen source for the combustion process. The turbine startup process used an air blower, as shown in the system schematic drawing (fig.2).

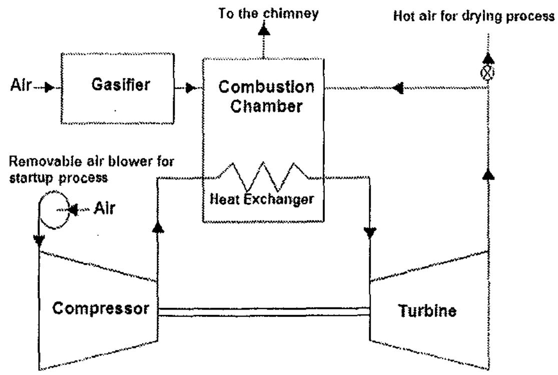

Fig. 1: EFGT system layout

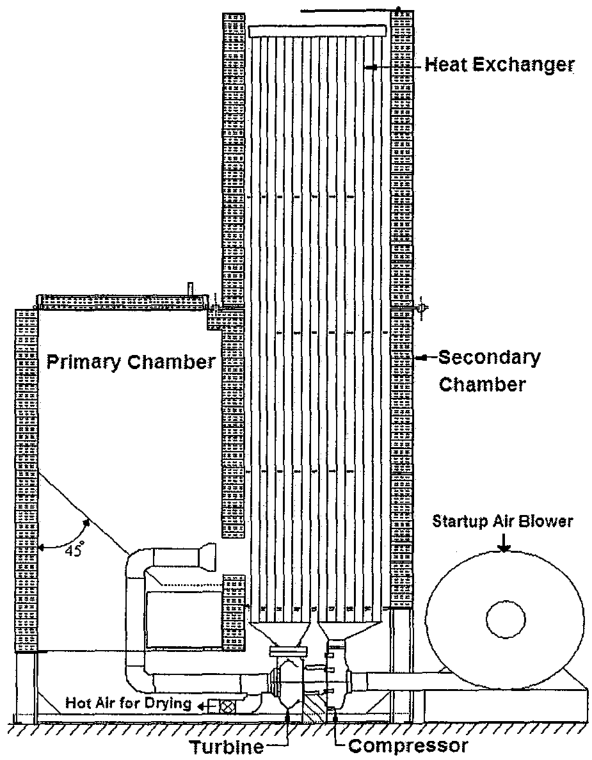

Fig. 2: Externally Fired Gas Turbine Using Biomass Gasifier-Combustor

HEAT EXCHANGER DESIGN

The heat exchanger (HEX) is the key to the success for this system. The higher temperature the HEX can provide, the higher the system efficiency. The main issue is how to build a HEX that can withstand the stresses caused by the working conditions, considering a reasonable building cost. Ceramic HEX can withstand much higher temperatures comparing with its metallic counterpart. However, the high developing cost makes this option not economical, at least with the current available ceramic fabrication methods.

Choosing the right type of HEX for the biomass based system is the most important step in the design. Therefore, the following points were considered during the HEX type selection:

- If the HEX passages are too narrow and tight, small particles, ash fraction and tar could block some passages. On the other hand, if the passages are too large, the gas flow speed will decrease causing a reduction in the over all heat transfer.
- The material used to build the HEX should has a reasonable oxidation resistance especially with high temperatures to get an acceptable metal lose rate due time.
- The HEX materials should be thick enough in order to withstand thermal stresses and high temperature up to 1100°C at the hot combustion gases inlet.

Stainless steel (SS) resists oxidation through selective oxidation with chromium that forms a barrier slows other oxide formation. Nickel content in the SS alloys often improves the metals stability at high temperature [13]. Therefore SS has been chosen in this project for the HEX.

The regenerator HEX type is not suitable for this case since there is a chance of mixing between the pure air and the exhausted gas, because of leakages or the gases trapped in the switching process. The plate-type HEX depends mostly on the small thicknesses of the plates to have a big surface area in a compacted size. This is not suitable for this case, because of the high temperature levels. Also, the extended surfaces are not desirable in this case because of the high temperatures, fouling and blocking problems.

The most suitable type of HEX for this case was found to be the shell-tube type. All the studies previewed in the introduction were proposing or using the shell-tube HEX type. Under shell and tube HEX, there are many different types as well, according to the flow arrangements. To choose the best HEX design, the following types have been studied:

- One pass counter flow.
- Two pass 1-2 TEMA E without baffles.
- Two pass 1-2 TEMA E with baffles.

The parallel flow type has not been considered due to the low efficiency. In order to compare between the above three types, Three design methods have been used in calculations, and result comparison has been made in order to check the design results, the methods are: (ε.C*.NTU) method, (P.R.NTU) method and MTD method. The calculation details have been recorded in the final report of the project [14].

Two pass 1-2 TEMA E with baffles HEX was found to be the most suitable type for this system. Two pipe sizes (½ and ¾ inches) were compared for this type and the ¾ inches size was found to be more suitable. The HEX tube bank contains 64 pipes (SS-304 type) of about 2.5mm wall thickness and 1.5 m in length with tube surface area of about $8.1 m^2$. Figure 3 shows the HEX during the inspection process after about 125 operation hours. The cold fluid (air) was in the tube side and the hot fluid (combustion gases) in the shell side.

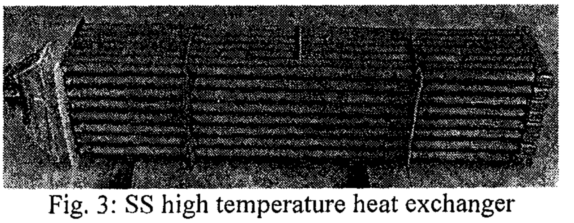

Fig. 3: SS high temperature heat exchanger

In order to determine the thermal performance of heat exchanger, effectiveness of the heat exchanger (ε) is calculated from Eq. 1:

$$\varepsilon = \frac{C_{air}(T_{air\ out} - T_{air\ in})}{C_{min}(T_{gas\ in} - T_{air\ in})} \times 100 \qquad (1)$$

Where C_{air} and C_{gas} are the heat capacity rate for air and gas respectively; C_{min} is the minimum heat capacity rate and it is equal to C_{air} in this design, $C_{air} > C_{gas}$ for this equation. The heat exchanger effectiveness of the design ε = 0.72. The heat capacity rate is calculated from Eq. 2:

$$C = \dot{m}.C_p \qquad (2)$$

Another equation (Eq. 3) was used to calculate effectiveness for $C_{gas} < C_{air}$.

$$\varepsilon = \frac{(T_{gas\ in} - T_{gas\ out})}{(T_{gas\ in} - T_{air\ in})} \times 100 \qquad (3)$$

Efficiency of the heat exchanger (η_{HEX}) is calculated from Eq. 4:

$$\eta_{HEX} = \frac{T_{\text{gas in}} - T_{\text{gas out}}}{T_{\text{gas in}} - T_{\text{ambient}}} \times 100 \qquad (4)$$

The HEX efficiency of about 62% and. This efficiency was acceptable in this prototype small scale system to keep the surface area and the fabrication cost in the acceptable limits. The air temperature at the turbine inlet was designed to be low of about 600°C for this prototype system to prevent any damages in the HEX tubes during the study of the EFGT concept. All the design parameters, power balance and efficiencies are shown in the power flow diagram (fig. 4).

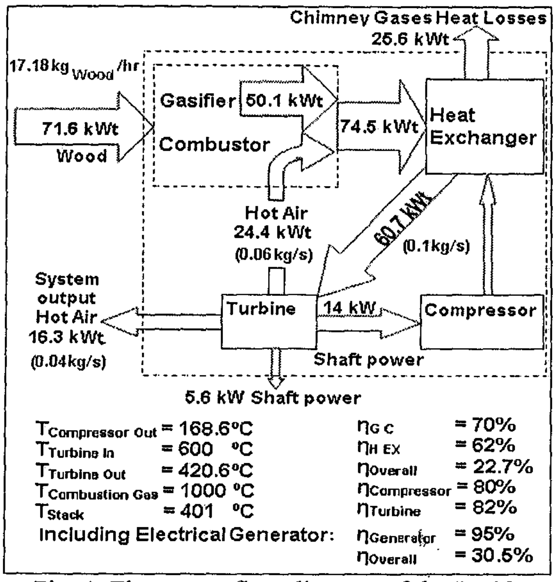

Fig. 4: The power flow diagram of the EFGT system

For the tube side, the following Nusselt number correlation [15] (Eq.5) was used:

$$Nu = 0.024\ Re^{0.8}\ .Pr^{0.4} \qquad (5)$$

Where (Re) is the Reynolds number, (Pr) is the Prandtl number. Nu = 19.442.

The heat transfer coefficient for the shell side (h_t) was calculated using Eq.6 :

$$h_t = Nu\ .K_a / di \qquad (6)$$

Where (K_a) is the thermal conductivity of air and (d_i) is the inner diameter of the tubes. h_t = 52.9 W/ m².K.

For the shell side, the following Nusselt number correlation [15] (Eq.7) was used:

$$Nu = 1.04\ Re^{0.4}\ Pr^{0.36}\,(Pr/Pr_w)^{0.25} \qquad (7)$$

Where (Pr_w) is the Prandtl number due to the wall temperature. Nu = 14.42.

The heat transfer coefficient for the shell side (h_s) was calculated using Eq.8:

$$h_s = [Nu\ .K_g / d_o](\mu_w / \mu_m)^{-0.14} \qquad (8)$$

Where (K_g) is the thermal conductivity of combustion gases and (d_o) is the external diameter of the tubes. The kinematics viscosity ratio is assumed to be ($\mu_w / \mu_m = 1$) for gases h_s = 35.63 W/ m².K.

The over all resistant was calcualtad using Eq. 9:

$$1/U = 1/h_g + Rf_g + d_o \ln(d_o/d_i)/2.K_w + Rf_a(d_o/d_i) + (d_o/d_i)/h_a \qquad (9)$$

Where (K_w) is the thermal conductivity of the tube wall and (Rf_g, Rf_a) are the fouling resistant for the gas and air sides respectively. Therefore, the overall heat transfer coefficient U = 19 W/ m².K.

The overall pressure drop for the tube and shell sides was found to be low of about 107 Pa and 222 Pa respectively.

RESULTS AND DISCUSSION

The HEX design calculations were based on the steady state operation conditions with 600°C, 0.1kg/s and 2bar, that is the turbine inlet temperature, air flow rate and air pressure (gauge pressure) respectively, at the turbine self running mode. However, during the experiment, the turbine didn't reach the self operation since the startup air blower was not supplying enough air pressure for the startup. In order to compare the theoretical and experimental performance of the HEX at the same steady state conditions except for the air pressure during the experiment (around 0.65bar gauge pressure), the difference in the air power inside the HEX between the two cases should be taken in to the consideration. The air power inside the HEX should reach theoretically up to 60kWt at the turbine self operation mode with 62% and 72% HEX efficiency and effectiveness respectively. However, the experimental results showed only 50kWt air power inside the HEX. Also, the air blower power (13kWe), supporting the turbine, was included in the power balance calculations, the HEX efficiency was low of about 50% and didn't provide a good indication on the HEX performance since the efficiency equation is including ($T_{ambient}$) instead of ($T_{air\ inlet}$) in equation (4). The effectiveness can provide better indication on the HEX performance. The experimental HEX effectiveness at the steady state point was about 62.5%.

The heat capacity rate for combustion gases (C_{gas}) was designed to be more than the heat capacity for air (C_{air}). However, in the experiments, it was noted that $C_{gas} < C_{air}$ in the beginning. But, with more air temperature and turbine power, the speed of the turbine shaft starts to increase and the compressor side of the turbocharger stars to supply more air flow rates up to 0.1kg/s more than the air blower can supply and C_{gas} became more than C_{air} as designed. Figure 5 shows the relation between the air flow increment and the air temperature at the HEX outlet.

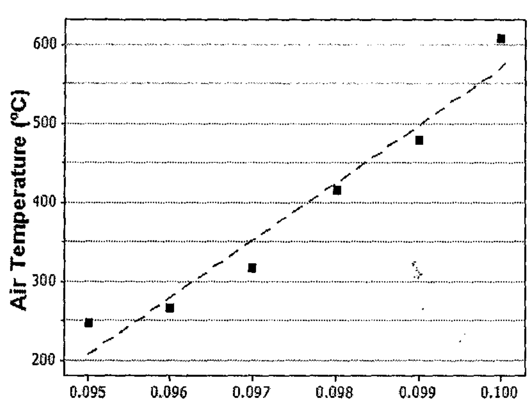

Fig. 5: Average Air Temperature with Different Air Flow Rates

Figure 6 shows the changes in the effectiveness against the air temperature at the HEX outlet. In the beginning, the difference between ($T_{gas\ in}$) and ($T_{air\ in}$) started to increase causing a drop in the effectiveness. However, at high temperature ($T_{air\ out}$ = 500 °C), the heat capacity $C_{gas} > C_{air}$ and the effectiveness values started to stabilize when the rate of the air temperature rise became equal or even more than the increasing rate in gas temperature.

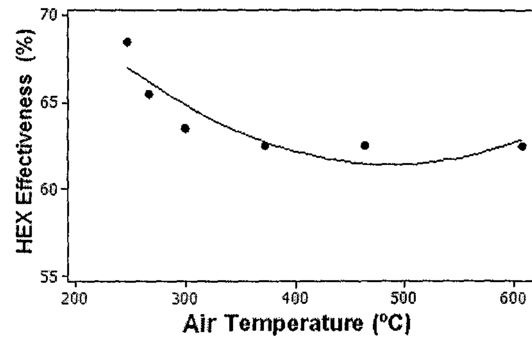

Fig. 6: Effectiviness of HEX with Different Air Temperatures

Figure 7 shows the effect of the air flow rate on the heat exchanger performance in the heating up period. The air and gas flow rates can affect the effectiveness of the heat exchanger indirectly by increasing the overall heat transfer coefficient of the heat exchanger.

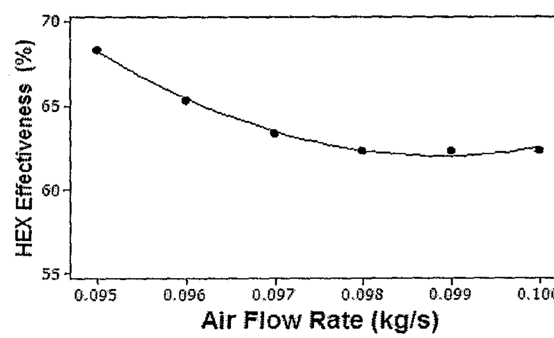

Fig. 7: Effectiviness of HEX with Different Air Flow Rates

Figure 8 shows the temperature distribution on the combustion chamber and heat exchanger at maximum air temperature (694°C), there is no crossing between the temperature lines of combustor and heat exchanger, which means, no air cooling process at any part of the heat exchanger.

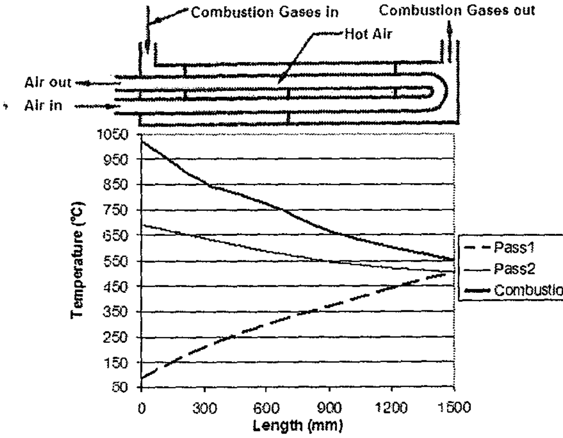

Fig. 8: Temperature Distripution on HEX at Maximum Temperature

CONCLUSION

The shell-tube SS heat exchanger was found to be the most suitable type for the gasifier-combustor EFGT system. The experiments showed that heat exchanger can increase the air temperature at the turbine inlet up to 694°C that is more than the designed temperature, with effectiveness of about 62%. The air pressure inside the heat exchanger was not enough for the micro turbine engine to operate in the self running mode. High flow compressor or high speed motor can be connected to the compressor side of the turbocharger to provide enough air flow and pressure for the turbine startup.

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CHAPTER 4

METHODOLOGY

4.0 Introduction

This research depended mainly on the experimental work, to investigate the externally fired micro gas turbine concept using off cut furniture wood as biomass fuel, and to study the performance of a hot air production system with and without the turbocharger using the same biomass fuel as shown in the flow diagram of the study methodology in figure 4.1.

In this chapter the following issues will be discussed:

- Design and fabrication of the EFGT system.
- Fabrication of hot air production unit without the turbocharger.
- Experiment setup.
- Experiment procedures.

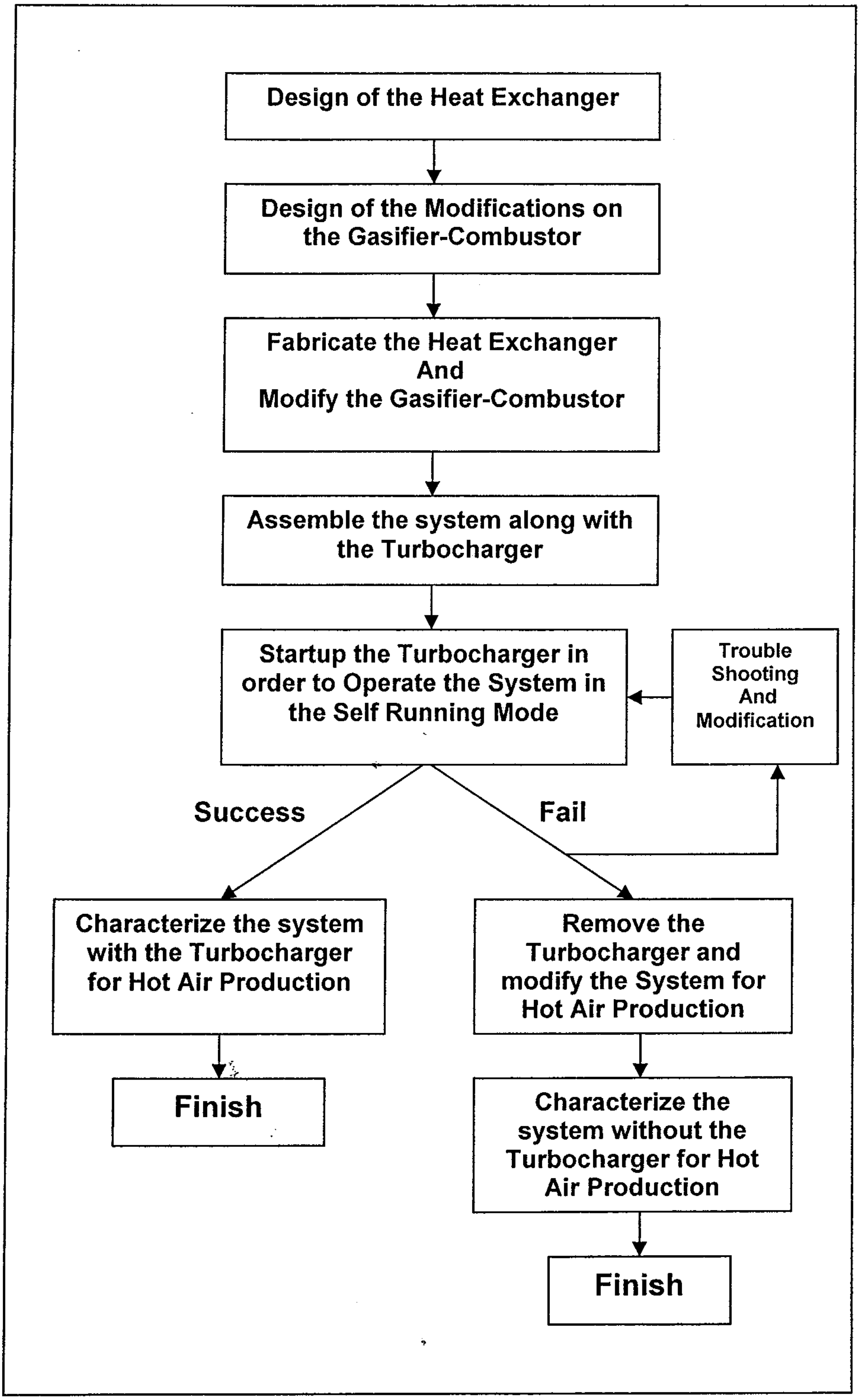


Figure 4.1: Flow Diagram of the Study Methodology

4.1 Design and Fabrication of the EFGT System

The main topics which will be discussed under the design and fabrication of the system are:

- System layout.
- Gasification unit and combustion chamber.
- Modification of the gasifier-combustor unit.
- High temperature heat exchanger.
- Turbocharger.
- Lubrication system.

4.1.1 System layout

In this project, a small scale EFGT unit has been developed for hot air production for industrial drying processes. In future studies, a power generation unit will be added to the system to produce both hot air and electrical power for more efficient system. Figure 4.2 shows the layout of the hot air production system. The producer gas from the gasifier was totally combusted in the combustion chamber. Combustion thermal power was transferred through the high temperature heat exchanger to the turbine working fluid (air) that expands through the turbine. In the experiments, some of the hot air stream after the turbine was released to the ambient as the hot air output of the system. The rest of the hot air was returned back to the combustion chamber as a thermal power and also an oxygen source for the combustion process. The turbine startup process used an air blower

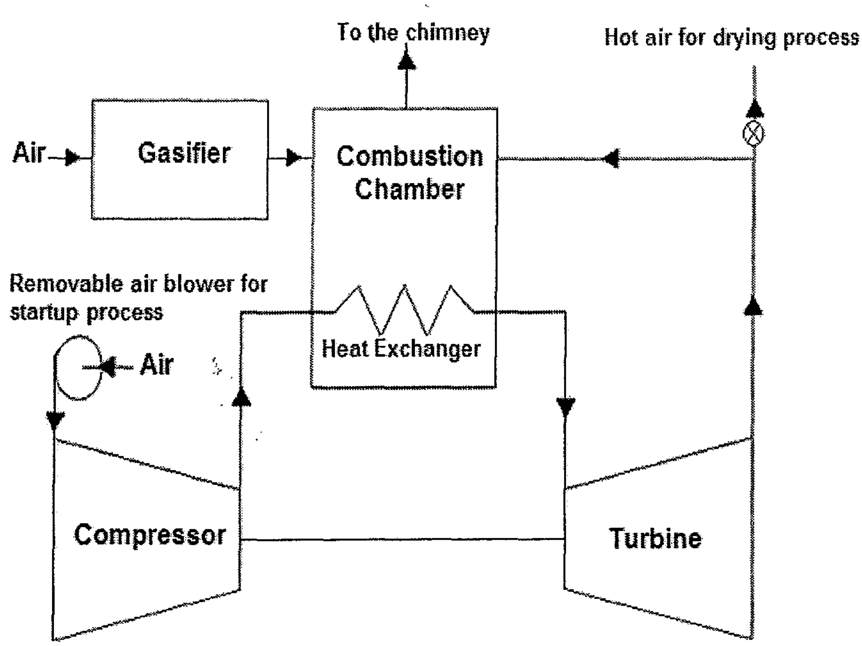


Figure 4.2: Layout of the EFGT System for Hot Air Production

Some essential points were considered in the design as:

I. Small floor area, so it can be transported easily through the off-roads to the rural places and for the industrial drying it is always desirable to have small floor area equipments.
II. Simple construction, so the turbocharger engine can be easily removed for maintenance. Also, the heat exchanger should be removable, for any repair.
III. Reasonable building cost for the unit, so it can be more convenient from the economical point of view for the industrial field, with low pay back time.
IV. Low CO emissions from the exhaust of the unit

Figure 4.3 and plate 4.1 show the schematic drawing and the description of the different parts of the EFGT system respectively.

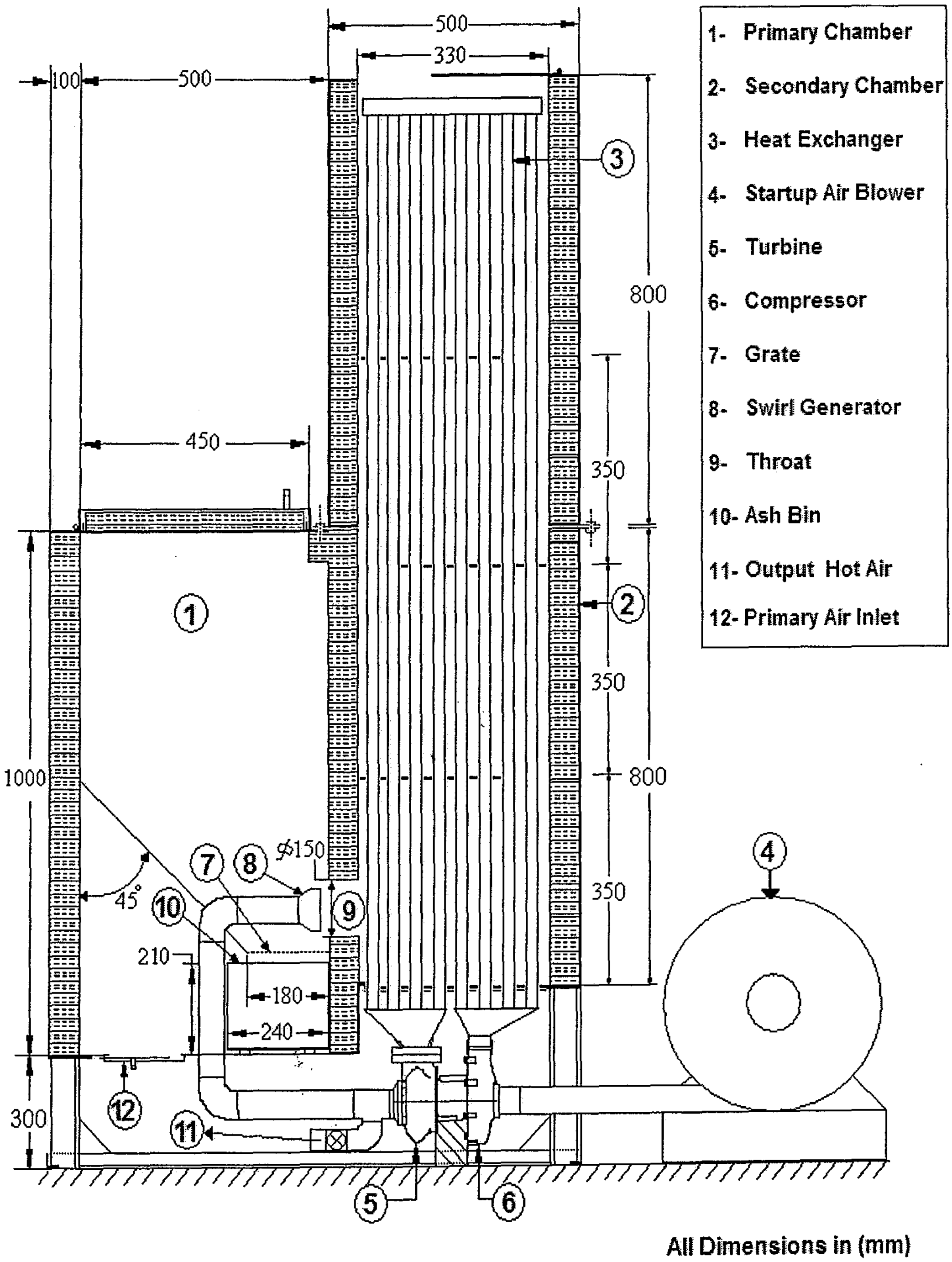
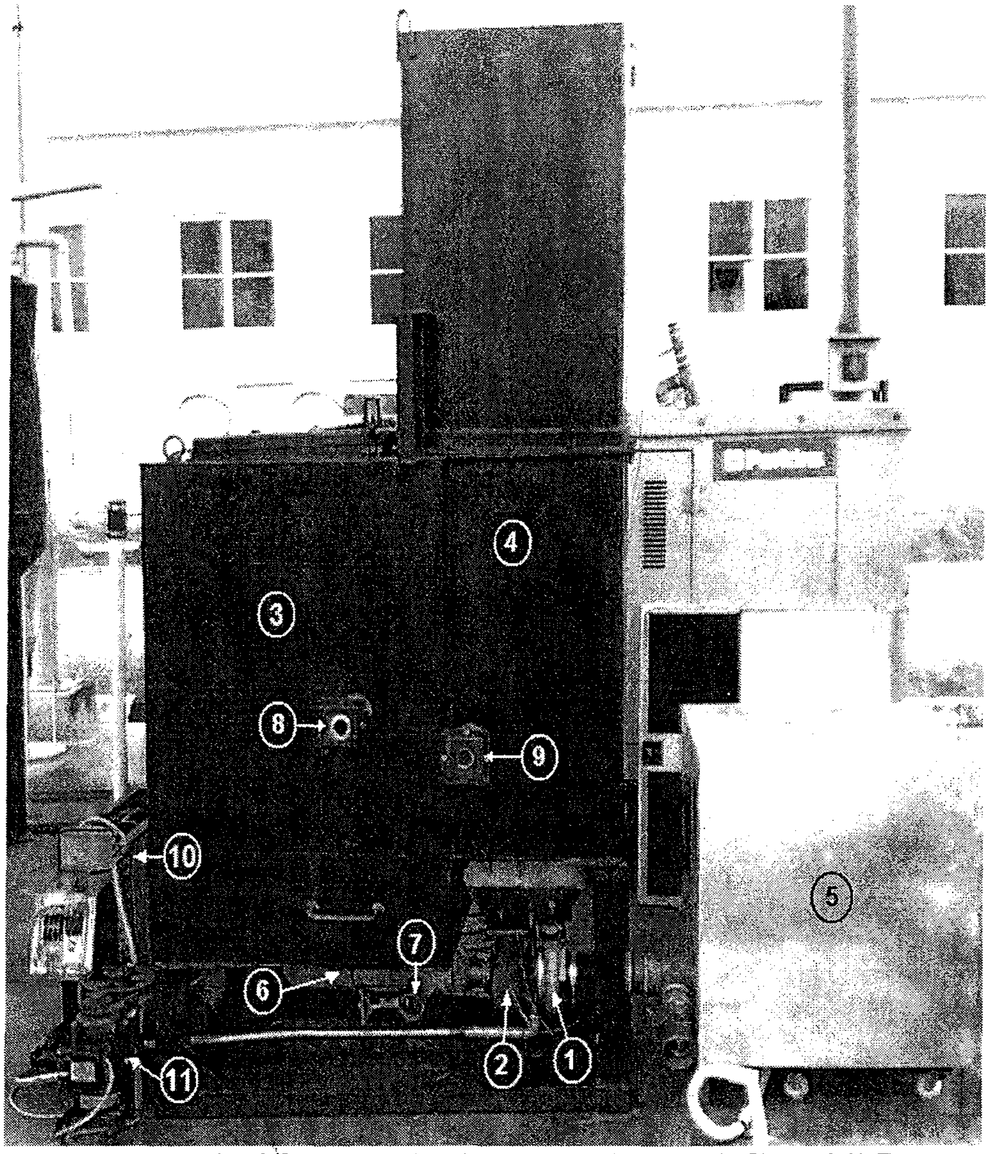


Figure 4.3: Schematic Drawing of the Biomass EFGT System



1-Compressor 2-Turbine 3-Gasifier Chamber 4-Combustion Chamber 5-Air Blower 6-Air For Combustion 7-Output Hot Air Valve 8-Primary Chamber Sight Glass 9-Secondary Chamber Sight Glass 10-Oil Cooling Radiator 11-Lubrication Oil Pump

Plate 4.1: Biomass EFGT System at the School of Mechanical Engineering, USM

4.1.2 Gasification Unit and Combustion Chamber

Two options were available for the gasification unit and the combustion chamber:

1) Using a downdraft gasifier with a separate combustion chamber. In this option, a separate combustion chamber should be designed, which gives more freedom in the design of the high temperature heat exchanger.
2) Using a gasifier-combustor, which is a gasifier combined with a combustion chamber in one unit.

Both downdraft gasifier and gasifier-combustor are available in the school of mechanical engineering, Universiti Sains Malaysia. The first option will occupy a large floor area comparing with the second one that is compact in size and can occupy just 1.1 x 0.7 m^2 (floor area) with the heat exchanger and turbocharger, also in the first option, an extra air blower will be used for the gasifier. The combustion chamber geometry in the second option is fixed, therefore the only variables in the heat exchanger design are size, number and length of the tubes. The available geometry was suitable for a good design, therefore the limitation in design options was not a significant factor to reject the gasifier-combustor option.

The gasifier-combustor has been chosen to provide thermal power for the externally firing of the micro turbine engine that is a truck turbocharger in this project, without any drying or cooling systems along with the gasifier. The gasifier-combustor was developed in a previous project for waste biomass combustion. It is a multi-chamber combustor with 2 stages namely, gasifier chamber and combustor chamber. The gasifier chamber dimensions are ($0.6\times0.5m^2$) with a height of (1m), and the combustor chamber is (0.5×0.4m2) and the height is (0.8m). Both chambers are made of a refractory cement layer of around (10cm) thickness and covered with a mild steel sheet of (2mm) thickness. Biomass loading port is placed on the top of the gasifier chamber to load the wood into the gasifier, and it has a fit clamped cover to avoid the produced gas from flowing out of the gasifier.

The distribution of the different expected zones inside the gasifier-combustor is estimated as shown in figure 4.4.

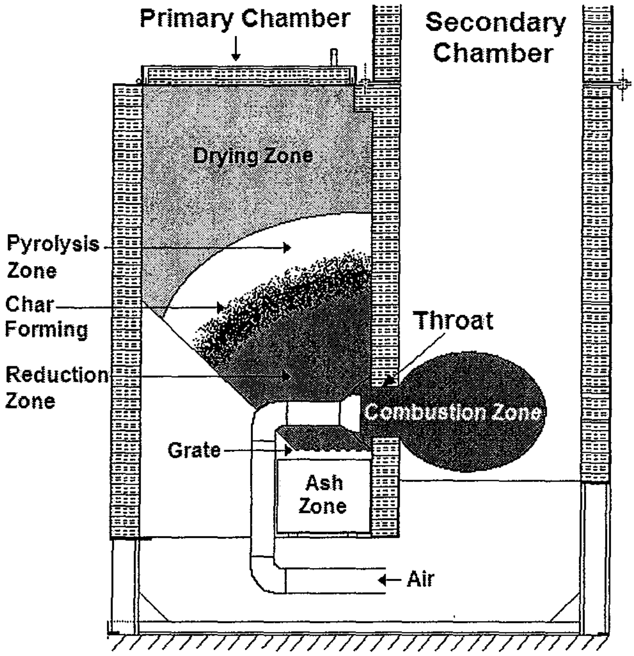


Figure 4.4: Sketch of the Expected Different Zones inside the Gasifier-Combustor

The cone structure in gasifier chamber is to provide a smooth gravitational movement of the wood to the combustion zone. The gasifier-combustor has a throat between the two chambers. It is a single flame gasifier-combustor so the flame in the combustion zone extends horizontally through the throat to the secondary combustion chamber. Therefore, no cold places in the sides of the throat, and that gives more chance for proper tar cracking while the gases are passing through the throat.

4.1.3 Modifications of the Gasifier-Combustor Unit

Some modifications were done on the gasifier-combustor to be suitable for the system, figure 4.5 shows the gasifier-combustor before and after the modification.

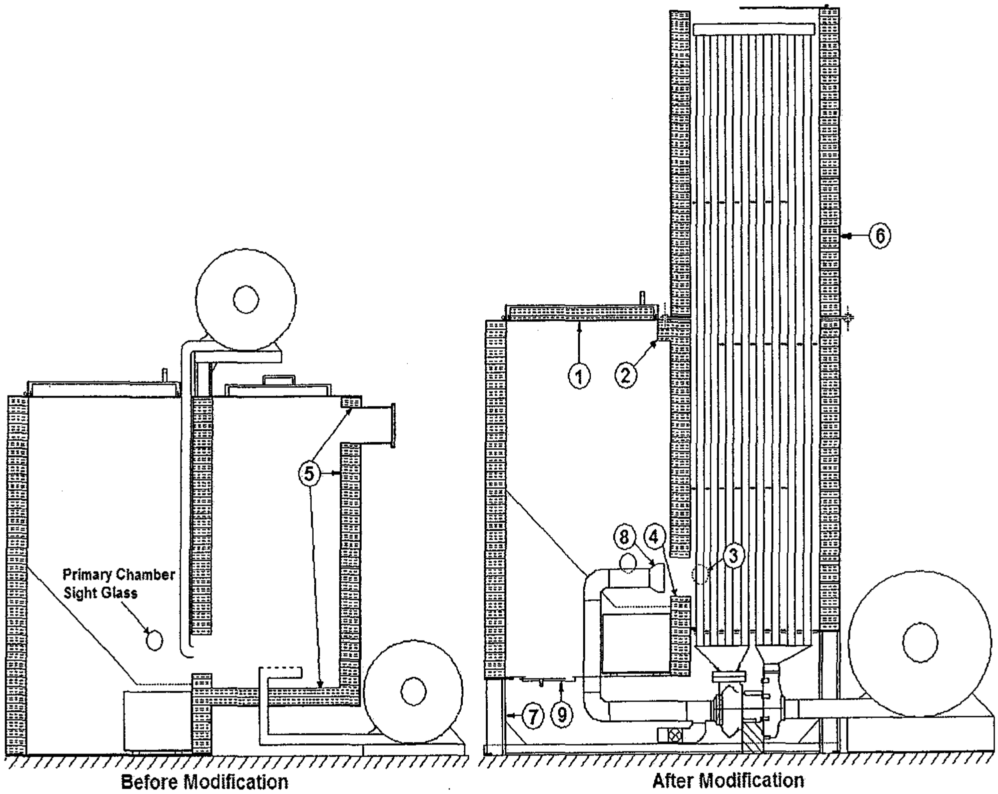


Figure 4.5: Gasifier-Combustor before and after the Modification

The numbers shown in figure 4.5 refer to the modifications in the following paragraphs, the modifications are:

1. Insulating the cover of the gasifier chamber with a (5 cm) layer refractory cement to reduce the thermal loses from the chamber.

2. The top plate of the primary chamber in the front side bended due to the high temperature and as a result of that, the cover was not closing fit and the gases were leaking. The plate was insulated using refractory cement layer of (10 cm) thickness a long the front side of the cover.

3. A sight glass has been added in the side of the combustion chamber right after the throat to allow a visual inspection of the flame and the heat exchanger in the hot zone (see Plate 4.2).

4. The throat between the two chambers had a square shape of ($15x15$ cm^2), and to get a uniform high temperature in the all sides of the throat for better tar cracking, the throat has been modified to a circular shape of (15 cm^2) diameter.

5. The combustion chamber should be exactly (33 x 33 cm^2) in size to place the heat exchanger inside it. Therefore some breaking work

on the chamber's refractory cement walls was done to get the exact size, Also, the bottom of the chamber was opened by cutting the mild steel sheet and breaking the refractory cement layer to enable the heat exchanger to pass through the chamber. The exhaust pipe of the combustion chamber was removed and the hole was sealed with refractory cement.

6. The height of the combustion chamber was just (0.8m). Therefore, an additional extension for the chamber of (0.8m) height has been fabricated. The extension was made of refractory cement layer, covered with mild steel sheet of (2mm) thickness with a stainless steel cover in the top of ($0.35x0.22m^2$) as the upper baffle of the heat exchanger. The inner and outer dimensions of the chamber are ($0.33 \times 0.33m^2$) and ($0.5 \times 0.5m^2$) respectively, with three thermocouple connection ports on the side.

7. In order to fix the hot air pipes and the turbocharger in the downside of the combustion chamber, and also to give enough space to repair or remove the turbocharger easily, a metallic stand has been built to lift the gasifier-combustor (0.3m) above the ground.

8. In order to increase the mixing efficiency between air and producer gas to get a complete combustion, a swirl generator with 45° vanes has been fabricated and fixed on the air inlet pipe inside the chamber (see Plate 4.3). A detailed drawing of the swirl generator is shown in appendix **A**.

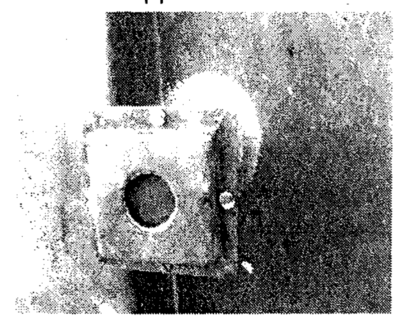


Plate 4.2: The Sight Glass on the Combustion Chamber

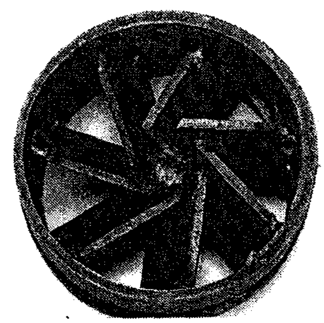


Plate 4.3: The Swirl Generator

9. A complete combustion in the combustion chamber creates a very hot zone that pushes the gases upward and creates a stack effect that creates a low pressure in the gasifier chamber. Therefore, a controlled air intake in the bottom of the gasifier chamber has been cut with a sliding gate to enable a controlled amount of air to be induced in to the chamber to complete the gasification process.

The design and modification drawings are presented in (appendix **A**). Plate (4.4) shows the bottom of the gasifier-combustor during the modification process, the bottom of the secondary chamber has been opened, as well as the air intake in the bottom of the gasifier chamber. The stand has not been fixed yet.

Plate (4.5) shows the top of the gasifier-combustor during the modification process. The cover of the gasifier chamber and the top sheet in the front side of the cover, both have been removed for insulating process. The cover of the combustion chamber has been removed as well to prepare the top side to be flanged with the extension of the chamber.

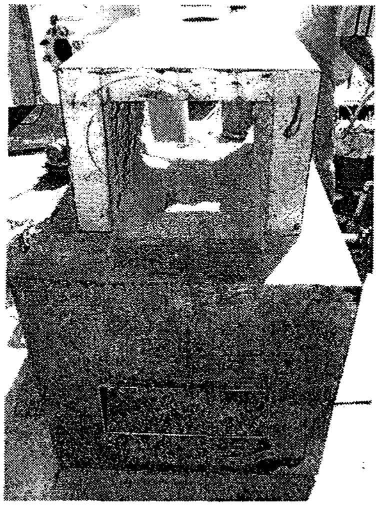


Plate 4.4: The bottom of the Gasifier-Combustor during the Modification

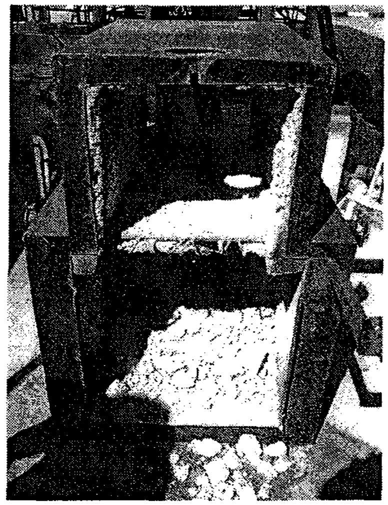


Plate 4.5: The top of the gasifier-Combustor during the Modification

4.1.4 High Temperature Heat Exchanger

The heat exchanger is the key to the success of this project. The higher temperature the heat exchanger can provide, the higher the system efficiency. The main issue is how to build a heat exchanger that can withstand the stresses caused by the working conditions, considering a reasonable building cost. Building a ceramic high temperature heat exchanger has not been taken in to our consideration because of the very high fabrication cost.

Different types of heat exchangers have been discussed earlier in chapter 3. Shell and tube type has been found to be the most suitable type for the project. Under shell and tube heat exchangers, there are many different types as well, according to the flow arrangements. To choose the best heat exchanger design, the following types have been studied:

- One pass counter flow.
- Two pass 1-2 TEMA E without baffles.
- Two pass 1-2 TEMA E with baffles.

The parallel flow type has not been considered due to the low efficiency. The materials and fabrication cost of the high temperature heat exchanger is high. Therefore, it has been decided to reduce the size of the system for this investigation. Eventhough, the small scale will have lower efficiency comparing with bigger scales. However, the fabricated prototype can provide sufficient information about the system for any further scale up study.

Figure 4.6 shows the actual Brayton cycle drawing for the turbocharger engine. The actual Brayton cycle parameters were calculated using the Engineering Equation Solver EES (limited academic version 6.190),

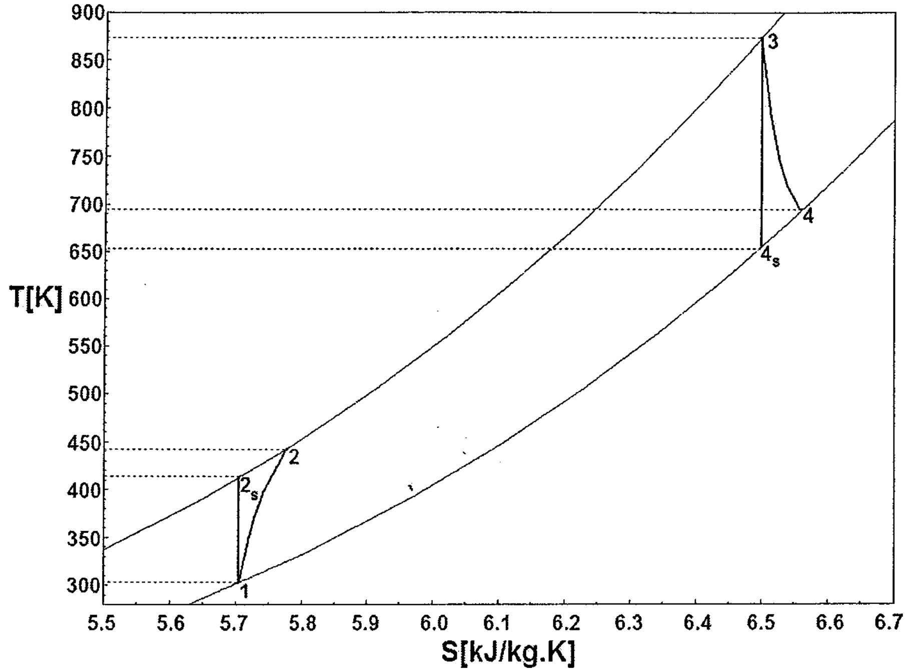


Figure 4.6: Actual Brayton Cycle of the Design for the Turbocharger Engine (Engineering Equation Solver version 6.190)

The turbine inlet temperature was assumed to be low of about 600°C. Therefore, the turbocharger was expected to run at low speed and the air mass flow rate from the compressor was assumed to be low of about (0.1) kg/s. The ideal air temperature in the exit of the compressor and inlet of the heat exchanger can be calculated using the ideal gas properties table of air (table B-1) in appendix **B**. Assuming the compressor air inlet temperature T_1=30°C, the relative pressure of this temperature is P_{r1} = 1.43558 (dimensionless quantity), and assuming the compressor exit pressure P_2 = 3 bar (absolute), the relative pressure for the compressor exit is calculated using equation 4.1:

$$p_{r2} = p_{r1}\left(\frac{P_2}{P_1}\right) \qquad 4.1$$

P_{r2} = 4.30674, and from the ideal gas properties table of the air (table B-1), the ideal air temperature at the outlet of compressor T_2s = 141°C. For the compressor side of turbocharger in Tariq et al., (1995), the efficiency was 80%. Therefore, same compressor efficiency was assumed for this design. The actual temperature was calculated using the EES program. T_2 = 178°C, it is also the air temperature in the inlet of heat exchanger $T_{air,in}$. The EES program calculation results of the Brayton cycle will be presented in section (4.1.5).

In the beginning, thermal output power of the gasifier was assumed to be Q = 70 kWt, and after calculating the complete combustion product flow rate, the power will be recalculated again. Assuming the efficiency of the gasifier is about 70%, the energy content for wood (20% moisture content) Cv_{wood} = 15 MJ/kg (Zainal, 1996). The biomass fuel consumption can be calculated using equation 4.2:

$$Q = \dot{m}_w \cdot Cv_{wood} \qquad 4.2$$

Wood mass flow rate will be around $\dot{m}_w$ = 24 kg_{wood}/hr. To calculate theoretically the mass flow rate of the producer gas out of the gasification chamber, for heating value Cv_{pg} = 4 MJ/kg (Zainal, 1996), from the power balance equation 4.3:

$$Q = \dot{m}_{pg} \cdot Cv_{pg} \qquad 4.3$$

The mass flow rate of the producer gas is around $\dot{m}_{pg}$ = 0.0175 kg/s. The air required for the gasification process is induced by the stack effect from the bottom of the primary chamber. For the complete combustion of the producer gas, Part of the hot air after the turbine is returned back to the chamber and the other part is released out of the system as a thermal power output as in equation 4.4:

$$\dot{m}_a = \dot{m}_{ca} + \dot{m}_{output} \qquad 4.4$$

$\dot{m}_a$ = 0.1 kg/s (assumed earlier). The amount of air required for complete combustion of (1 kg) wood is 6.27kg air, as mentioned in chapter 3. Since the wood consumption is 24 kg_{wood}/hr. Therefore, the air supply to the primary chamber will be $\dot{m}_{ca}$ = 0.0425 kg/s. Using equation 4.4, the power output air flow rate $\dot{m}_{output}$ = 0.04 kg/s. The mass flow rate of the combustion product is calculated from equation 4.5:

$$\dot{m}_{cg} = \dot{m}_{pg} + \dot{m}_{ca} \qquad 4.5$$

The combustion gases flow rate $\dot{m}_{cg}$ = 0.06 kg/s. Assuming the combustion gases inlet temperature is $T_{g,i}$ = 1000°C, and the air outlet temperature is $T_{a,o}$ = 600°C, All the air properties were taken from table (B-4) in appendix **B** at 650K, Cp_{air} = 1.066 kJ/kg.K. The combustion product heat capacity is calculated in appendix **B** at 900K, Cp_{cg} = 1.28 kJ/kg.K. The power transferred to the air through the heat exchanger is calculated using the power balance equation 4.6:

$$Q_{HE} = \dot{m}_{cg} \cdot Cp_{gas} (T_{g,i} - T_{g,o}) = \dot{m}_{a} \cdot Cp_{air} (T_{a,o} - T_{a,i}) \qquad 4.6$$

The power is around Q_{HE} = 45.98 kW. The same power was transferred from the combustion gases, therefore the combustion gases outlet temperature after the heat exchanger can be calculated using the same equation 4.6, and therefore the combustion gases outlet temperature will be around $T_{g,o}$ = 401°C.

Assuming the ambient temperature is 30°C, the thermal output power of the gasifier was assumed earlier to be 70 kW. Equation 4.7 is used to calculate the accurate power value.

$$Q_{GC} = \dot{m}_{cg} \cdot Cp_{cg} (T_{g,i} - 30) \qquad 4.7$$

Q_{GC} = 71.8 kWt. The wood consumption and the efficiency of the system will be calculated in section (4.1.5). The Efficiency of the heat exchanger that is the ratio between the power transferred through the heat exchanger and the power contained in the hot fluid, is calculated using equation 4.8:

$$\eta_{HE} = \frac{Q_{HE}}{Q_{GC}} \times 100 \qquad 4.8$$

4.1.4.1 Heat Exchangers Design Methods

Three design methods have been used in calculations, and result comparison has been made in order to check the design results, the methods are:

- (ε.C*.NTU) method.
- (P.R.NTU) method.
- MTD method.

In the beginning, some parameters have to be calculated for each method.

4.1.4.1.1 First Method: (ε.C*.NTU)

NTU is the number of transfer units. The heat capacity rate for air C_a and combustion gases C_{cg} both can be calculated using the general equation 4.9:

$$C = \dot{m} \cdot Cp \qquad 4.9$$

C_{air} = 106.6 W/K; C_{cg} = 76.8 W/K. The heat capacity rate for the combustion gases is less than air, so $C_{air} = C_{max}$, and $C_{cg} = C_{min}$. The heat capacity ratio C* can be calculated using equation 4.10:

$$C^* = \frac{C_{min}}{C_{max}} \qquad 4.10$$

C* = 0.72; the maximum temperature difference (ΔT_{max}) can be calculated using equation 4.11:

$$\Delta T_{max} = T_{g,in} - T_{a,in} \qquad 4.11$$

ΔT_{max} = 831°C, since $T_{a,in}$= 168.6°C as mentioned earlier. In order to measure the thermal performance of the heat exchanger, effectiveness (ε) is used. It is defined as the ratio of the actual heat transfer rate through the heat exchanger to the maximum possible heat transfer rate. It is calculated using equation 4.12:

$$\varepsilon = \frac{Q}{Q_{max}} = \frac{C_g(\Delta T_g)}{C_{min}(\Delta T_{max})} = \frac{C_a(\Delta T_a)}{C_{min}(\Delta T_{max})} \qquad 4.12$$

ε = 0.72.

4.1.4.1.2 Second Method: (P.R.NTU)

Assuming fluid 1 is air and fluid 2 is gas, the temperature effectiveness for air (P1) can be calculated using equation 4.13:

$$P_1 = \frac{T_{a,o} - T_{a,i}}{\Delta T_{max}} \qquad 4.13$$

P_1= 0.52, the heat capacity rate ratio (R_1) can be calculated using equation 4.14:

$$R_1 = \frac{C_a}{C_g} \qquad 4.14$$

R_1 = 1.39; the heat exchanger surface area (A) is calculated using equation 4.15:

$$A = \frac{C_{min}.NTU}{U} \qquad 4.15$$

4.1.4.1.3 Third Method: MTD

This is the mean temperature difference method. The log-mean temperature difference (LMTD=ΔT_{lm}). It is calculated using equation 4.16 for the cross flow arrangement:

$$\Delta T_{\mathrm{lm}} = \frac{\Delta T_I - \Delta T_{II}}{\ln\left(\Delta T_I / \Delta T_{II} \right)} \qquad 4.16$$

ΔT_I and ΔT_{II} are the temperature difference at each end, and can be calculated from equations 4.17 and 4.18 respectively:

$$\Delta T_I = T_{g,in} - T_{a,out} \qquad 4.17$$

$$\Delta T_{II} = T_{g,out} - T_{a,in} \qquad 4.18$$

LMTD = 309°C; the surface area (A) of the heat exchanger is calculated using equation 4.19:

$$A = Q / U.F.\Delta T_{\mathrm{lm}} \qquad 4.19$$

Where (F) is the mean temperature difference correction factor.

4.1.4.2 Primary Selection of the Heat Exchanger Type

For a primary comparison between different heat exchanger designs, the overall conductivity of the heat exchanger was estimated from (table **C** -1) in appendix **C**, for low pressure gases (both hot and cold fluids), the overall heat transfer coefficient U = 55 W/m².K. This value is practically high compared to the actual values that will be calculated in section (4.1.4.3). Therefore, the results are just for comparison between the different type of designs.

4.1.4.2.1 First Design

The one pass counter flow schematic drawing is shown in figure 4.7.

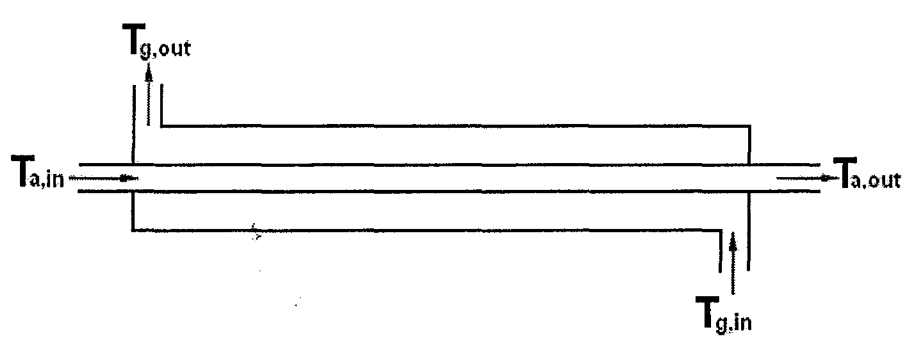


Figure 4.7: One Pass Counter Flow Heat Exchanger

Using MTD method, the correction factor (F = 1) for the cross flow arrangement. The surface area that is required to transfer 45.98 kWt, is calculated using equation 4.19; the surface area A = 1.43m². The heat exchanger geometry parameters for different pipe sizes can be calculated using equation 4.20:

$$A = L.Nt.\pi.d_o \qquad 4.20$$

Where Nt is the total number of pipes; L is the total length of the heat exchanger; do is the external pipe diameter. The distance between two pipes (Pt) depends on the number and size of pipes, since combustion chamber cross section area is fixed (330x330 mm^2). The parameters are shown in table 4.1 for different type (size) of the stainless steel pipes available in the market.

Table 4.1: First Design

Pipe type	do(mm)	Pt(mm)	Nt	L(m)
1"	36	46	36	0.35
¾"	27	37	64	0.26
½"	22	32	100	0.21

4.1.4.2.2 Second Design

The two pass 1-2 TEMA E tube-shell heat exchanger with out baffles design schematic drawing is shown in figure 4.8. The desirable arrangement is (A) since there is no cross between the combustion gas and the second air pass plots, and that mean there are no air cooling occurs along the heat exchanger.

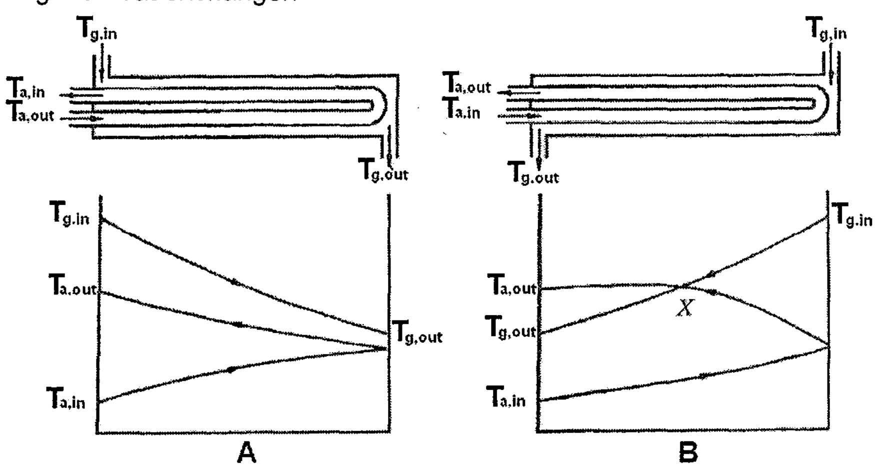


Figure 4.8: Two Pass 1-2 TEMA E Tube-Shell Heat Exchanger without Baffles

Using (P.R.NTU) method, the NTU value can be calculated from the following equation 4.21 for fluid 1 unmixed (air), fluid 2 mixed (gas):

$$P_1 = \frac{1-\exp(-K.R_1)}{R_1} \qquad 4.21$$

Where K is calculated using equation 4.22:

$$K = 1-\exp(-NTU) \qquad 4.22$$

P_1=0.52; R_1=1.39 (calculated earlier). Therefore, NTU can be calculated using trial and error method; NTU = 2.5. In order to check the NTU value, the (ε.C*.NTU) method was used, the NTU value can be found from the ε.C*.NTU (chart **C**-3) in appendix **C**; NTU = 2.5. The heat transfer surface area can be calculated using equation 4.15; the surface area is A = 3.49 m². The heat exchanger geometry parameters for different pipe sizes can be calculated using equation 4.20, since the surface area is known. The number of tubes per pass (Np) is half the number of the pipe, since the two pipe pass design have been chosen. The distance between two pipes (Pt) depends on the number and size of pipes, as mentioned in the First design, section (4.1.4.2.1). The parameters are shown in table 4.2.

Table 4.2: Second Design

Pipe type	do(mm)	Pt(mm)	Nt	Np	L(m)
1"	36	46	36	18	0.85
¾"	27	37	64	32	0.64
½"	22	32	100	50	0.51

4.1.4.2.3 Third Design

The two pass 1-2 TEMA E tube-shell cross flow with baffles schematic drawing is shown in figure 4.9.

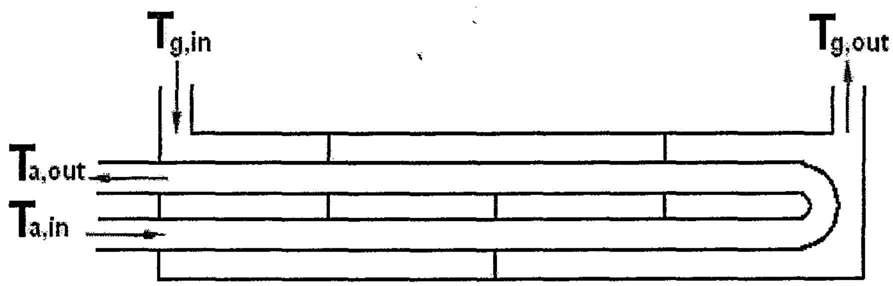


Figure 4.9: Two Pass 1-2 TEMA E Tube-Shell Cross Flow Heat Exchanger with Baffles

Using (P.R.NTU) method, the NTU value can be found from the P1.R1.NTU (chart **C**-2) cross flow unmixed/ unmixed in appendix **C**. NTU = 2.1; surface area can be calculated using equation 4.15; A = 2.932 m².

The heat exchanger geometry parameters for different pipe sizes can be calculated using equation 4.20, since the surface area is known. The parameters were explained in the second design, section (4.1.4.2.2). The parameters are shown in table 4.3.

Table 4.3: Third Design

Pipe type	do(mm)	Pt(mm)	Nt	Np	L(m)
1"	36	46	36	18	0.72
¾"	27	37	64	32	0.54
½"	22	32	100	50	0.42

The surface area was found to be in the first second and third designs, 1.43 m^2, 3.49 m^2 and 2.93 m^2 respectively. The first design has the minimum surface area, however it was rejected because it has only one pass, and the compressor is coupled with the turbine in one set, therefore a returning pass outside the chamber should be added, That will cause a pressure and temperature losses. The second and third design will be compared

4.1.4.3 Secondary Selection of the Heat Exchanger Type

In this stage of the design, the overall conductivity of the heat exchanger will be calculated more accurately with the surface area for the following designs:

- Two pass 1-2 TEMA E tube-shell cross flow with baffles heat exchanger.
- Two pass 1-2 TEMA E tube-shell without baffles heat exchanger

The calculations will include two size of pipes ¾" and ½".

4.1.4.3.1 Two Pass 1-2 TEMA E Tube-Shell Cross Flow With Baffles

The detailed calculations of the design are presented in appendix **D**. For the first pipe size (¾"), the heat transfer coefficient for the tube side was h_t= 52.9 W/m^2.K, and for the shell side h_s= 35.6 W/m^2.K. The overall heat transfer coefficient was U= 19 W/m^2.K. The required heat transfer area was around 8.1 m^2, and since the number of pipes is (64) pipe, the height of the heat exchanger is 1.5 m.

For the second pipe size (½"), the heat transfer coefficient for the tube side was h_t= 60.6 W/m^2.K, and for the shell side h_s= 43.47 W/m^2.K. The overall heat transfer coefficient was U= 21.76 W/m^2.K. The required heat transfer area was around 7.1 m^2, and since the number of pipes is (100) pipe, the height of the heat exchanger is 1.03 m.

4.1.4.3.2 Two Pass 1-2 TEMA E Tube-Shell Heat Exchanger without Baffles

The detailed calculations of the design are presented in appendix **E**. For the first pipe size (¾"), the heat transfer coefficient for the tube side was h_t= 53 W/m^2.K,and for the shell side h_s= 8.4 W/m^2.K. The overall heat transfer coefficient was U= 7 W/m^2.K. The required heat transfer area is around 28.6 m², and since the number of pipes is (64) pipe, the height of the heat exchanger is 5.2 m.

For the second pipe size (½"), the heat transfer coefficient for the tube side was h_t= 60.6 W/m^2.K, and for the shell side h_s= 9 W/m^2.K. The overall heat transfer coefficient was U= 7.5 W/m^2.K. The required heat transfer area is around 26.7 m², and since the number of pipes is (100) pipe, the height of the heat exchanger is 3.8 m.

For the second design (without baffles), the small gas flow rate (0.06kg/s) passing parallel with the tubes produced a very low heat transfer coefficient for the shell side. Therefore, the overall heat transfer coefficient was also low, and that required a higher heat exchanger. This was not acceptable due to the high cost of the heat exchanger's materials (stainless steel) and also the fabricating cost of a very high isolated combustion chamber. Therefore, the second design was rejected and the only design left is the Two pass 1-2 TEMA E tube-shell cross flow with baffles heat exchanger for two pipe sizes, ½" and ¾".

For a final heat exchanger type selection, the pressure drop for each pipe size (½" and ¾") will be calculated, to choose the most suitable pipe size.

4.1.4.4 Pressure Drop Calculations

The pressure drop will be calculated for the two pipe sizes (½" and ¾") of the 1-2 TEMA E tube-shell cross flow with baffles heat exchanger type. The detailed calculations are presented in appendix **F**.

For the first pipe size (¾"), the pressure drop in the pipes is about 107 Pa, and that is be around 0.036 % of the pressure inside the tubes. The over all pressure drop in the shell side is about 222 Pa.

For the first pipe size (½"), the pressure drop in the pipes is about 140 Pa, and that is be around 0.047 % of the pressure inside the tubes. The overall pressure drop in the shell side is about 400 Pa.

4.1.4.5 Final Selection of the Heat exchanger Type

Two pipe sizes are presented for the final selection of the heat exchanger type, the ¾" size and the ½" size. Both sizes are of the same type which is the 1-2 TEMA E tube-shell cross flow with baffles heat exchanger.

From the cost point of view, the standard length of the pipes available in the market is 20 feet that is around 6m. For the ¾" size, the number of pipes is 16 pipes and every pipe will be cut into four pieces of 1.5m length. However, for the ½" size, the number of pipes is 20 pipes and every pipe will be cut into four pieces of 1.2m length. The primary cost of the 20 pipes of the ½" is higher than the 16 pipes of the ¾" size, and that gives preference to the ¾" pipe design.

From the fabrication point of view, the ¾" pipe design is easier and cost less in fabrication (cutting and welding) compared to the ½" pipe design. The only advantage in ½" pipe design is that the height of combustion chamber will be less which is less cost in modification of the combustion chamber, but this cost actually is not high.

From the pressure drop point of view, the most important thing for this design is the pressure drop in the shell side because the combustion in the secondary combustor will be almost at atmospheric pressure, and any large drop in the pressure could cause a flame failure. In the shell side for the ¾" pipes the pressure drop is around 222 Pa, however for the ½" pipes is 400 Pa and that gives preference to the ¾" pipe design. Therefore, the ¾" pipe size has been chosen for this design.

Plate 4.6 shows the stainless steel heat exchanger during the fabrication process in the workshop of the School of Mechanical Engineering, Universiti Sains Malaysia. Figure 4.10 shows the heat exchanger of the system with the turbocharger engine.

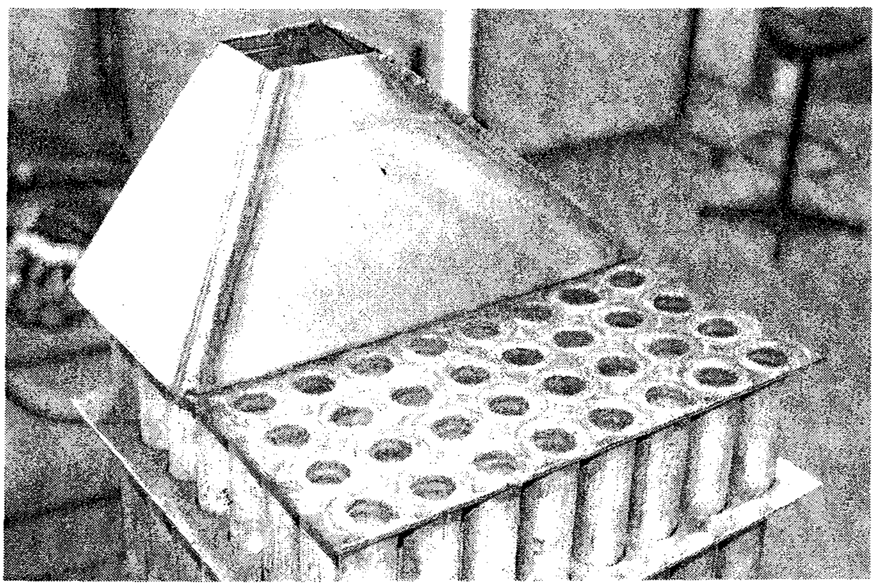


Plate 4.6: The High Temperature Heat Exchanger During the Fabrication

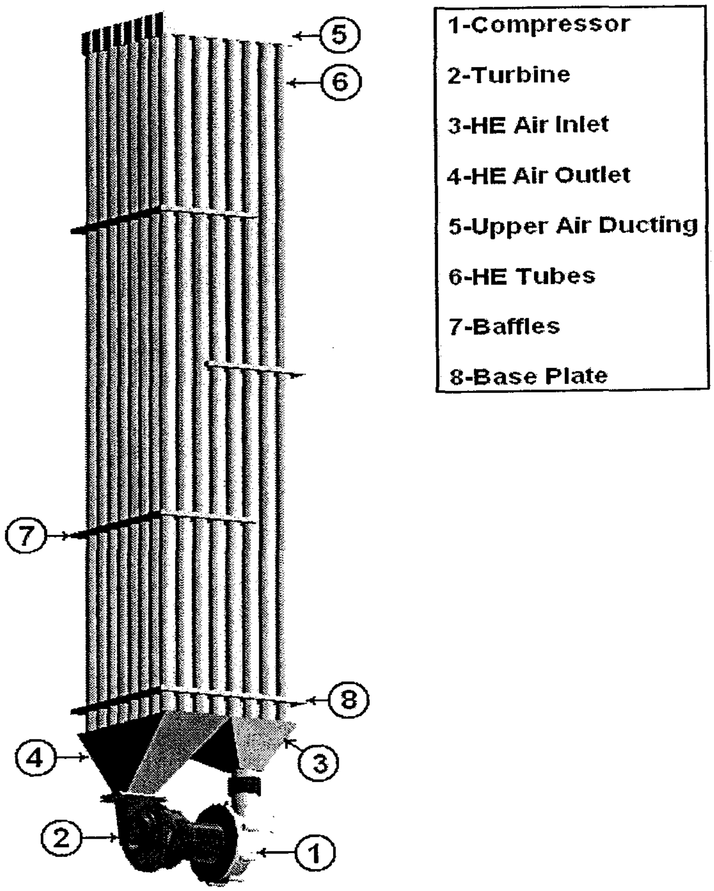


Figure 4.10: The Stainless Steel High Temperature Heat Exchanger of the System

4.1.5 Turbocharger

In this project, a truck turbocharger was used as a micro turbine engine since turbochargers are much lower in prices compared to the micro turbine engines. HOLSET-3LD type turbo-charger (see Plate 4.7) that is used on Ford engine (180 H.P.) has been used. The characteristic chart of this turbocharger was not available because it is an old model with no information about it in the manufacturer website. Some information from another turbocharger TKR-7CT-05 (ISC, 2005) that has the same dimensions as the HOLSET-3LD turbocharger were used. The maximum compressor air flow rate is about 0.25kg/s with 3 bar absolute pressure. This turbocharger was damaged during the turbocharger startup runs as will be discussed in Chapter 5. A new turbocharger HOLSET- H1C with the same dimensions was used (see plate 4.8).

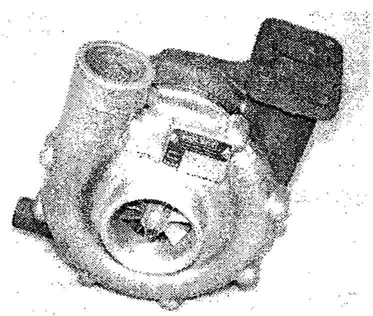


Plate 4.7: HOLSET-3LD Turbo Charger

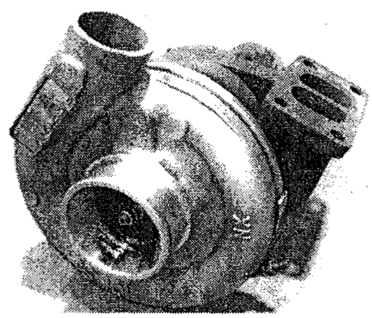
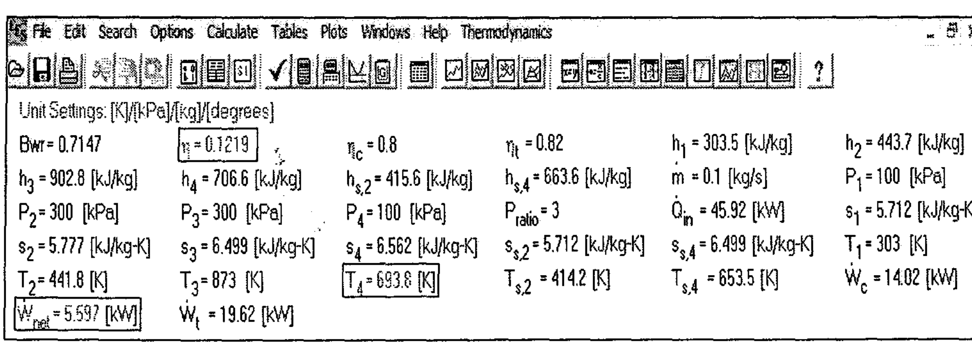


Plate 4.8: HOLSET- H1C Turbo Charger

The theoretical calculations of the compressor side were discussed earlier with the heat exchanger design. For the turbine side, the ideal air temperature in the exit of the turbine can be calculated using the ideal gas properties table of air (table B-1) in appendix **B**. Since the turbine air inlet temperature T_3=600°C as shown in figure 4.6, the relative pressure of this temperature is P_{r3} = 1.43558 (dimensionless quantity), and assuming the turbine inlet pressure P_3 = 3 bar (absolute), the relative pressure for the turbine inlet is calculated using equation 4.23:

$$p_{r4} = p_{r3}\left(\frac{P_4}{P_3}\right) \qquad 4.23$$

P_{r4} = 4.30674, and from the ideal gas properties table of the air (table B-1), the ideal air temperature in the outlet of turbine $Ts_{,4}$ = 141°C. For the turbine side of turbocharger. In Tariq et al.,(1995), the efficiency was 82%. Same turbine efficiency is assumed for the present study. The actual temperature was calculated using the Engineering Equation Solver (limited academic version 6.190), $T_{t,0}$ = 168.6°C. The complete results of the brayton cycle are shown in figure 4.11:



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Unit Settings: [K]/[kPa]/[kg]/[degrees]

Bwr = 0.7147	$\boxed{\eta = 0.1219}$	η_c = 0.8	η_t = 0.82	h_1 = 303.5 [kJ/kg]	h_2 = 443.7 [kJ/kg]
h_3 = 902.8 [kJ/kg]	h_4 = 706.6 [kJ/kg]	$h_{s,2}$ = 415.6 [kJ/kg]	$h_{s,4}$ = 663.6 [kJ/kg]	$\dot{m}$ = 0.1 [kg/s]	P_1 = 100 [kPa]
P_2 = 300 [kPa]	P_3 = 300 [kPa]	P_4 = 100 [kPa]	P_{ratio} = 3	$\dot{Q}_{in}$ = 45.92 [kW]	s_1 = 5.712 [kJ/kg-K]
s_2 = 5.777 [kJ/kg-K]	s_3 = 6.499 [kJ/kg-K]	s_4 = 6.562 [kJ/kg-K]	$s_{s,2}$ = 5.712 [kJ/kg-K]	$s_{s,4}$ = 6.499 [kJ/kg-K]	T_1 = 303 [K]
T_2 = 441.8 [K]	T_3 = 873 [K]	$\boxed{T_4 = 693.8\ [K]}$	$T_{s,2}$ = 414.2 [K]	$T_{s,4}$ = 653.5 [K]	$\dot{W}_c$ = 14.02 [kW]
$\boxed{\dot{W}_{net} = 5.597\ [kW]}$	$\dot{W}_t$ = 19.62 [kW]				

Figure 4.11: Actual Brayton Cycle Calculation Results at 0.1kg/s and 600°C Air Flow and Turbine Inlet Temperature Respectively (Engineering Equation Solver version 6.190)

The compressor and turbine powers are calculated using the general power balance equation 4.24:

$$Q = \dot{m} \cdot Cp\,(\Delta T) \qquad 4.24$$

The turbine net power is calculated using equation 4.25:

$$\text{Turbine net power (shaft power)} = \text{Turbine power} - \text{Compressor power} \qquad 4.25$$

The air flow rate inside the heat exchanger was (0.1kg/s), and it splits into two air streams after the turbine. (0.04 kg air/s) was released as an output thermal power and (0.06 kg air/s) was recuperated back to the chamber. The amount of thermal power recuperated back to the chamber was 24.4 kWt. This value is calculated using equation (4.24). This thermal power caused a reduction in the wood consumption, and with 74.5 kWt combustion power out of the gasifier- combustor, as calculated earlier in section (4.1.4), and 70% gasifier-combustor efficiency, the wood consumption is 17.18 kg_{wood}/hr (equation 4.2).

Figure 4.12 shows a schematic drawing of the power flow of the EFGT system for hot air production, with the efficiency of the different parts of the system, based on the theoretical calculations. The overall efficiency of the system in case of using an electrical generation unit is calculated using equation 4.26:

$$\eta = \frac{\text{Output hot air power} + \text{Electrical output power}}{\text{Input wood power}} \qquad 4.26$$

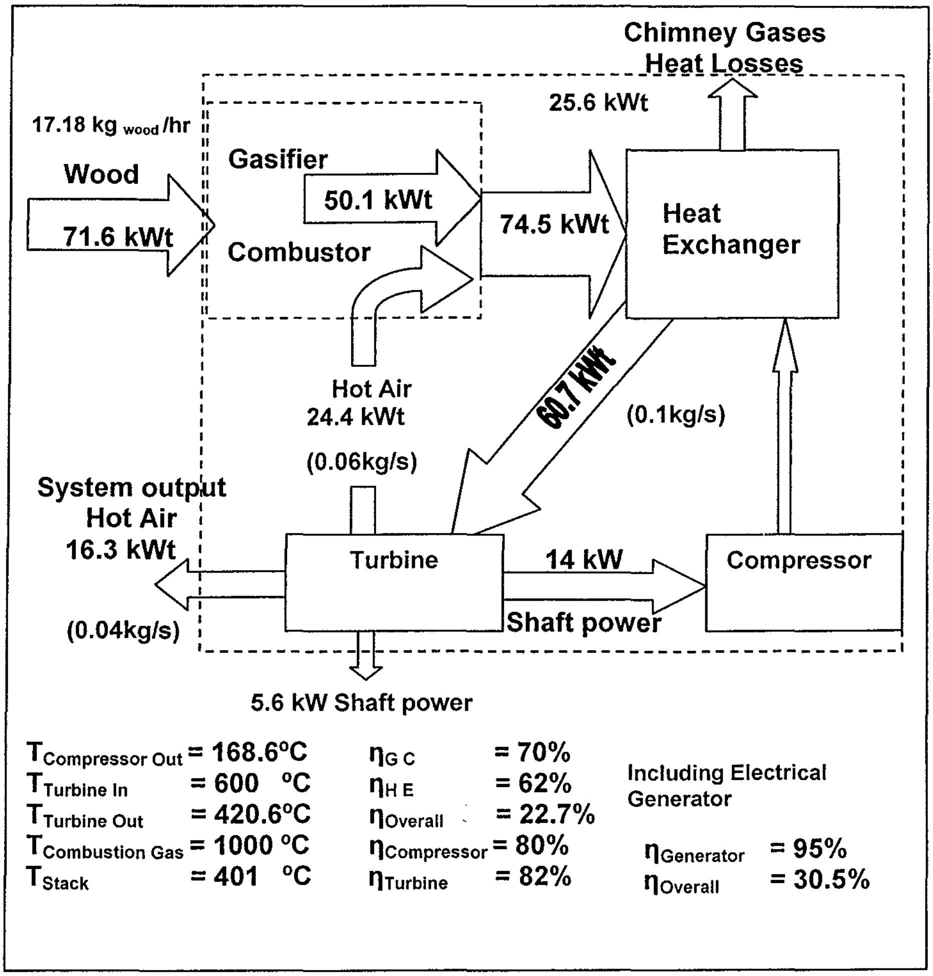


Figure 4.12: The Theoretical Power Flow Diagram of the EFGT System

4.1.6 Lubrication System

It is a secondary system to lubricate and cool the turbocharger's bearings, but that does not mean it is not important. Turbocharger depends totally on the lubrication system since the rotating shaft is totally floating on the oil film which is also acting like a damping system to eliminate the vibrations. The modern micro turbine engines are using air bearings with no oil, but the turbocharger is more suitable for this project for it is much cheaper.

The main components of the lubrication system are:

1. High pressure automotive oil gear pump.
2. Driving electrical motor.
3. Cooling unit.

A secondhand automotive oil gear pump with a driving electrical motor along with a reduction gear box has been used for the first setup with no oil cooling unit (see Plate 4.9). The motor was a conveyer driving motor with a power of 90W and speed regulator, the maximum output speed is 82 rpm.

It has been decided to develop a simple oil cooling unit for the second setup using water. The unit had oil / water heat exchanger where the water was discharged to the drain with no water recirculation, even though it is not practical for commercial purposes, but it will be acceptable for the experiment as a first trial because it was easy to build and it enabled us to control the oil temperature easily by just increasing or decreasing the water flow rate since we did not know how the actual oil working conditions will be. Plate 4.10 shows oil cooling unit in the second set up of lubrication system.

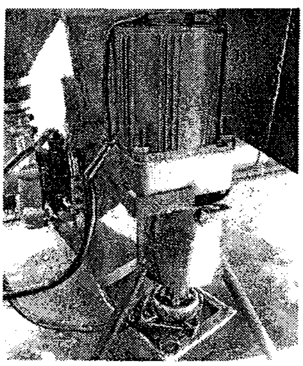


Plate 4.9: 90W Driving Motor, Oil Pump and Oil Tank (First setup)

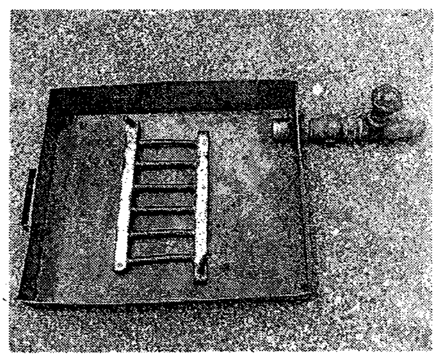


Plate 4.10: Water Cooling Unit with The Oil / Water Heat Exchanger

The third setup was the same as the first one except the electrical driving motor was a 350W motor with a speed reduction pulley system and the output speed was around 900 rpm (see Plate 4.11). In the fourth setup, the motor was 750W with speed reduction pulley system of about 1100 rpm output speed (see plate 4.12).

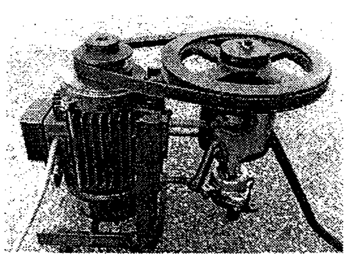


Plate 4.11: 350W Driving Motor with the Oil Pump

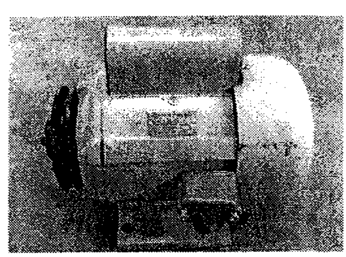


Plate 4.12: 750W Driving Motor

An air radiator along with a fan has been developed as the oil cooling unit for this setup (see plate 4.13).

In the fifth setup, a bigger electrical driving motor of 1.5 kW with speed reduction pulley system of about 2200 rpm output speed has been used (see Plate 4.14) with the same oil cooling unit. A 3 kW voltage regulator was used to reduce the speed of the 1.5kW driving motor, in order to decrease the oil pressure inside the bearings of the turbocharger.

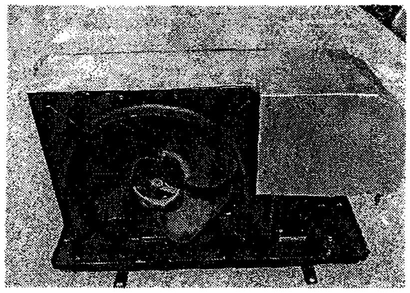


Plate 4.13: Oil Cooling Radiator

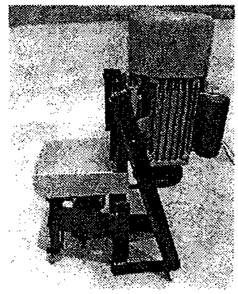


Plate 4.14: 1.5 kW Driving Motor with the Oil Pump

4.2 Fabrication of Hot Air Production Unit Without The Turbocharger

The new system layout after removing the turbocharger is shown in figure 4.13:

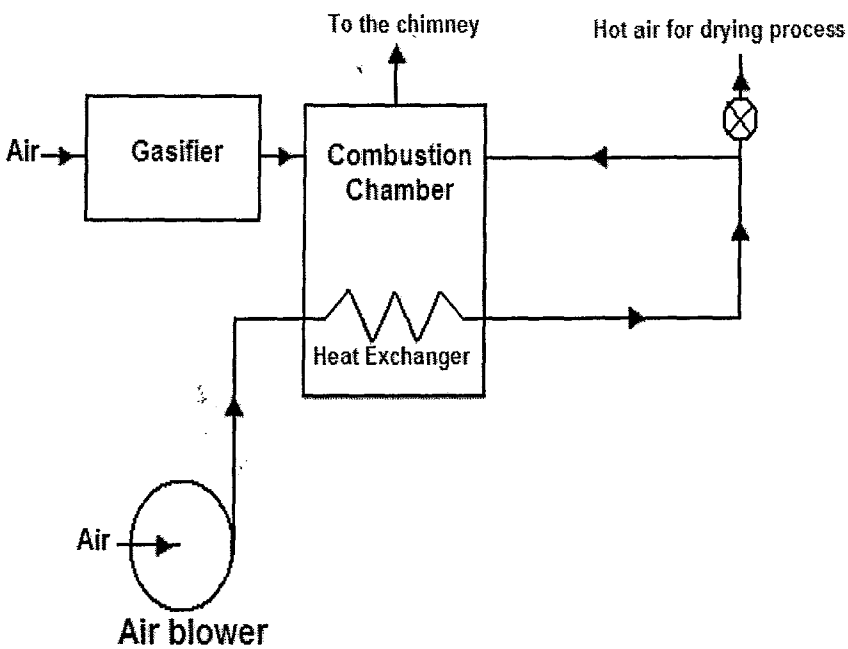


Figure 4.13: Layout of the Unit as a Hot Air Production Unit without the Turbocharger

Between the air blower and the inlet of heat exchanger, a connection pipes have been placed, and a special flange has been fabricated to join the outlet of heat exchanger and the pipes which have been connected to the exhaust of turbocharger (see Plate 4.15).

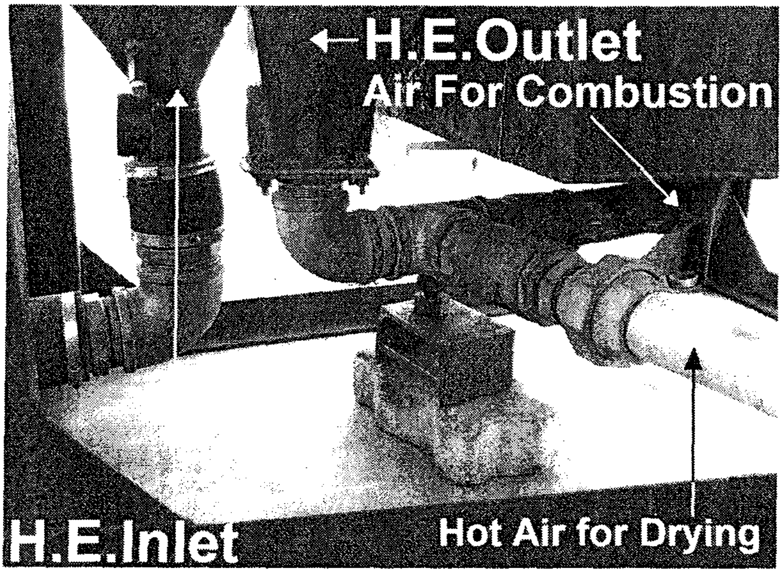


Plate 4.15: The System after Removing the Turbocharger

During the experiments, some modifications have been done on the gasifier- combustor to get a stable performance and low CO emissions, the results of the modifications will be discussed in chapter 4. The modifications were:

1) Increasing the inclining degree inside the primary chamber to reduce the bridging phenomenon. The inclination degree was increased from 45° to 70° to the horizontal axis, by welding a 3mm thickness mild steel sheet on the old sheet (see figure 4.14). This has reduced the weight of wood batch in the first loading from 40kg to around 38 kg.

2) The swirl generator shown earlier in plate (4.3) was fixed on the air inlet pipe at about 30mm before the throat. The mixing process between air and producer gas occurs simultaneously with the combustion process inside the throat which is just about 90mm in length. In order to reduce the CO emissions in the exhaust gases by increasing the residence time and mixing quality, a metallic combustion chamber of about 150mm external diameter and 5mm wall thickness was placed inside the throat and extended about 90mm inside the primary chamber and the swirl generator was placed about 30mm before the metallic combustion chamber (see Plate 4.16).

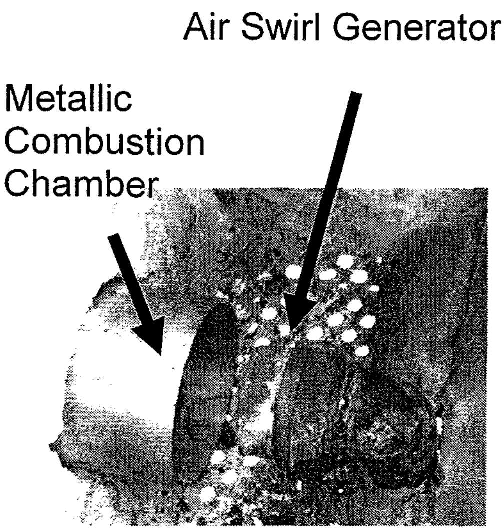


Plate 4.16: The Metallic Combustion Chamber inside the Throat

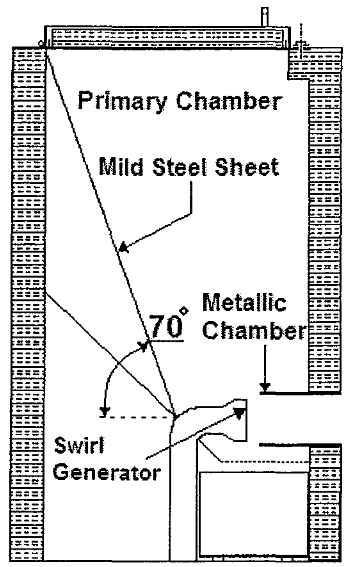


Figure 4.14: 70° Inclination inside Primary Chamber

3) In a subsequent development, the setup of the combustion chamber was changed to get lower CO emissions. A metallic screen with 8mm holes has been placed in the first third of the metallic chamber at about 60mm from the chamber's inlet, the swirl generator was placed at the edge of the metallic chamber (see Plates 4.17 and 4.18). The idea was to create a mixing zone in the first part of the chamber which is about 60mm since the metallic screen increases the air turbulence. The rest of the chamber which is about 120mm was assumed to be enough to complete the combustion process.

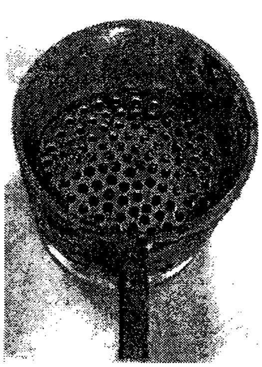


Plate 4.17: The Metallic Screen inside the Combustion Chamber

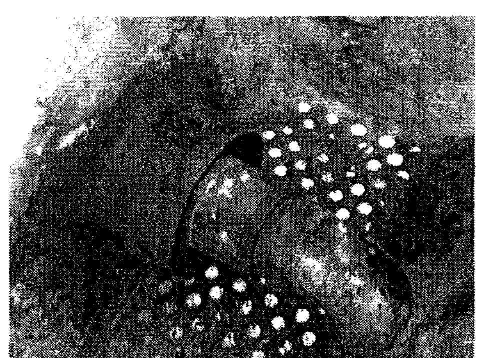


Plate 4.18: The Combustion 'Chamber with the Metallic Screen

The metallic screen has caused a back pressure inside the primary chamber, even after opening a 60mm hole in the center of the screen; this issue will be discussed in chapter 5. New set up has been carried out without the screen, the swirl generator was placed about 30mm before the metallic combustion chamber (see Plate 4.19).

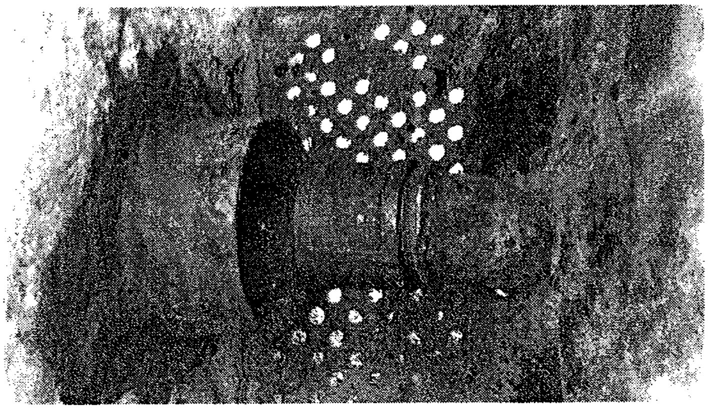


Plate 4.19: The Combustion Chamber without the Screen, with 30mm Distance between the Chamber and the Swirl Generator

4) It has been decided to cut and remove the extension of the metallic combustion chamber inside the primary chamber (see Plate 4.20). The metallic chamber was extended 20mm towards the secondary chamber until it became in direct contact with the heat exchanger to prevent charcoal from dropping in to the 20mm gap between the heat exchanger and the refractory cement wall (see figure 4.15).

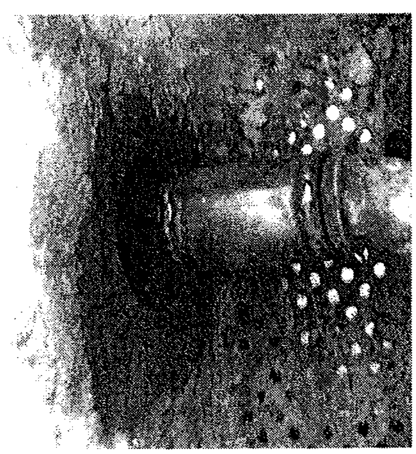


Plate 4.20: The Throat without Extension towards the Primary Chamber

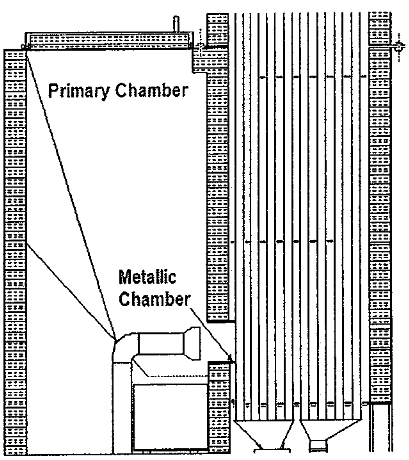


Figure 4.15: Extending the Metallic Chamber towards the Heat Exchanger

4.3 Experiment Setup

4.3.1 Setup of The Turbocharger Startup Experiment

The gasifier startup process was performed using the Liquefied Petroleum Gas (LPG) burner connected to the LPG container (see plate 4.21).

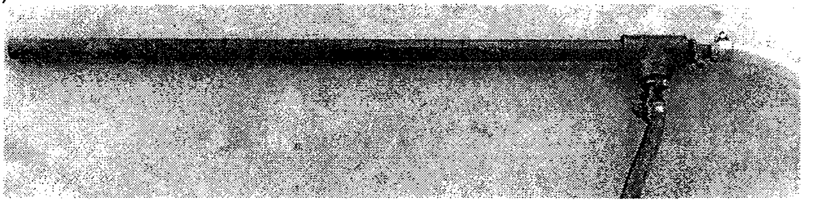


Plate 4.21: LPG Burner

For the turbocharger startup process, the following air blowers were used:

I. First air blower(see plate 4.22):

- Manufacturer : Apex
- Model : ES-729
- Power : 5.5 kW
- Voltage : 41 5V
- Frequency : 50 HZ
- Rotation of speed : 2935 rpm
- Max. air flow rate : 8.8 m^3/min
- Max. air pressure : 2900 mmAQ

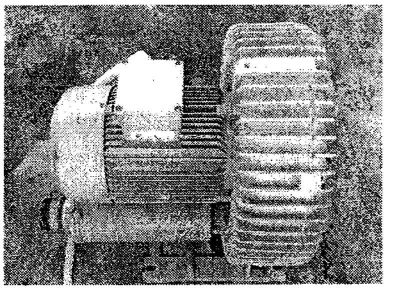


Plate 4.22: First Air Blower, 5.5kW

II. Second air blower (see plate 4.23):

- Manufacturer : Apex
- Model : ES-6375
- Power : 7.5 kW
- Voltage : 415 V
- Frequency : 50H Z
- Rotation of speed : 2800 rpm
- Max. air flow rate : 9 m^3/min
- Max. air pressure : 2600 mmAQ

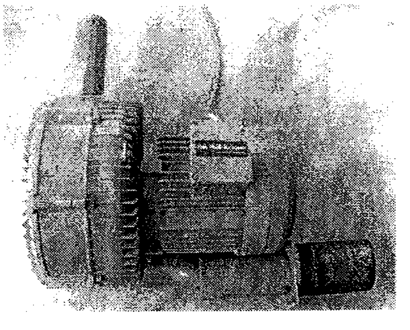


Plate 4.23: Second Air Blower, 7.5kW

In order to get more pressure for the startup process, the 5.5kW and 7.5kW air blowers were also connected in series. To test the system with higher air pressure, the turbocharger has been connected to a 5.5kW reciprocating air compressor, but with low air flow rate. The measuring equipments used in the experiments will be discussed in section (4.3.3).

4.3.2 Setup of the Hot Air Production Experiments without Turbocharger

After removing the turbocharger, the only startup process left was the gasifier startup. The same 5.5kW and 7.5kW air blowers mentioned in section (4.3.1) were alternatively used with the system without the turbocharger.

4.3.3 Measuring Equipment and Apparatus

The required measuring equipments for the experiments with and without the turbocharger can be divided into the following sets of tools:

1. Temperature measurement.
2. Pressure measurement.
3. Air flow measurement.
4. Rotating speed measurement.
5. Biomass fuel weight.
6. Exhausted gases analysis.
7. Wood moisture content measurement.
8. Wood calorific value measurement.

4.3.3.1 Temperature Measurement

In order to determine the thermal performance of the gasifier-combustor, heat exchanger and the turbocharger, 14 pieces of type K thermocouples were used. Thermocouples were connected to a two thermocouple scanner (12 channel Digi-Sense, Mode 69202-30) shown in Plate (4.24).

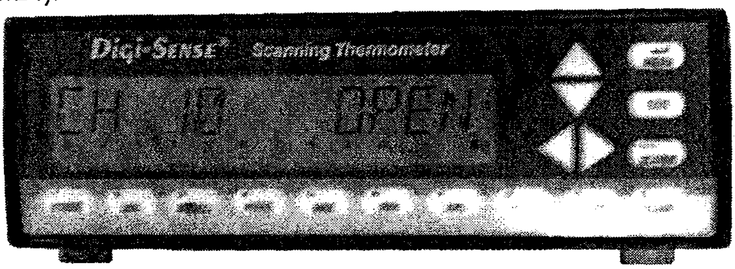


Plate 4.24: 12 Channel Thermocouple Scanner

For the gasifier-combustor, ten thermocouples were located on the gasifier chamber and the combustion chamber as shown in figure 4.16.

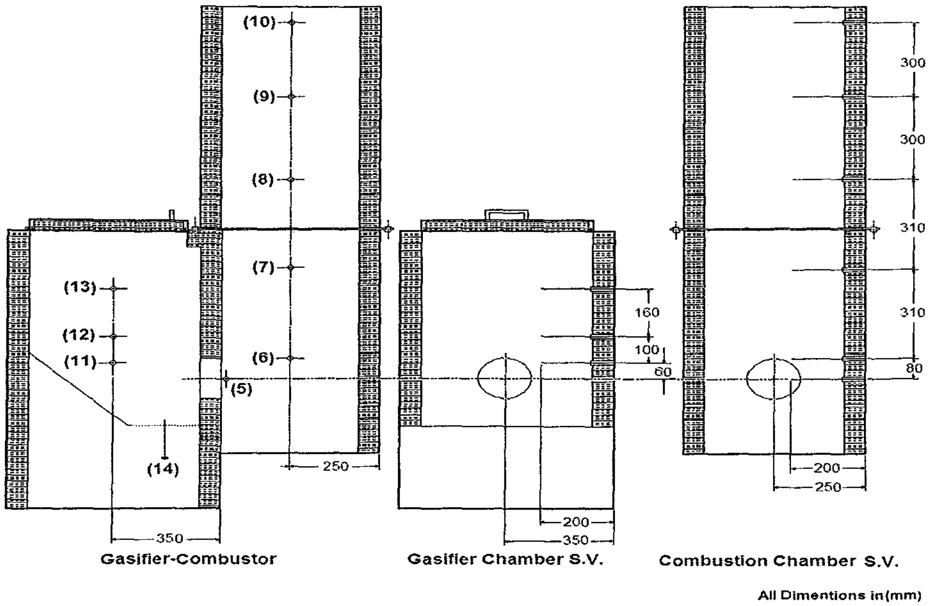


Figure 4.16: Thermocouple Positions on the Gasifier-Combustor

For the heat exchanger and the turbocharger, four thermocouples were used, the positions of these thermocouples are shown in figure 4.17. The same positions were on the system without the turbocharger, but with only three thermocouples (see figure 4.18).

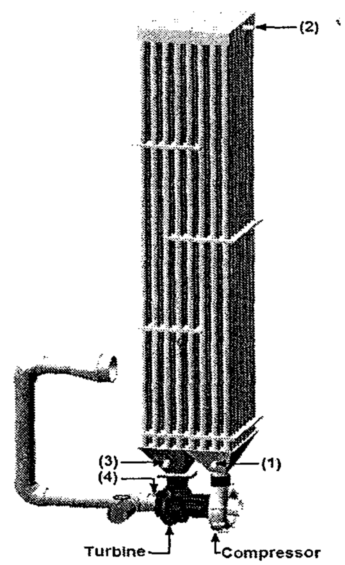


Figure 4.17: Thermocouple Positions on the Heat Exchanger and Turbocharger

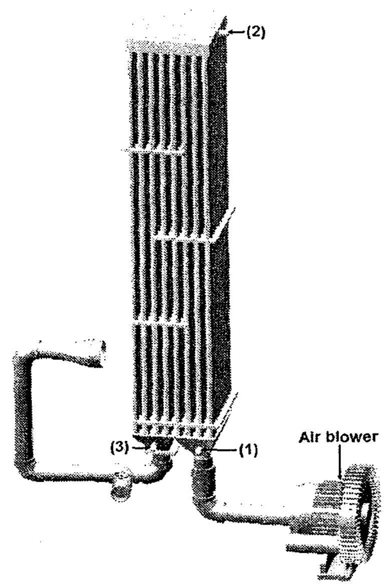


Figure 4.18: Thermocouple Positions on the Heat Exchanger without

4.3.3.2 Pressure Measurement

The maximum air pressure after the compressor was expected to be about 3 bar absolute pressure. Therefore, a pressure gauge of 2 bar (gauge pressure) was placed at the inlet of the heat exchanger (see Plate 4.25). The pressure drop inside the heat exchanger tubes was determined in the cold mode with the 7.5 air blower using a U-tube manometer shown in Plate (4.26). The pressure drop between the inlet and outlet of the heat exchanger was about 2mm water (about 20 Pa). According to theoretical calculations (section 4.1.4.4), the air pressure and flow rate supplied by the compressor of the turbocharger are expected to be about 3 bar absolute and 0.1kg/s respectively. The pressure losses inside the heat exchanger tubes were calculated to be (107 Pa) that is about 0.036 % drop, due to the higher air speeds inside the tubes.

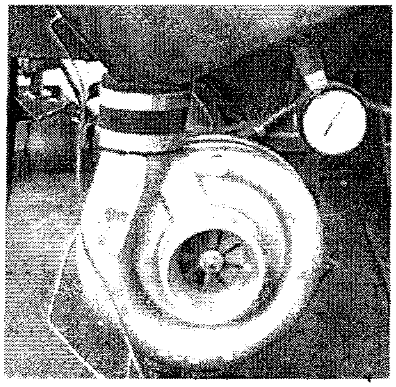


Plate 4.25: The Pressure Gauge in the inlet of Heat Exchanger

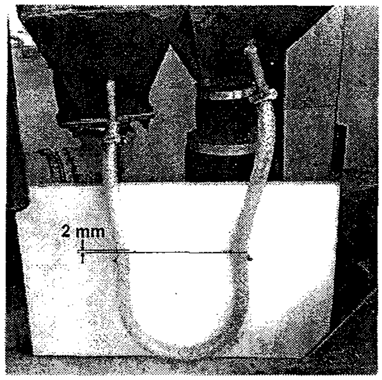


Plate 4.26: U-Tube Manometer to Determine the HE Pressure Drop

4.3.3.3 Air flow Measurement

In the turbocharger startup experiments, the air from the startup blower was passed through the compressor, heat exchanger and the turbine. Therefore, the cold air flow rate in the inlet of the blower was measured using a hot wire anemometer (see Plate 4.27). However, in the system without the turbocharger, the hot air flow rate after the heat exchanger was split into two streams, one to the combustion chamber and the other goes out of the system for any drying process. Therefore, the hot wire anemometer can be used only in the main air stream in the cold place that is the inlet of the air blower. For the output hot air flow rate measurement, an expansion/cooling drum was used to cool down the air (see plate 4.28), and then the air flow rate can be determined using orifice flow meter (see plate 4.29) along with U-tube water manometer.

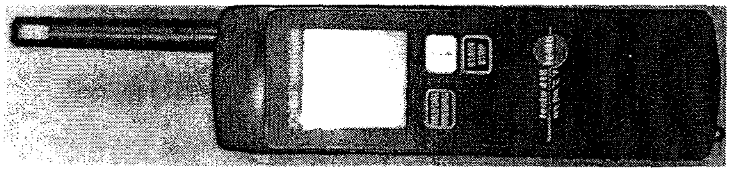
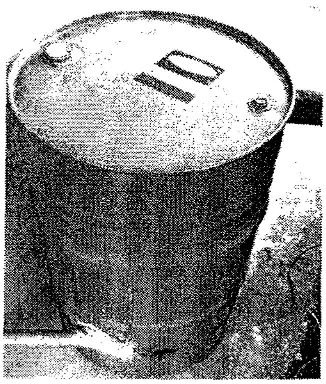


Plate 4.27: Hot Wire Anemometer



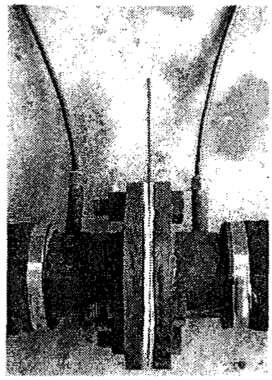


Plate 4.28: Air Expansion Drum Plate 4.29: Orifice Flow Meter

4.3.3.4 Rotating Speed Measurement

Speed of turbocharger should be determined mainly in the self running mode in order to characterize the turbocharger stability and to prevent the over speed. A Lutron photo/contact tachometer model: DT-2236 (see plate 4.30), was used to detect the speed from the compressor side of the turbocharger, since the photo mode can detect up to 100,000rpm. In the startup runs, the air blower was connected to the inlet of the compressor. Therefore, 8mm hole was drilled in the compressor's inlet pipe to allow the light beam of the tachometer to reach the shaft of the turbocharger. However, the tachometer was not able to detect the speed without a special reflecting sticker provided with the device. The readings were available for about 30 minutes after startup, and then the reflecting sticker was fallen off because of the high temperature and speed of the shaft. Replacing the reflecting sticker with reflecting painting colors (black color with a white color line) enabled the readings for just the low speeds, around 10,000rpm.

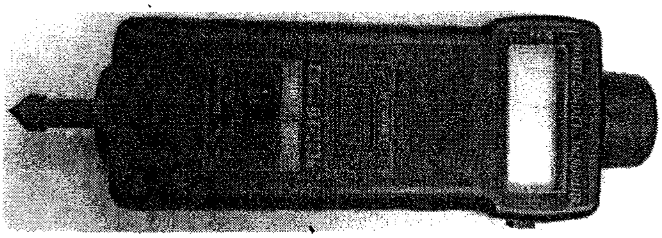


Plate 4.30: Photo/Contact Tachometer

Another possibility was available to detect the speed, by pointing the light beam towards the inlet of compressor, aligned with the axis of the shaft, eventhough without the reflecting sticker. The reason was the manufacturing balancing chamfer in one place on the impeller of the compressor that was reflecting the light beam. However, this option was applicable only after removing the air blower for the turbocharger startup check, and removing the blower takes around 5 seconds. Therefore, the speed readings were less than the maximum turbocharger speed.

4.3.3.5 Biomass Fuel Weight

In order to determine the wood consumption, the first wood batch was weighed using a 10 kg balance, and then the same wood level (full chamber) was maintained in every reloading process. The amount of every reloaded batch was weighed. Therefore, the wood consumption was known for every period of time between two reloading processes. This was the only way to determine the wood consumption, because the output power and therefore the wood consumption start to drop when the amount of wood becomes less than half.

The ash collected in the ash bin were weighed as well, after separating the small pieces of char dropped through the holes of the grate (maximum 6mm in all dimensions). The weight of ash and char coal pieces were about (1-1.1%) and (0.042-0.05%) of the total weight of the wood during the different runs. The carbon conversion efficiency will be discussed in chapter 5.

4.3.3.6 Exhaust Gases Analysis

The gases from the gasifier chamber are directly burned in the combustion chamber. Therefore, the producer gas composition cannot be collected and analyzed. The amount of CO and NOx emissions in the combustion chamber exhaust were determined using the (Kane Automotive, model Auto 4-1&5-1) gas analyzer (see plate 4.31).

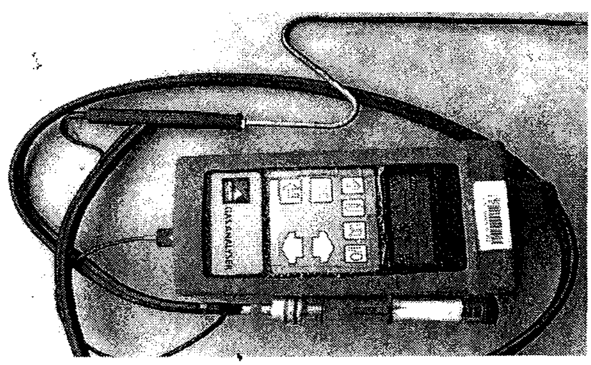


Plate 4.31: Gas Analyzer

4.3.3.7 Wood Moisture Content Measurement

Wood moisture content is one of the most important fuel properties. It can affect the performance of the gasification process and therefore the performance of all the system. The wood sample was cut into shaving and tested using the moisture determination balance, model number: MB200, shown in plate (4.32). The moisture content was found to be around 11.4%. The test procedure and results are presented in appendices **G** and **H** respectively.

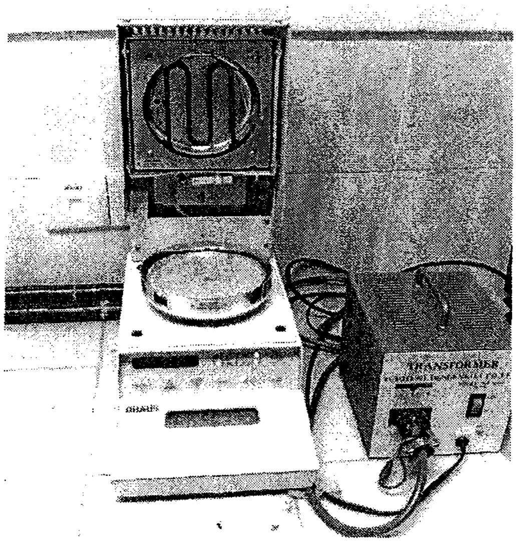


Plate 4.32: Moisture Determination Balance

4.3.3.8 Wood Heating Value Measurement

In order to calculate the efficiency of the system accurately, the wood heating value was determined using a bomb calorimeter, model 1013-B shown in plate 4.33. The high heating value (HHV_{wood}) based on wet wood was found to be around 19.81 MJ/kg, and the low heating value (LHV_{wood}) was 17.29 MJ/kg. The test procedure, and then the results and calculations are presented in appendices **G** and **I** respectively.

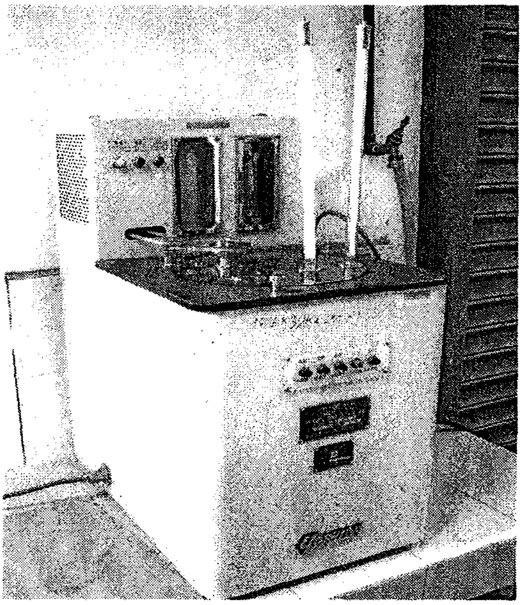


Plate 4.33: Bomb Calorimeter

4.4 Experiment Procedures

Two different procedures with and without turbocharger were applied during the system operation, because the turbocharger startup trials included the turbocharger and the lubrication unit. The procedures are presented in appendix **J**.