BIOMASS COMBUSTION GAS TURBINE CHP

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Contractor

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Executive Summary

Generation of electricity from biomass is not common in the UK at the moment. The traditional method of generating electricity from biomass involves producing steam by passing the hot biomass combustion gasses through a waste heat steam boiler. This steam is then used to drive a steam engine or turbine generator. Steam based systems provide low fuel to electrical output efficiency's, due to the dumping of heat in the condensation phase, as a result fuel feed rates high and the electric outputs are low.

Small-scale steam based generation using biomass combustion is very expensive due to steam raising and dissipation equipment needed, this leads to unacceptable payback periods.
The economics of steam are hard to justify under 1MW due to low efficiencies and high capital costs. Steam is easier to justify when there is a use for the excess heat (CHP), but this relies on a constant demand for heat when the system is generating.

Steam engine tends to have higher efficiencies, but costs are high compared to turbines. Manufacturers of steam engines have reduced dramatically, due to the low demand.
Alternative methods are currently being studied these include gasification and direct firing of gas turbines. These methods also have problems and are unlikely to economically viable under 1MW, in fact 10MW gasification plants struggle to be viable.

There is a large demand for simple highly efficient biomass generators under 1MW. Energy crops scheme's being pushed currently require an efficient energy end use. The more efficient the biomass to electrical energy system is, the easier it will be to justify and promote energy crops. The smaller size system provides benefits in that fuel transportation costs can be reduced. An added benefit is that power can be provided where it is needed, reducing the strain, cost and power losses in cabling to remote locations.

The objective of this work was to develop a small-scale biomass combustion generating system. The system using a biomass combustor will fire a micro-gas turbine indirectly via a high temperature heat exchanger. The successful demonstration of the indirect fired gas turbine will provide confidence and further investment in this high efficiency system.

A short report was first commissioned to outline the specification of commercially available micro-turbines and determine their suitability for indirect firing. From this report an understanding of the emerging micro-turbine market was obtained. The report also provided the bases for a preliminary evaluation of suitability.

A biomass combustor based on Talbott's standard range was manufactured in Talbott's test room in Stafford. A high temperature heat exchanger was designed, manufactured and fitted to the biomass combustor. A series of tests were conducted to evaluate the thermal performance characteristics of the biomass combustor and heat exchanger in a simple low-pressure test rig.

A PC based mathematical model of the system was written to predict compatibility of the biomass combustor and heat exchanger with various compressors and turbines packages. From this software Talbott's evaluated the suitability of gas turbine and micro-turbines. Separate compressors and turbines were also studied.

After this evaluation, turbine manufacturers of compatible equipment were contacted for further detailed analysis. For the installed biomass combustor and heat exchanger, it was decided that the Bowman Power TG50 was the best match.

Bowman Power and Talbott's worked together to modify a standard TG50 turbine to incorporate our biomass combustion system. Modifications were made to the compressor and turbine housings along with extensive software changes. Redundant gas fuel lines were removed and the oil cooling system had to be re-sited. A high temperature control valve was developed as an emergency override for over-speed protection. A system to control speed by grid load was also developed, along with startup and shutdown procedures.

Waste heat from turbine exhaust is captured by modified ductwork and returned into combustion air streams. This provides a reduction in fuel consumption, but required extensive modification to combustor internals to cope with temperatures without causing back pressure problems.

The biomass generation system was tested and data captured in real time via PC based ModBus and other data logging packages.

The main results of the tests were as follows:

- The system achieved turbine generation speed with steady acceleration although much slower than conventional gas turbines. Heat exchanger performance increased with turbine speed. Temperatures and mass flows are critical for turbine acceleration. Higher thermal inertia is required to quickly accelerate passed turbine critical speeds. The heat exchanger has proved the principle works, and should now be developed to increase its efficiency and life cycle.

- From the test data the predicted power generated from the Biomass Combustion Turbine was 30kW.

- Although the system was still at an early stage of development, and could benefit from an increase in both combustor and heat exchanger size, capital equipment costs for this 30kW$_e$ prototype are around £2,500 per kW$_e$. Comparing this with our current steam based 50kW$_e$ CHP system with an electrical efficiency of 8% and cost of £6,455 per kW$_e$. This represents a great leap forward at this size.

- Commercial prospects for this technology are good with existing heat only installation, it is predicted a large number will be interested in producing electric from their existing waste fuel supplies. The 30kW$_e$ unit could provide an offset to normal base load of 250kW$_e$ average to wood working factories. With electric being purchased at 5p/kW this would save £1.50 per hr.

operating this unit for 8000hrs per year will save £12,000, ROC will add another 3p/kW (although sold on the open market may yield 5p/kW), this add a further £7,200. Therefore the payback period can be calculated to 4years. Heat output will make system more attractive as well, by cutting gas or oil costs. Larger units with higher efficiencies and lower costs per kW will of course, provide much faster pay back, a 250kW$_e$ system is expected to have a payback period of just 2 ½ years.

From the above results and discussions the following conclusions may be drawn:

- Biomass Combustion Turbine project has successfully provided proof of principle
- Biomass Micro Gas Turbine acceleration is slower that conventional gas turbines.
- The system has demonstrated it is possible to produce 26-34kWe of electrical renewable energy plus recoverable thermal energy.
- Approximately 100-150kW of high-grade thermal energy is available for simple conversion to heating water or air.
- Low heat losses results in overall system efficiency between 80-85%
- Electrical efficiency is 17% and will rise with further development

It is recommended:

- To run the system for a period of 1 year to provide data on the long term effects on materials at the high temperatures.

- Improve the overall cycle efficiency.

- The biomass combustor should be re-design to incorporate the higher air return temperatures and volume flow rates, other improvements could be incorporated inline with our experience of this system.

- To utilise waste heat from the system to be used in heating and cooling applications. Thus making an efficient biomass fired Co-gen and Tri-gen system.

Contents

1. INTRODUCTION

Heat only biomass combustion systems are well established now with in excess of 3000 industrial installations from Talbott's alone. But the continuing and increasing demand for electricity has left power producers few economical options for truly green renewable electricity generation. Large-scale systems whilst providing economies of scale can suffer large fuel transportation costs in terms of fossil fuels used. This makes them less environmental friendly than they first appear.

Wood based biomass fuels have a much lower calorific value than fossil fuels and a low density. Combining these factors results in transporting biomass expensive. Small scale distributed power can reduce transport and provide power where it is needed. Electricity distribution losses through long cable runs can also be avoided, as much as 20% can be lost in this way. Also it is more difficult to find a use for the left over heat in larger CHP systems.

Current small-scale biomass combustion CHP systems are based on raising steam to drive a steam engine or turbine. This system has poor efficiency at such small-scale ($30kW_e$), with typical efficiencies of 6-8%. Capital costs per kW_e of generation capacity are also high due to complex systems and low efficiencies, typically £6500 per kW_e. Commercial opportunities for this system are small. To be economically viable this system requires very high electricity prices, a constant need for heat and an abundance of free or waste fuel.

To assist with the renewables obligation in way that will be economically viable, a biomass combustion generation system is needed to provide high efficiency at a price the market can afford.

In response to these needs Talbott's heating are developing a small-scale biomass combustion CHP system based on indirect firing of a micro gas turbine. One of the main concerns in this area is capital cost. Small combustion based systems have always had a high cost per kW, which has stopped many a project. The objective of this project is to integrating low cost micro-turbines with the flexibility of biomass combustion, to provide a simple low cost solution to this problem and open a new market for biomass combustion generation.

2. EVALUATION OF BIOMASS INDIRECT FIRING OF A MICRO-TURBINE

The project began by reviewing the work already carried out in this area which included some carried out by BG Technology. Contacts with them indicated that they had established links with microturbine manufactures to seek collaboration and carry out trials. However, they suggested that turbine manufacture Capstone were unable to provide the necessary technical assistance to back up their turbine with Talbott’s new solid fuel system. Other companies were approached to determine suitability.

If a suitable manufacturer could not be found, Talbott’s still had an option to revert to a test rig system to prove performance, although this was not a preferred option.

2.1 Computerised System Model

To enable a better understanding of Gas Turbines in their current technology, Talbott's project manager attended a short five-week course on Gas turbines at Staffordshire University. This course has provided a good understanding of turbine cycles and the prediction of performance. Following this course an evaluation of our system was carried out, giving us a prediction of the systems performance. Talbott's project manager has continued links with Staffordshire University, and they have provided numerous hours of in-kind support.

The original design specification for the heat exchanger at the start of the project defined conditions to satisfy a Capstone micro-turbine with a 150kW of thermal energy transferred to the turbine.

The heat exchanger design calculations were also added to the model. This allowed the performance of the whole system to be predicted and the effects of modifying compression ratios, Combustion temperatures and heat exchanger surface areas etc to be studied. Modifying these parameters gave an insight to how the system would operate and what could be modified to improve system performance.

From this study it was established that turbine inlet temperature (TIT) has large effect on system efficiency. This is caused by increasing volumetric flow seen by the turbine for a given unit of pressure and mass flow generated by the compressor. This increases the mechanical work done by turbine for any given parasitic compressor losses. This increases the net work in the "Work Ratio". Work Ratio is the ratio of net mechanical power to the total mechanical power generated by the turbine. Therefore higher net power in the work ratio gives better efficiency.

Pressure ratio and mass flow are also critical, these factors heavily effect compressor consumed power but also benefit turbine work. These should, therefore, be optimised for peak efficiency.

2.2 Heat exchanger design

As the heat exchanger is the key to the success of the project, a detailed thermal analysis of the proposed cost-effective heat exchanger was performed. The results of this analysis demonstrated that the original design was not suitable. The calculations predicted a thermal performance was less than the required heat transfer.

The heat exchanger was redesigned with several geometries studied, a new design was found to be suitable, although testing will be needed to confirm this. Alternative designs should also be tested to find the best type for our system.

The original design was to transfer 150kW thermal energy to the compressed air. The exchanger was to output the compressed air at a temperature of up to 800^{o}C. Compression of the air would provide 200^{o}C, therefore a lift of 600^{o}C was required. Combustion gasses from biomass were to enter the heating side of the exchanger at a nominal 900^{o}C.

The compressor pressure for the system was specified at 3.5 bar gauge. This pressure at ambient temperatures generally causes very low stresses on steelwork, unless of course the area on which it acts is large. However at elevated temperatures, materials become less resistant to stress. Therefore stresses caused by pressure must be calculated at the design temperatures.

As the heat exchanger is subjected to high temperatures, thermal expansion must be allowed for. Also there is a high temperature gradients in both flow paths, therefore careful directing of flows is required to ensure that the heated air is not cooled by combustion gasses. To achieve this a counter flow design was selected, with geometry to provide the highest temperature difference available.

Biomass combustion often results in high fouling of heat exchangers. Lighter ash fractions can easily be drawn through the combustion chamber and into the heat exchangers narrow passages. If the passages are too tight, the ash could block sections completely. But if the passages are too large, the heat transferred is substantially reduced. Unburnt hydrocarbons that have combined with water vapour (soot) cool and settle on the heat transfer surface reducing heat transfer over time. Regular cleaning will be required, when thermal performance diminishes.

2.3 Selection of a turbine

Designs were developed for the test rig, using various turbine packages, including twin shaft and a multistage compression and inter-cooled systems. The twin shaft arrangement will benefit our control of the turbine/compressor and allow a much wider compressor map without the need for physical modifications to the compressors geometry. Also the exact mass flow rate can be set and adjusted independent of turbine speed, giving us greater flexibility and increase turn down. The twin shaft design also lends it self well to the geometric layout of our system with the compressor mounted at the inlet end of the exchanger and the turbine at the outlet end. Even the turbine exhaust could be neatly returned to the combustor. This twin shaft design could easily be upgraded by a multistage compressor with inter-cooling, apart from the obvious benefit of increased power, the temperature at turbine inlet will drop providing and increased factor of safety and a longer exchanger life.

Talbott's identified a supplier of gas turbine package with 30years experience in this field. The Firm Columbia Power Systems Inc, based in Calgary repair, maintain, modify and package turbines. They have worked with a company that was investigating a similar system, so their input would be very useful. After lengthy phone calls Talbott's decided to visit CPS to see their factory and installations, with a view to getting some sort of collaboration. The visit to Calgary went very well. CPS was run by an experienced turbine engineer, with assistance from a group of 10-15 staff. Their expertise has been rewarded by Capstone give them the rights to distribute the new 30kW microturbine. Talbott's personnel was able to see various gas turbines from Rolls Royce Allison 3.2MW to Solar 75kW units being package or reworked.

On return CPS and Talbott's personnel discussed the IFGT system. CPS was keen to assist on the turbine side and agreed to provide help on the turbine side. Talbott's agreed to purchase a turbine and CPS would modify it to suit our needs. CPS suggested a 75kW Solar Sparten liquid fueled unit; this turbine was suitable because of the external combustion chamber, which should allow the introduction of hot air to the turbine. If this unit cannot be modified, we could use a separate compressor, driven by an inverter controlled motor, and a separate turbine. This approach has benefits in that we may vary the mass flow easily to fine tune performance and assist the control of the turbine shaft speed.

Unfortunately shortly afterwards CPS were taken over, and did not have the resources to continue this project, in our time scale.

However, Talbott's continued evaluation for twin shaft design by contacting suppliers for separate compressor and turbine components. Most compressor manufactures offered a screw type unit, which was economically viable, but were low in isentropic efficiency and therefore consumed too much power for the system. To increase isentropic efficiency centrifugal and axial compressors were evaluated. These types gave better performance and would allow the system to achieve a net power output from the system, but costs were high. Talbott's however found one company whose product price/performance was acceptable.

After evaluating the compressors, Talbott's moved on the turbine expander. There were a number of avenues to evaluate here. Firstly Steam turbines, on first inspection these seamed to be a good prospect robust design with lower costs. However, these were not suitable due to the relatively low temperature limits of between 250^{o}C and 400^{o}C giving poor overall performance. Gas expanders were also looked at but also suffered from lower design temperatures; two companies produce high temperature units but unfortunately not in our size.

Talbott's talked to two suitable gas turbine manufactures. One company was interested in modifying a 600kW$_e$ unit to run on hot air. They are confident that it would produce between 270kW$_e$ and 330kW$_e$. They were only willing to provide one on a commercial basis at this time. Cost for this unit was way over Talbott's budget and its thermal input was also much higher. The other was a new low-cost design to produce between 100kW$_e$ and 200kW$_e$, but its cost was well above Talbott's budget and also would require a larger combustor and associated heat exchanger.

Other possibilities included a second hand APU type gas turbine that would require modifications. A microturbine with recuperator would cost too high unless the manufacturer provided some assistance. Lastly, an automotive derivative test rig was also an option. It appeared from Talbott's discussions with turbine manufactures that the main area for concern was the heat exchanger, the turbines being the low risk.

After reviewing all the data collected, Talbott's reproached a company who had declined to offer, thinking the system working at lower temperatures would not have an acceptable performance. After meeting this company, Bowman Power, and discussing the system face to face, Talbott's concluded that a modified Bowman power 50kW$_e$ turbine was suitable. Bowman Power offered to provide assistance and a reduced cost turbine to be inline with Talbott's available budget.

3 TEST RIG FOR BIOMASS INDIRECT FIRING OF A MICRO-TURBINE

3.1 Heat Exchanger Evaluation

After completion of the manufacturing and fitting to the biomass combustor of the heat exchanger, a test rig was constructed to evaluate the heat exchangers performance. A low-pressure test rig was designed to simulate the thermal conditions of the biomass fired turbine cycle.

The fuel silo was filled with biomass fuel. The combustor is fed via a feed system consisting of agitator, rotary valve and transfer screw. The rotary valve regulates the rate of fuel feed. This rotary valve can be manually adjusted to vary the feed rate, depending on density and calorific valve of fuel.

Figure 3.1.1: Combustion Chamber and Heat Exchanger

Before the fuel enters the combustion chamber it is pre-heated, by hot gasses passing over semi-conductive refractory ceramics (Fig.1). Primary air from combustion fans provides a vigorous combustion reaction. Carefully adjusted secondary air ensures complete combustion and reduced emissions. The hot gasses from the combustion chamber are drawn by an induced draught fan over and under walls, to minimise particulate carry over. Then up into the heat exchanger, and through its multiple passes. Exhaust gases are then cleaned via a cyclonic separator. A second fan pushes our heated air through the exchanger on the other side of the tubes. A proportion of the heated air is then ducted back and combined with ambient air to simulate the heating effect of compression. The remaining air is returned to the combustion chamber.

3.2 Biomass Generator Test Rig

After the analysis of the thermal performance, a micro-turbine manufactured by Bowman Power was installed. This turbine was extensively modified to suit our system. Modifications include:

- Compressor housing and ductwork
- Turbine housing and ductwork
- Removal and sealing of combustor liners and gas lines
- Turbine Exhaust ductwork
- Development of a fast acting high temperature dump valve for over-speed protection
- New software for start-up and speed control

System description

The fuel silo was filled with biomass fuel. The combustor is fed via a feed system consisting of agitator, rotary valve and transfer screw. The rotary valve regulates the rate of fuel feed. This rotary valve can be manually adjusted to vary the feed rate, depending on density and calorific valve of fuel.

Figure 3.2.1: Biomass Combustor connected to Micro-Turbine

Before the fuel enters the combustion chamber it is pre-heated, by hot gasses passing over semi-conductive refractory ceramics (Fig.1). Primary air from combustion fans provides a vigorous combustion reaction. Carefully adjusted secondary air ensures complete combustion and reduced emissions. The hot gasses from the combustion chamber are drawn by an induced draught fan over and under walls, to minimise particulate carry over. Then up into the heat exchanger, and through its multiple passes. Exhaust gases are then cleaned via a cyclonic separator.

Turbine air is drawn in through an air filter into the compressor. The compressor provides the motive force to drive the mass flow of air through the heat exchanger at pressure. Heat is transferred to the pressurised air in the heat exchanger. The heated and pressurised air is then expanded over the power turbine. The power turbine drives the compressor and high-speed generator. The expanded gases are reduced in temperature and pressure. Exhaust gases are then returned to the combustion chamber in high temperature ducts for reuse.

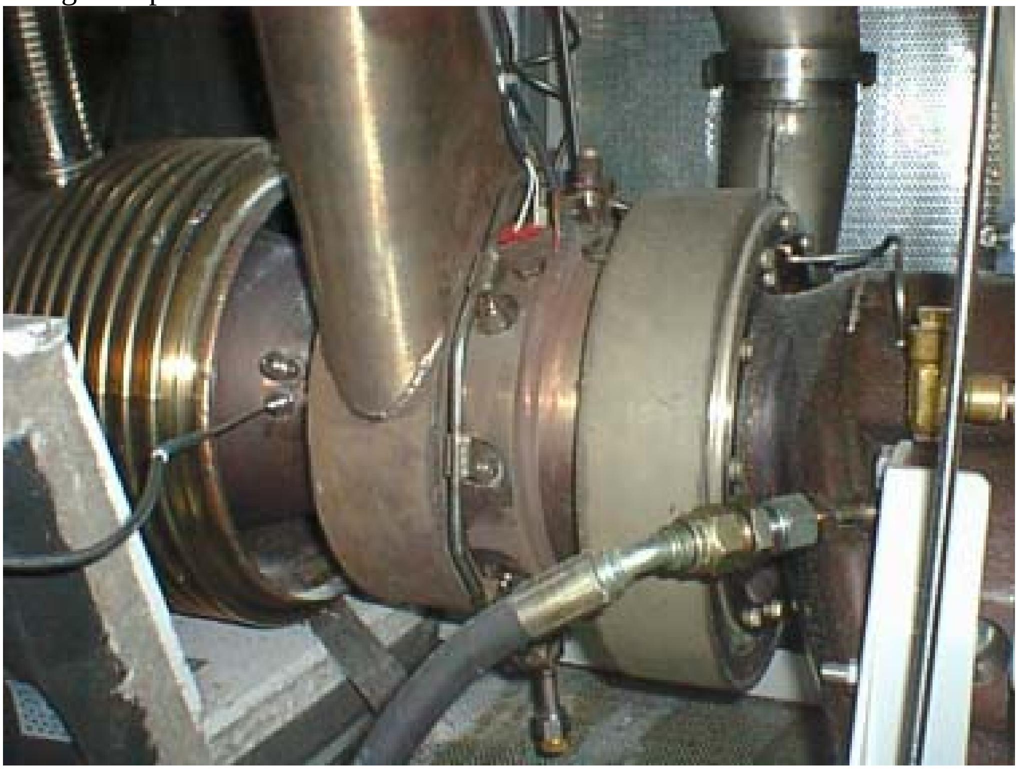
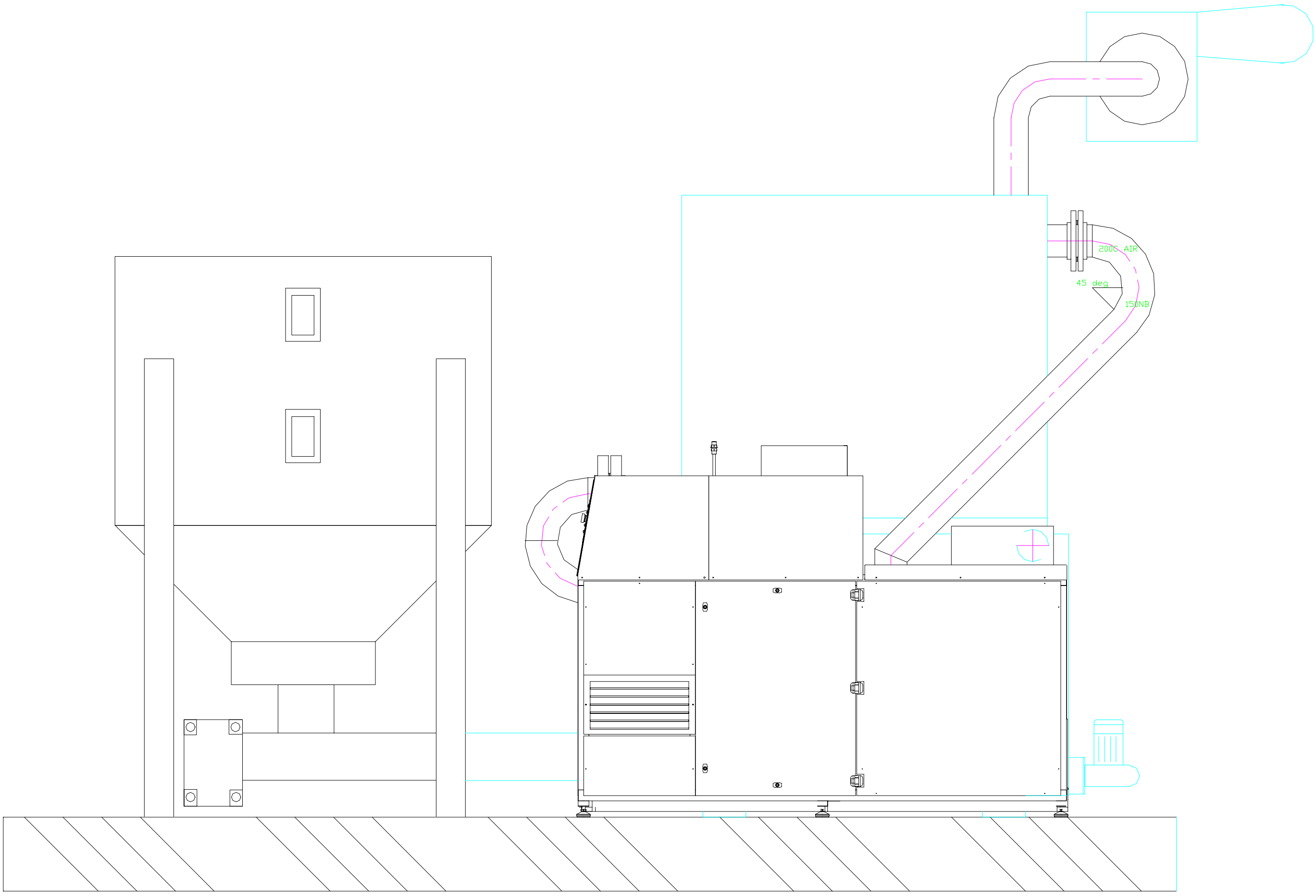


Figure 3.2.2: Turbine Engine assembly

A small compressor was used to provide instrument air to drive the turbine quick pressure release valve. The turbine control software electrically actuates this pneumatic operated valve, thus providing a fail-safe means of protection.

Photo 3.2.3: Turbine Pressure Relief Valve

Biomass Generator General Arrangement Drawing



Drawing 3.2.1: GA of Biomass Generator System

3.3 Instrumentation

The following equipment was used to record temperatures

Picolog TC-08 connected to
a PC with the following
thermocouples

Measurement	Tagg	Location	Device	Signal
Combustor Temperature	4	Inlet	TC Type K	MV
	7	1st Pass	TC Type K	mV
	5	2nd Pass	TC Type K	mV
Exhaust	2	Outlet	TC Type K	mV
Air Temperature	1	Inlet	TC Type K	mV
	3	Outlet	TC Type K	mV
Return 1	6		TC Type K	mV
Return 2	8		TC Type K	mV

A Testo turbine type anemometer was used to measure velocity, and converted to mass flow. Actual mass flow is corrected for each reading.

A hot wire type anemometer was also used to measure velocity

A Digitron 2081P differential pressure gauge was used to measure pressure drops.

The following Turbine Conditions were monitored and logged using MODBUS:

Temperatures
- Air intake Temperature
- Compressor discharge Temperature
- Turbine Inlet Temperature
- Exhaust Gas Temperature

Pressures
- Compressor discharge Pressure
- Turbine Inlet Pressure
- Exhaust Gas Back Pressure

Power
- Current on 3 Phases
- Voltage on 3 Phases
- Frequency

Speed
- Turbine Engine Speed

4 TESTING

After the commissioning test runs a series of experiments were conducted with resulting data being logged. The following experiments were performed:

1. Initial Heat Exchanger performance test
   8 channels of temperature recordings were captured as the combustor was lit, warmed up and run for 4hrs then shut down. Performance was compared with design.

2. Second Heat Exchanger performance test after modification
   The first experiment was repeated, after performance improving modifications were completed. Again 8 channels of temperature recordings were captured as the combustor was lit, warmed up and run for 4hrs then shut down. Performance was compared with design and experiment 1

3. Micro Gas Turbine Performance Test
   This test was conducted to provide a benchmark and an understanding of the performance of TG50 micro-turbine. The turbine entry temperature is higher than the Biomass Generator system, hence higher power output.

4. Initial Biomass Generator performance test
   This test was to initial data on system performance and to define starting procedures. 8 channels of temperature recording were captured as the combustor was lit, warmed up and run for 8hrs then shut down. Compressor air was initially off, then run up to 34,000rpm for approx. 15min. Data was recorded in real time, with measurements taken every 30sec, heat transferred is in kW. The ModBus system logged the turbine operating conditions.

5. Second Biomass Generator performance test after modifications
   This test was aimed at accelerating the turbine up to its normal running speed of 100,000rpm. Also the systems performance due to heat exchanger modifications was observed. 8 channels of temperature recording were captured as the combustor was lit, warmed up and run for 8hrs then shut down. Compressor air was initially off, and then run up to various speeds. At 42,000rpm and 700^{o}C unit began a strong acceleration up to its operational speed. Thermal performance from the heat exchanger was good, with temperatures rising to a maximum 728^{o}C under full flow conditions. Data was recorded in real time, with measurements taken every 30sec, heat transferred in kW. The ModBus system logged the turbine operating conditions.

Biomass Fuel Used

The Biomass fuel use for the above tests was wood pellets. Wood pellets were selected because of their high calorific value, low moisture content and uniformity. They are readily available and remove fuel inconsistencies.

Wood Pellet Data

Size	8mm dia x 15-20mm long
Calorific Value	17,600kJ/kg
Moisture Content	20% (assumed)
Ash Content	0.5%
Costs	£100 per tonne delivered

Other Biomass fuels were tested, including hardwood shavings, softwood chips, MDF dust and miscanthus. The results of these tests are not detailed in this report, but heat outputs were similar.

Emissions

Emissions were not measured during these tests. Emissions should be tested at a later date. It is expected however, that they will be inline with previous "heat only" tests completed on other Talbott's units and detailed early in this report.

Total testing time for the combustor and heat exchanger is estimated at 150hrs. Turbine testing is estimated at 18hrs with electrical assist and 2hrs at generating speed (hot air only).

5 RESULTS

Listed below are the results from experiments performed. Logged data was captured on an ongoing basis.

5.1 Initial Heat Exchanger performance test

8 channels of temperature recording were captured as the combustor was lit, warmed up and run for 2-3hrs then shut down. Data was recorded in real time, with measurements taken every 30sec, heat transferred in kW.

Below is a sample of the results.

Table 9: Test 1

Time (sec) (From Start)	Air in (°C)	Air out (°C)	ΔT (°C)	Cp kJ/kg °K	M (kg/s)	$Q_{Exchanged}$ kW
7680	101.3	560.44	459.14	1.14	0.23263	121.7631
7710	101.42	560.89	459.47	1.14	0.232555	121.8116
7740	101.59	561.23	459.64	1.14	0.23245	121.8013
7770	101.77	561.54	459.77	1.14	0.232338	121.7773
7800	101.84	561.83	459.99	1.14	0.232295	121.8128
7830	101.94	562.07	460.13	1.14	0.232233	121.8174
7860	102.11	562.13	460.02	1.14	0.232128	121.733
7890	102.16	562.01	459.85	1.14	0.232097	121.6718
7920	102.13	561.81	459.68	1.14	0.232115	121.6366
7950	102.19	561.56	459.37	1.14	0.232078	121.5351
7980	102.22	561.18	458.96	1.14	0.23206	121.4169
8010	102.22	560.64	458.42	1.14	0.23206	121.2741

Table 10: Test 1

time (sec) (From Start)	Combust (°C)	1st pass(°C)	2nd pass(°C)	Exhaust (°C)	M(kg/s)	$Q_{comb\ used}$ kW
7680	862.28	448.98	382.02	315.9	0.509204	317.1694
7710	857.47	449.85	382.25	316.19	0.508953	314.0543
7740	856.31	450.91	382.77	316.66	0.508548	312.859
7770	859.56	451.73	383.39	317.17	0.508108	314.1757
7800	856.56	453.09	383.52	317.33	0.50797	312.2607
7830	849.47	453.59	383.59	317.42	0.507893	308.0559
7860	840.71	453.57	383.47	317.55	0.507781	302.8419
7890	834.92	454.09	383.61	317.76	0.507601	299.2623
7920	832.09	454.72	383.88	317.85	0.507523	297.5272
7950	828.01	455.2	383.8	317.98	0.507412	295.0265
7980	820.11	455.7	383.28	317.88	0.507498	290.5638
8010	811.47	456.94	382.68	317.81	0.507558	285.6395

Test 1 Results

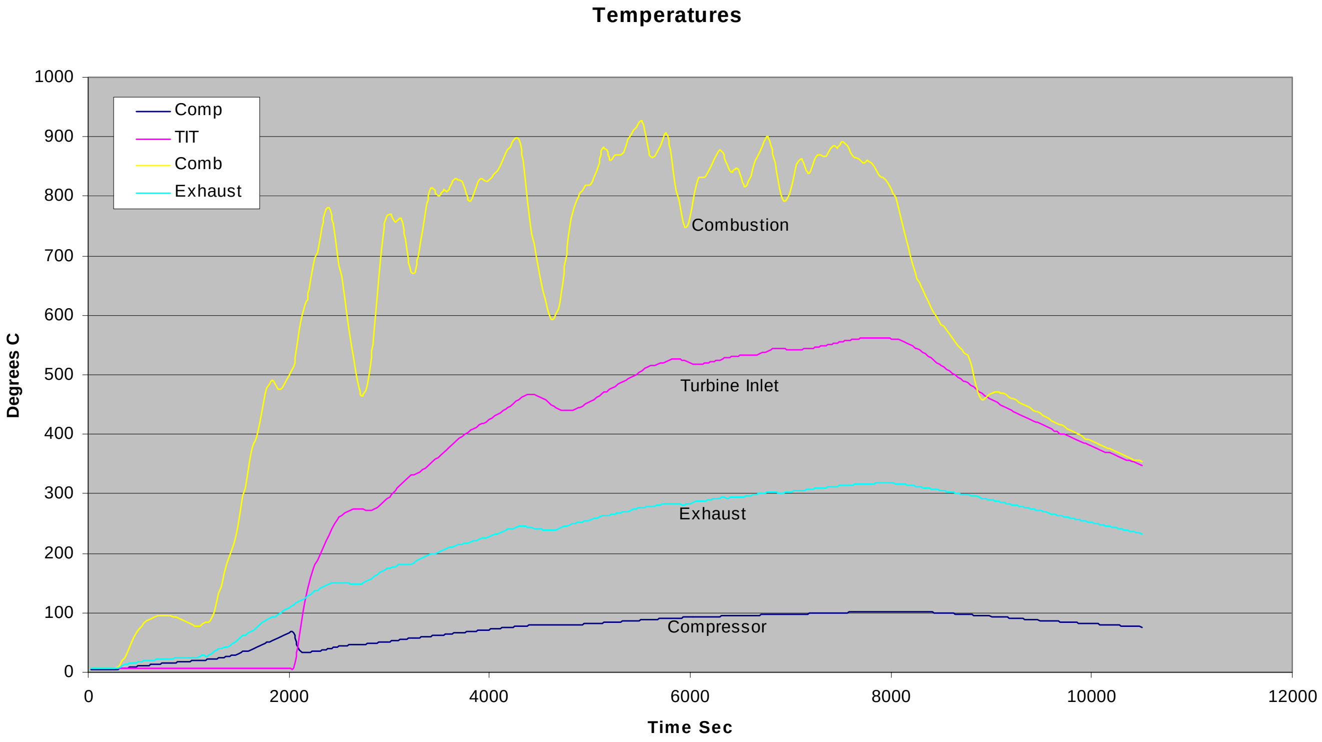


Figure 5.1.1

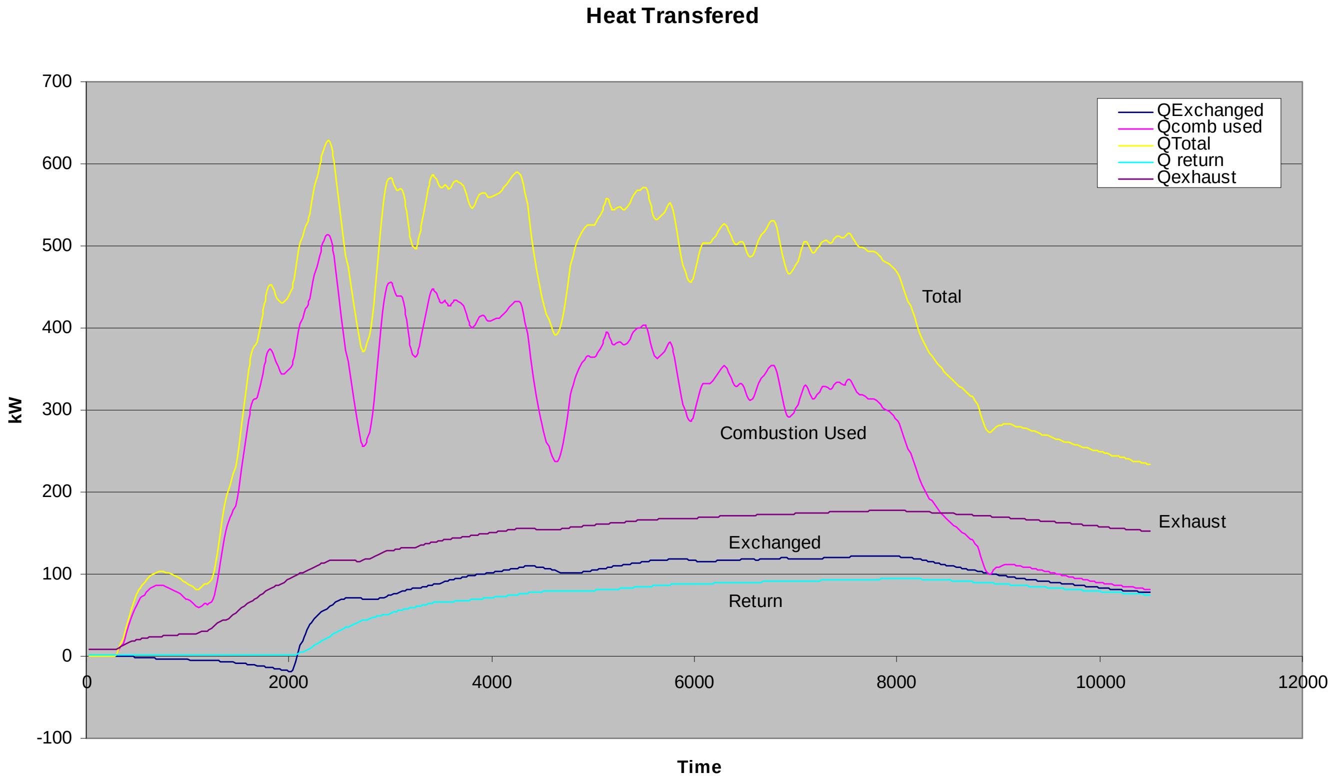


Figure 5.1.2

Test 1 Results

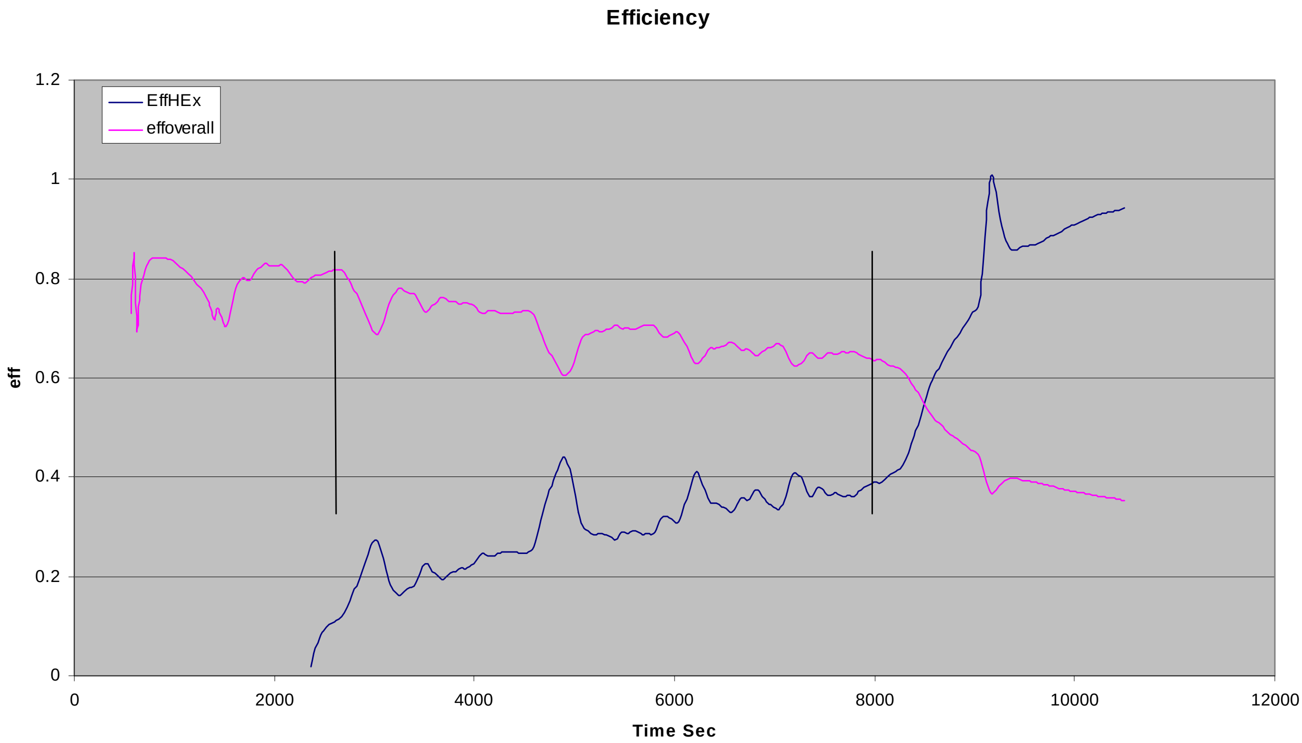


Figure 5.1.3

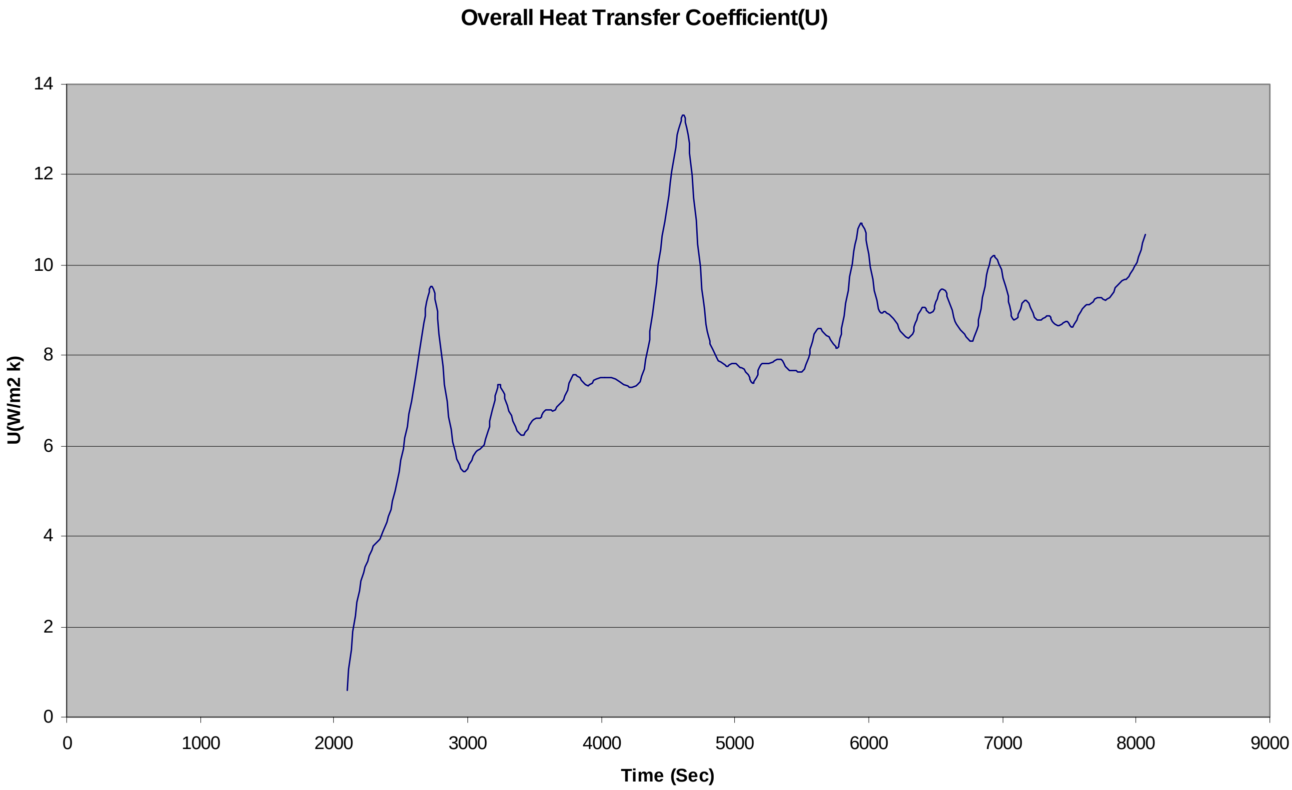


Figure 5.1.4

5.2 Second Heat Exchanger performance test after modification

A second test was performed, after small modifications to the internal geometric layout. Unfortunately during this test the rotary valve jammed, due to incorrectly prepared fuel. This failure meant combustion temperature control was erratic. This test should really be repeated, but this is difficult due to time restraints. Had the test been repeated, the results would have illustrated a small improvement in performance over test one

Table 11: Test 2

Time(sec) (From Start)	Air in (^oC)	Air out (^oC)	ΔT (^oC)	Cp kJ/kg ^{o} K	M (kg/s)	$Q_{Exchanged}$ kW
9390	115.54	614.51	498.97	1.14	0.224104	127.4762
9420	115.82	617.45	501.63	1.14	0.223943	128.0635
9450	116.12	619.41	503.29	1.14	0.22377	128.3882
9480	116.36	620.31	503.95	1.14	0.223632	128.4773
9510	116.46	620.37	503.91	1.14	0.223575	128.4341
9540	116.51	619.66	503.15	1.14	0.223546	128.224
9570	116.58	618.39	501.81	1.14	0.223506	127.8595
9600	116.57	616.99	500.42	1.14	0.223512	127.5086
9630	116.51	615.35	498.84	1.14	0.223546	127.1256
9660	116.43	613.47	497.04	1.14	0.223592	126.6929
9690	116.26	611.34	495.08	1.14	0.22369	126.2484
9720	116.15	608.91	492.76	1.14	0.223753	125.6923

Table 12: Test 2

time(sec) (From Start)	Combust (^oC)	1st pass(^oC)	2nd pass(^oC)	Exhaust (^oC)	M(kg/s)	$Q_{comb\ used}$ kW
9360	1048.71	331.39	304.24	346.92	0.483724	386.9988
9390	1016.45	331.7	304.74	346.38	0.484146	369.8291
9420	983.52	331.36	305.24	346.11	0.484357	351.9566
9450	955.83	331.22	305.76	345.87	0.484545	336.9303
9480	931.09	331.27	306.29	345.46	0.484866	323.7053
9510	910.49	331.71	306.73	345.29	0.484999	312.4986
9540	901.96	332.4	307.22	345.4	0.484913	307.6668
9570	893.24	332.53	307.56	345.1	0.485148	303.1593
9600	877.6	332.54	307.94	344.73	0.485439	294.8904
9630	861.78	332.83	308.45	344.33	0.485753	286.5425
9660	846.96	333.64	308.84	343.9	0.486092	278.7681
9690	833.26	334.28	309.11	343.44	0.486455	271.6338

Test 2 Results

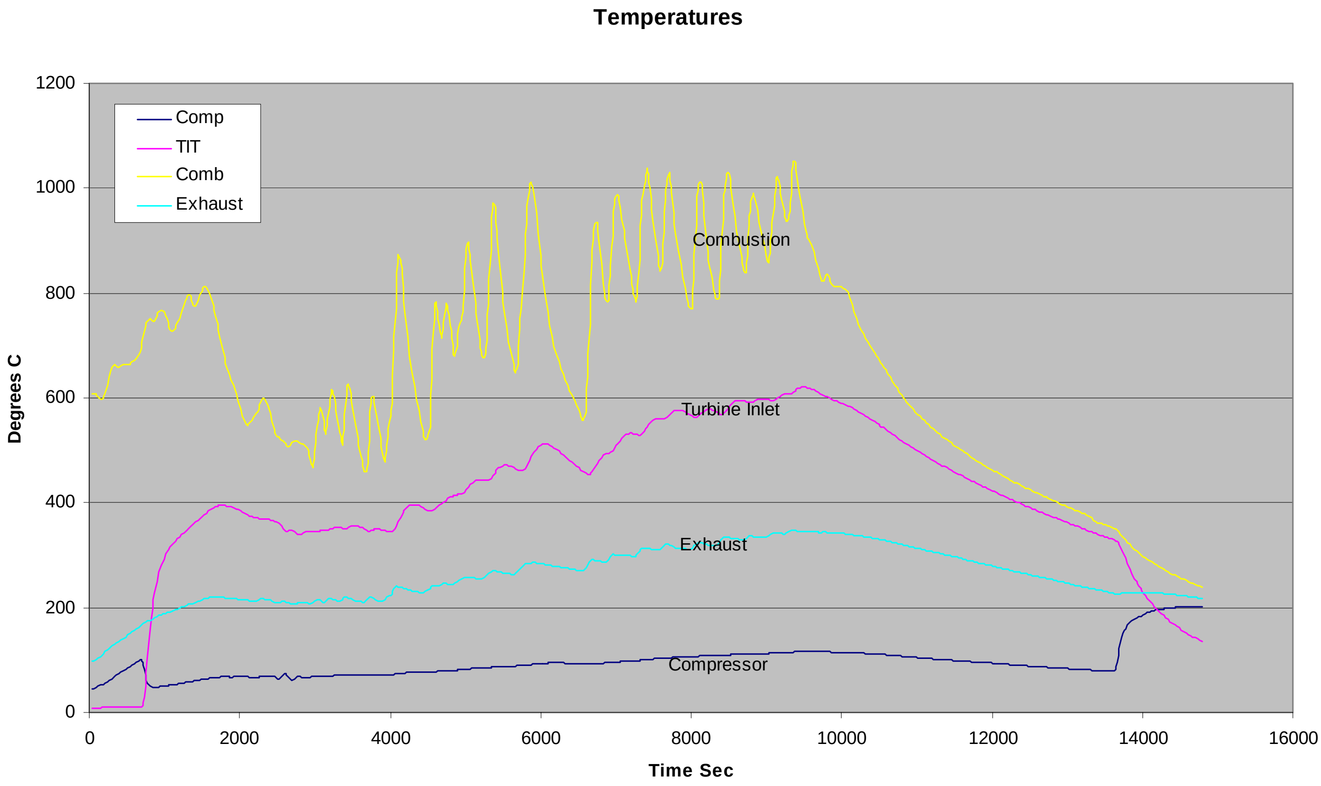


Figure 5.2.1

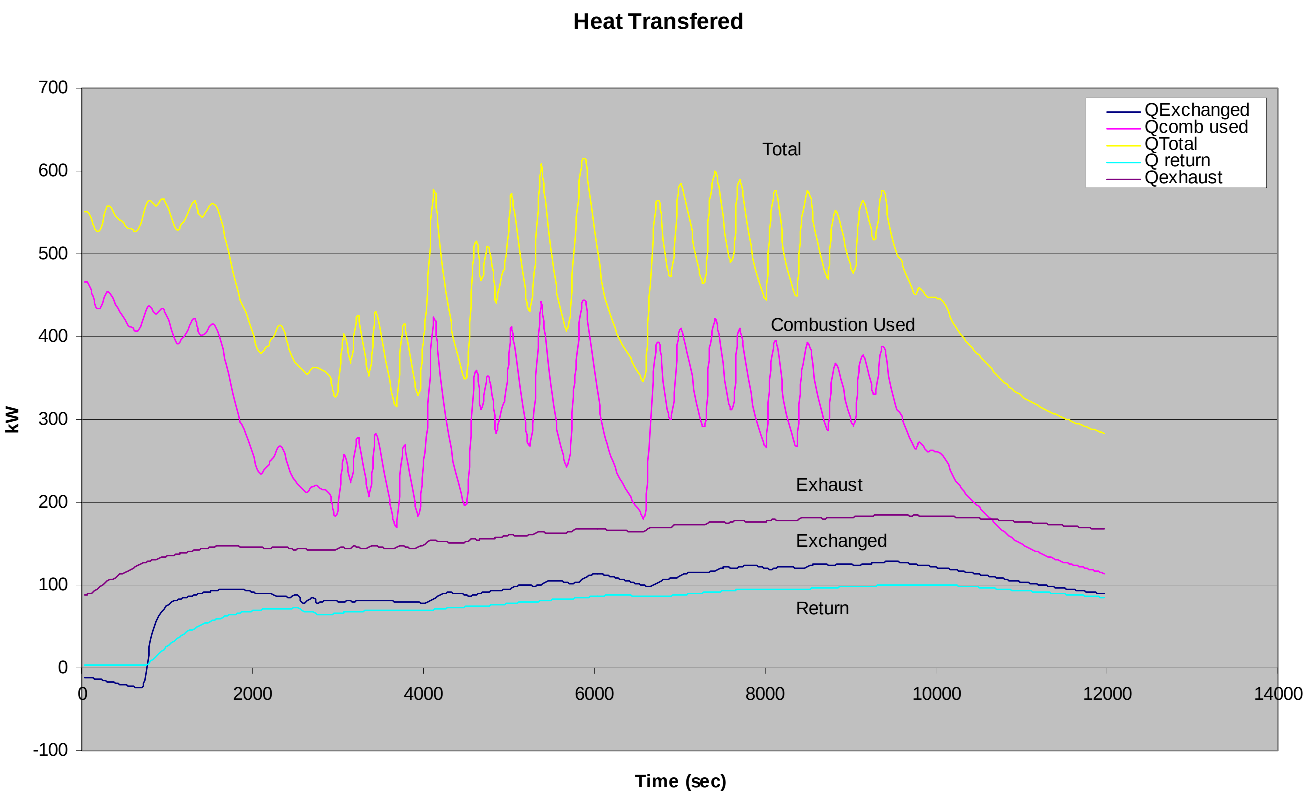


Figure 5.2.2

Test 2 Results

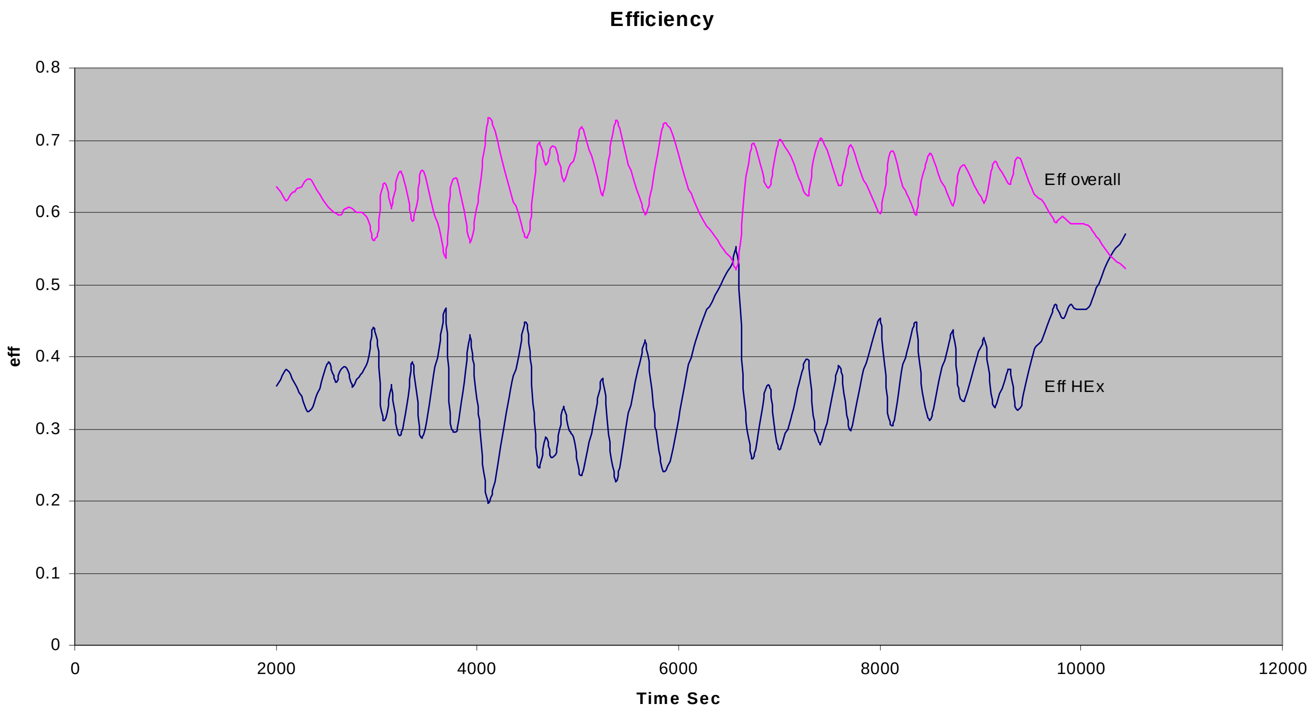


Figure 5.2.3

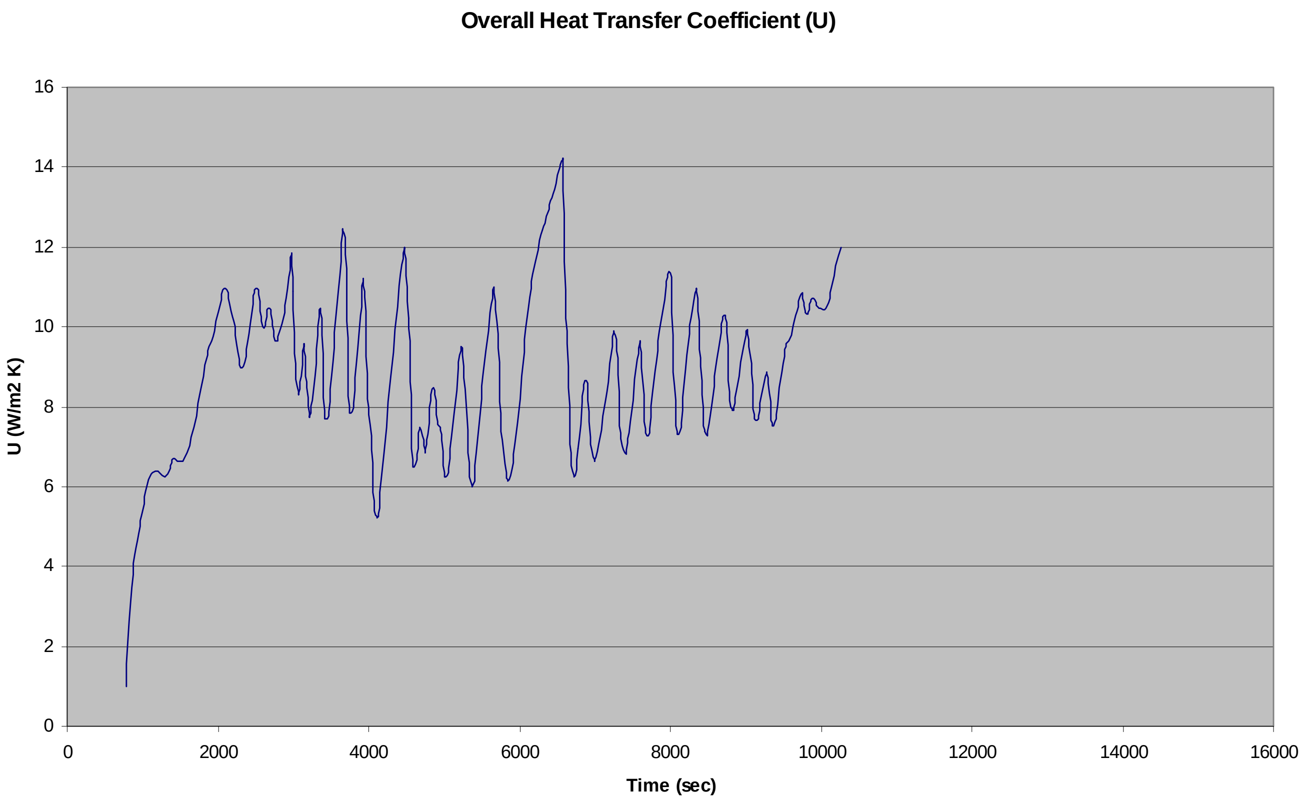


Figure 5.2.4

5.3 Micro Gas Turbine

This test was conducted to provide a benchmark and an understanding of the performance of TG50 micro-turbine. A ModBus based computer system recorded and logged data every 5sec. The turbine entry temperature is higher than the Biomass Generator system, hence higher power output.

Figure 5.3.1: Bowman Power's TG50 on test

Below is a sample of the results.

Table 13: Gas Turbine Sample Results

Time (Sec)	Compression Temperature ^oC	Comp Pressure Bar g	TeT ^oC	EG Press Bar g	Engine Speed	EGT ^oC	W_c kW	W_t kW	Total Power Generated
5	226.8	4.63	897	0	105220	607	114.1	158.7	44.6
10	226.6	4.64	898	0	105010	608	113.7	158.3	44.6
15	226.8	4.6	900	0.01	104890	610	113.7	158.1	44.4
20	227.1	4.65	903	0	104890	612	113.9	158.5	44.6
25	227.1	4.64	902	0.01	104940	611	114.0	158.4	44.4
30	227.3	4.63	903	0.01	105010	612	114.1	158.5	44.4
35	227.5	4.64	903	0	105010	612	114.3	158.5	44.2
40	227.2	4.63	904	0	104870	613	113.9	158.4	44.5

45	227.5	4.65	904	0	104990	613	114.2	158.5	44.3
50	227.4	4.61	905	0	105000	614	114.2	158.5	44.3
55	227.5	4.65	905	0	104880	614	114.1	158.5	44.4
60	227.5	4.65	903	0	104930	612	114.2	158.5	44.3
65	227.3	4.63	905	0	104900	614	114.0	158.3	44.3
70	227.6	4.63	906	0	104910	615	114.2	158.5	44.3
75	227.4	4.63	906	0.01	105060	615	114.3	158.8	44.5
80	227.1	4.62	905	0	104840	614	113.8	158.4	44.6
85	227.5	4.64	905	0	105020	614	114.3	158.9	44.6
90	227.5	4.62	906	0	105080	615	114.3	158.9	44.6
95	227.6	4.63	906	0	104950	615	114.3	158.8	44.5
100	227.5	4.65	905	0	104840	614	114.1	158.6	44.5
105	227.6	4.62	905	0	104860	613	114.2	158.9	44.7
110	227.7	4.64	906	0.01	105080	615	114.4	158.6	44.2
115	227.4	4.64	907	0	104990	615	114.2	159.0	44.8
120	227.6	4.65	906	0.01	104900	615	114.2	158.7	44.5
125	227.7	4.61	907	0	105040	615	114.4	159.1	44.7
130	227.9	4.65	905	0	104990	614	114.5	158.9	44.4
135	227.5	4.64	906	0	104960	615	114.2	158.7	44.5
140	228	4.63	906	0	105090	615	114.6	159.0	44.4
145	227.5	4.65	903	0	105120	612	114.4	158.8	44.4
150	227.5	4.66	906	0.01	105050	615	114.3	158.9	44.6

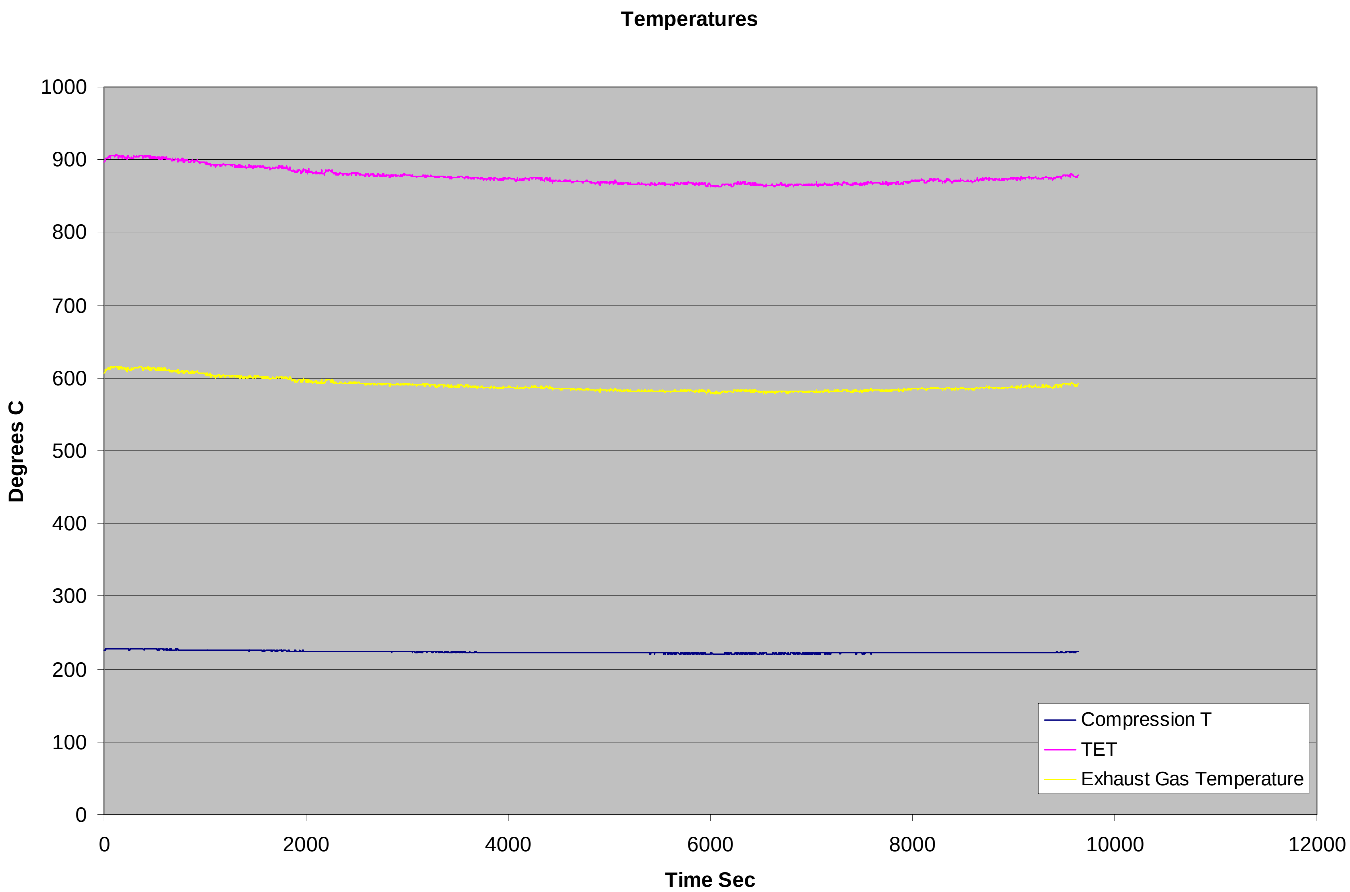


Figure 5.3.2

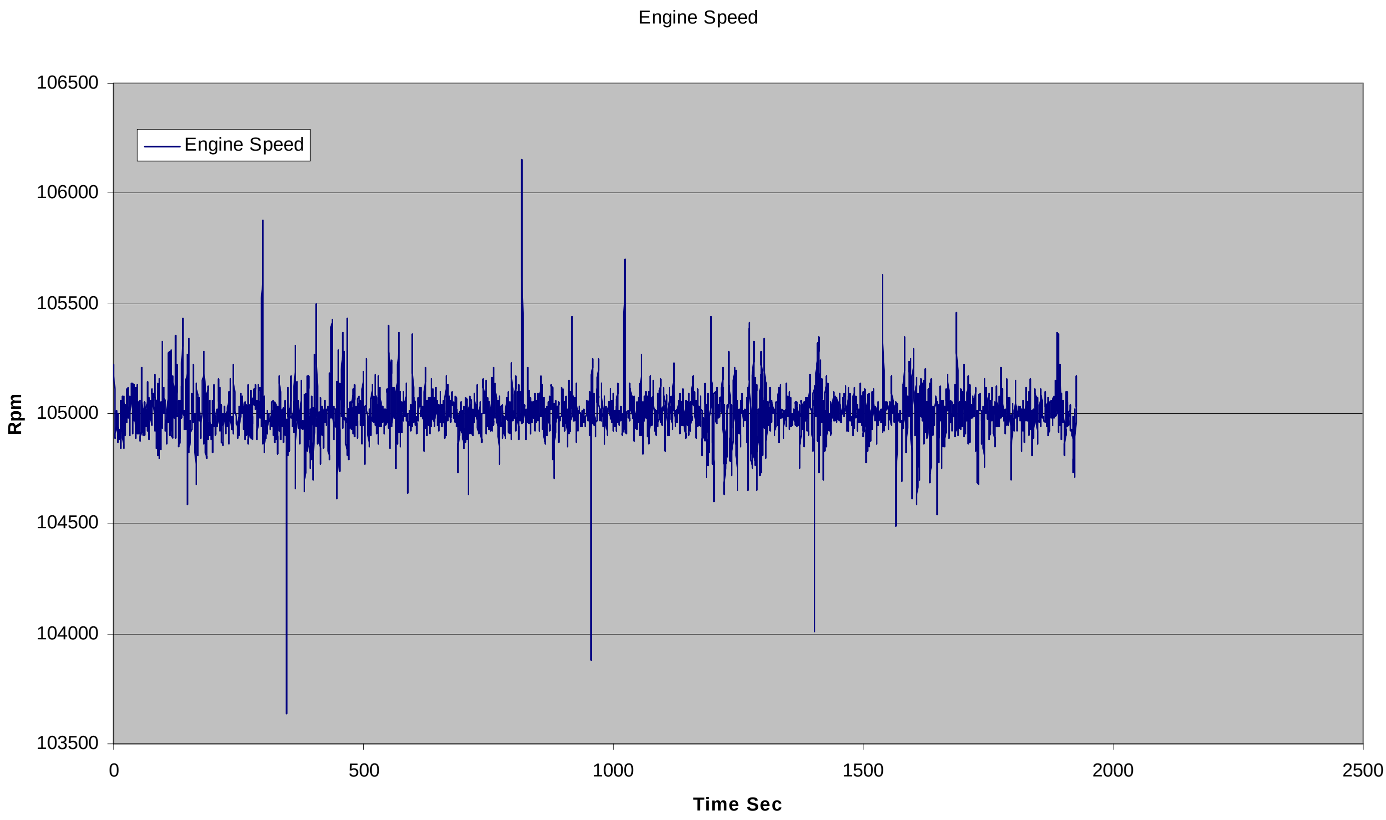


Figure 5.3.3

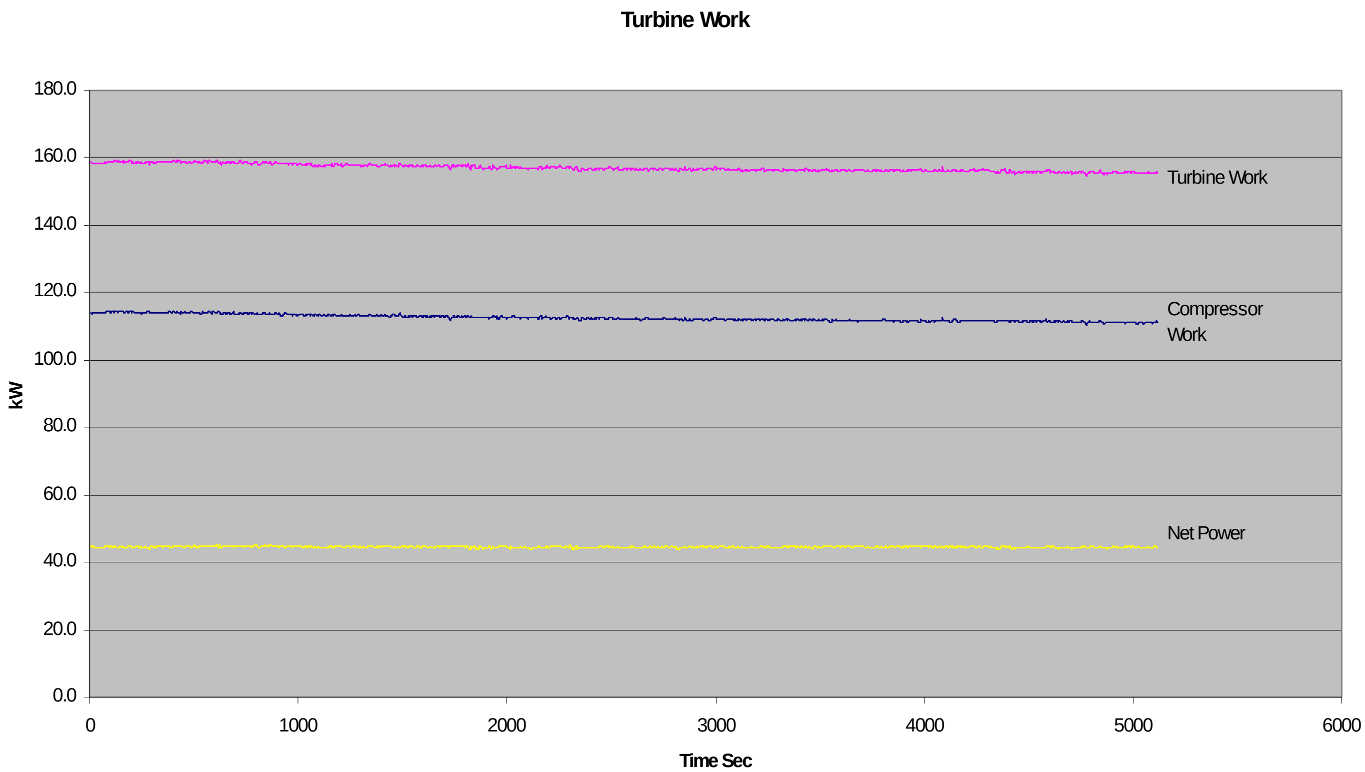


Figure 5.3.4

5.4 Initial Biomass Generator performance test

This test was the first with the biomass generator connected. Software was still being developed, based on results from these trials. Eight channels of temperature recording were captured as the combustor was lit, warmed up and run for 8hrs then shut down. Compressor air was initially off, then run up to 34,000rpm for approx. 15min. Data was recorded in real time, with measurements taken every 30sec, heat transferred is in kW. System was unable to maintain speed without electrical assistance. Biomass was providing part of the driving force, but not enough to hold speed unaided. Following the test, leaks were detected in some of the connections. These leaks are considered the major cause of the generator not self-sustaining.

Below is a sample of the results.

Table 14: Biomass Generator Sample Results

Time(sec) (From Start)	Air in (°C)	Air out (°C)	ΔT (°C)	Cp kJ/kg °K	M (kg/s)	$Q_{Exchanged}$ kW
8400	54.34	606.33	551.99	1.14	0.266003	167.3873
8430	54.41	607.12	552.71	1.14	0.265946	167.5698
8460	54.56	607.95	553.39	1.14	0.265824	167.6991
8490	54.75	608.66	553.91	1.14	0.26567	167.7594
8520	54.99	609.34	554.35	1.14	0.265476	167.7698
8550	55.18	609.85	554.67	1.14	0.265322	167.7695
8580	55.3	610.17	554.87	1.14	0.265225	167.7686
8610	55.41	610.47	555.06	1.14	0.265136	167.7699
8640	55.54	610.8	555.26	1.14	0.265031	167.7639
8670	55.54	611.02	555.48	1.14	0.265031	167.8304
8700	55.48	611.15	555.67	1.14	0.26508	167.9184
8730	56.54	607.96	551.42	1.14	0.264227	166.0981

Table 15: Biomass Generator Sample Results

time(sec) (From Start)	Combust (°C)	1st pass(°C)	2nd pass(°C)	Exhaust (°C)	M(kg/s)	$Q_{comb\ used}$ kW
8400	955.84	384.85	366.26	389.76	0.452457	291.9844
8430	951.31	385.59	366.48	389.66	0.452525	289.7431
8460	946.1	385.41	366.88	389.76	0.452457	286.9605
8490	941.53	386.18	367.32	389.97	0.452313	284.4048
8520	940.88	387.58	367.46	390.17	0.452177	283.8809
8550	940.04	388.46	367.57	390.27	0.452109	283.3536
8580	938.98	388.38	367.87	390.27	0.452109	282.8073
8610	937.13	388.67	368.16	390.24	0.452129	281.882
8640	934.27	388.71	368.33	390.3	0.452088	280.3516
8670	931.26	388.62	368.43	390.3	0.452088	278.8003
8700	929.32	389.35	368.87	390.26	0.452116	277.8378
8730	925.46	391.98	369.44	391.17	0.451496	275.002

Test 4 Results

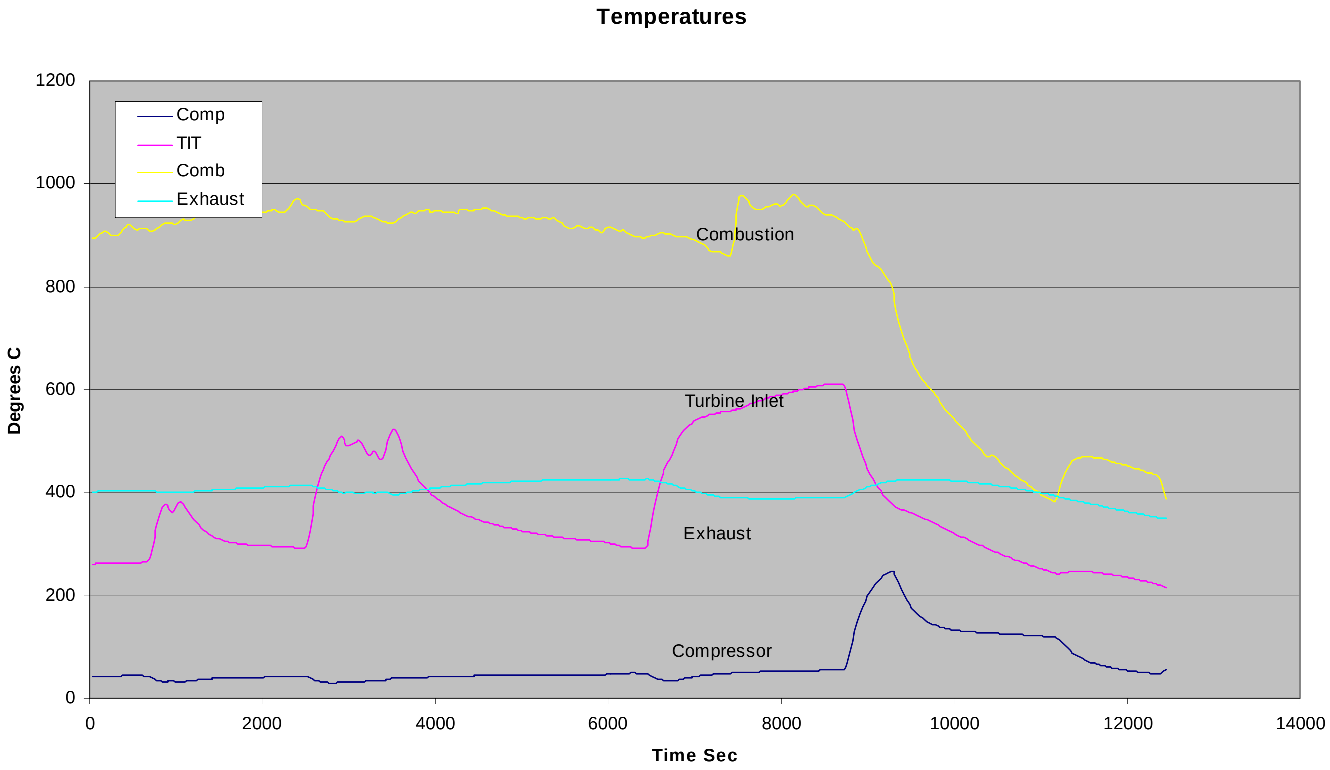


Figure 5.4.1

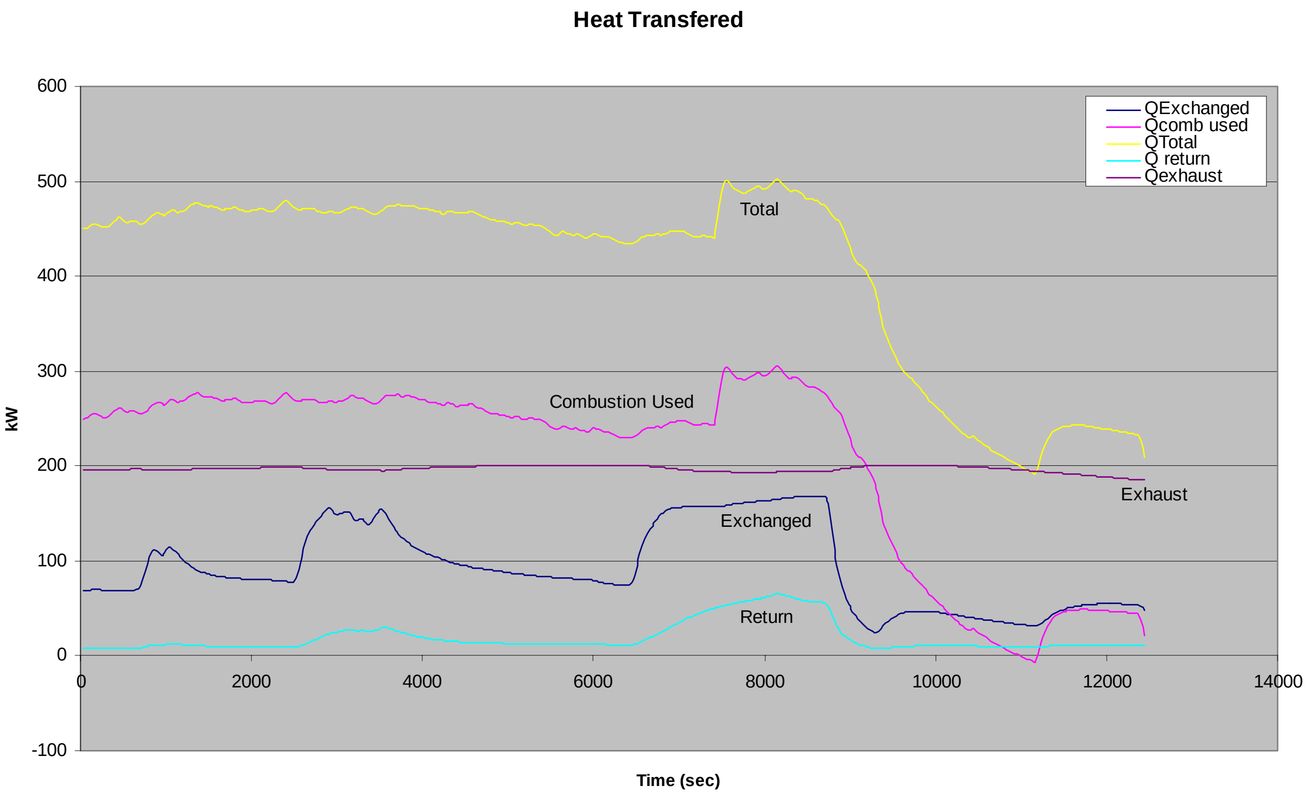


Figure 5.4.2

Test 4 Results

Figure 5.4.3

Figure 5.4.4

5.5 Second Biomass Generator Performance Test

The aim of this test was to accelerate the turbine up to generating speed (100,000rpm) and to prove the operation of the over speed protection pressure relief valve. Also we wish to gain data on system performance under no-load conditions.
8 channels of temperature recording were captured as the combustor was lit, warmed up and run for 8hrs then shut down. Data was recorded in real time, with measurements taken every 30sec, heat transferred in kW.

The over speed protection pressure relief valve was initially open, bypassing the turbines power rotor. The starter motor/generator electrically accelerated the turbine to 31,000rpm until the heat exchanger heat the turbine entry air to 630^{o}C. On reaching this temperature, the over speed protection pressure relief valve was closed. This began the chemical acceleration phase.

It was believed by the turbine manufacture that the system would self sustain at 40,000rpm then accelerate to generation speed. The turbine entry temperature slowly rose to 690^{o}C but the turbine only accelerated to just under 40,000rpm with no sign of further acceleration.

Efforts were made to increase heat input, in order to push the turbine into is self sustain and acceleration modes. We requested a higher electrical starting speed, but the Bowman engineers were concerned that the alternator/starter motor may be damaged by electrically accelerating to a speed much higher than normal. Also the alternator/starter was not designed to run in motor mode for as long as we required. Our heat transfer system however requires a good mass flow, to transfer the energy.

It was decided to increase the motor speed in small increments. Software changes were made to increase electrical acceleration on the turbine. After a few iterations in the software, the speed was increased to 41,500rpm. At this speed the turbine self sustained and accelerated strongly past its critical speeds to 100,000rpm. Turbine inlet temperature increased to 736^{o}C. On reaching generation speed the unit was stopped by operating the over speed protection pressure relief valve via the turbine controls.

Measured pressure differential across the heat exchanger, recorded 100mbar. Combustion gas pressure losses were 20mbar through the heat exchanger.

Test 5 Results

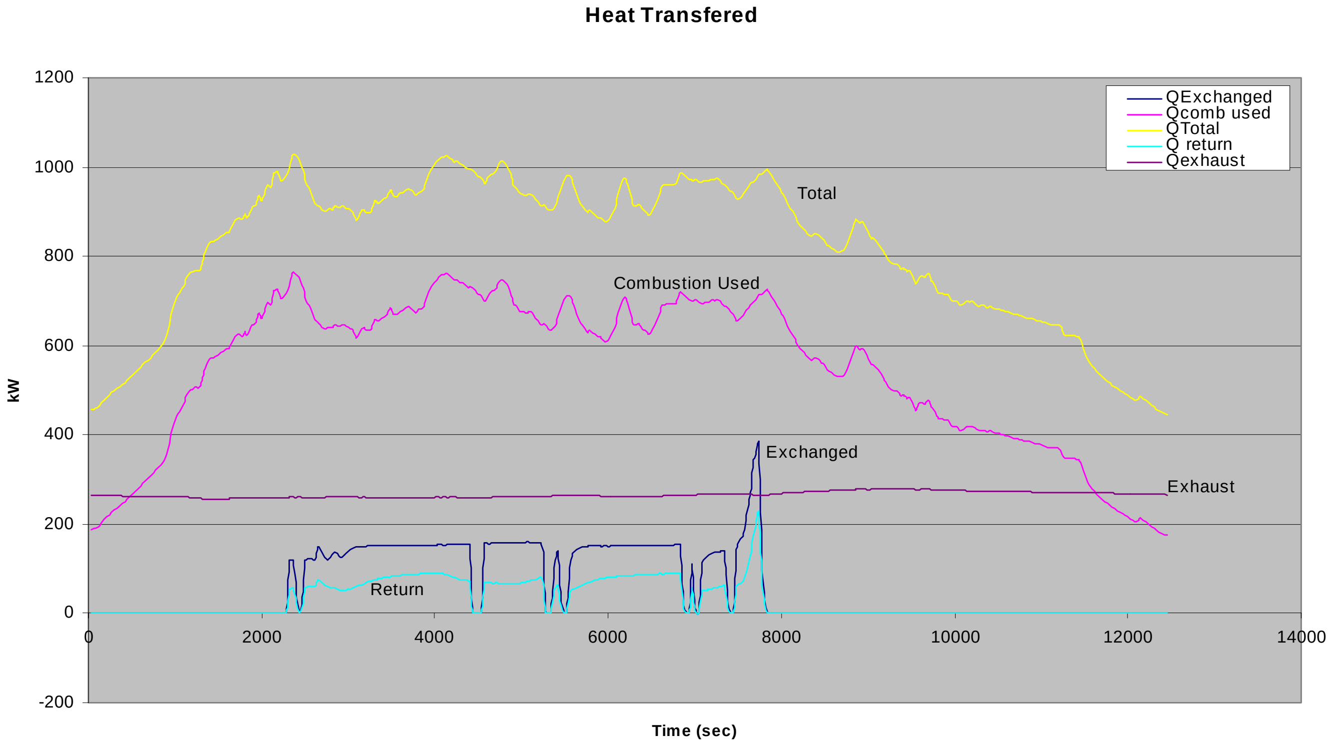


Figure 5.5.1

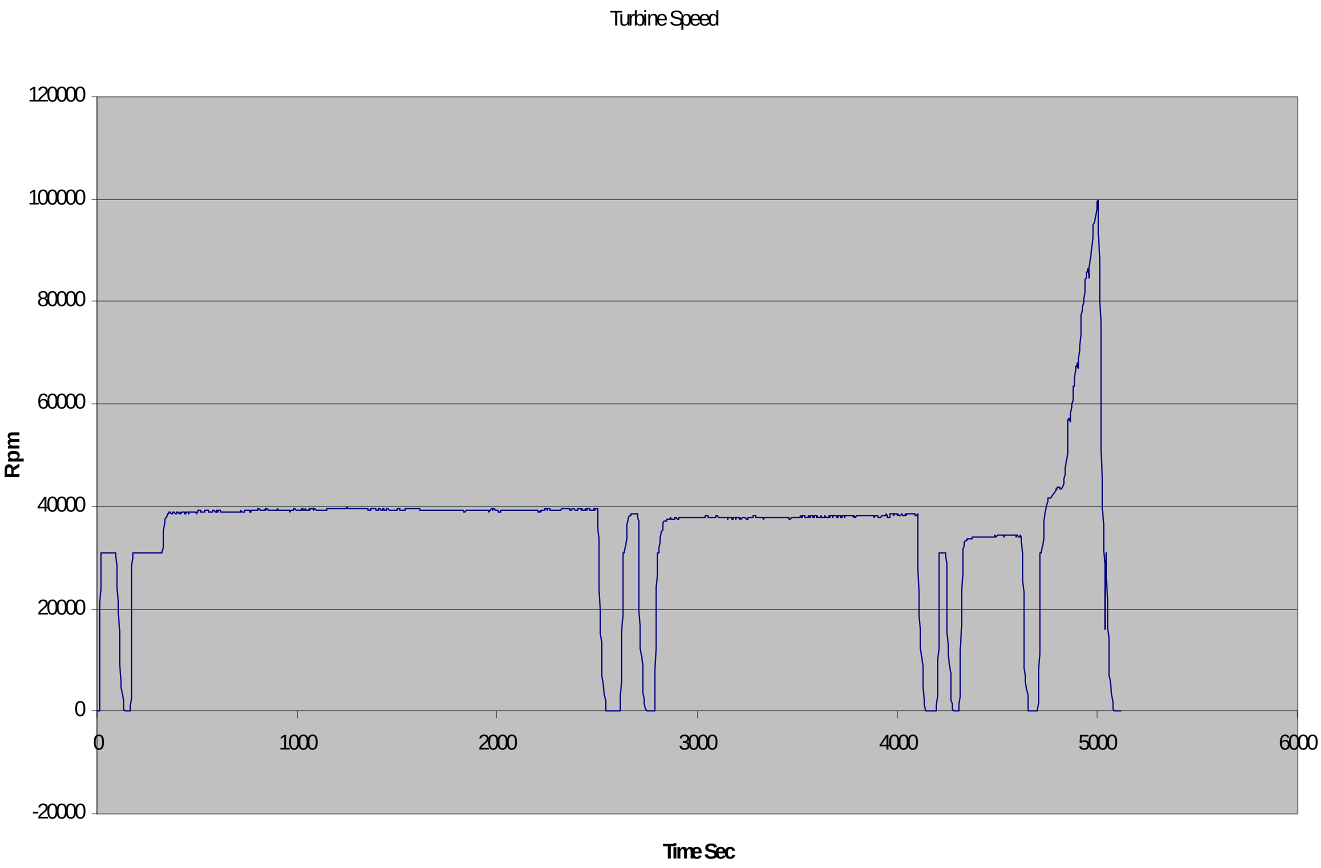


Figure 5.5.2

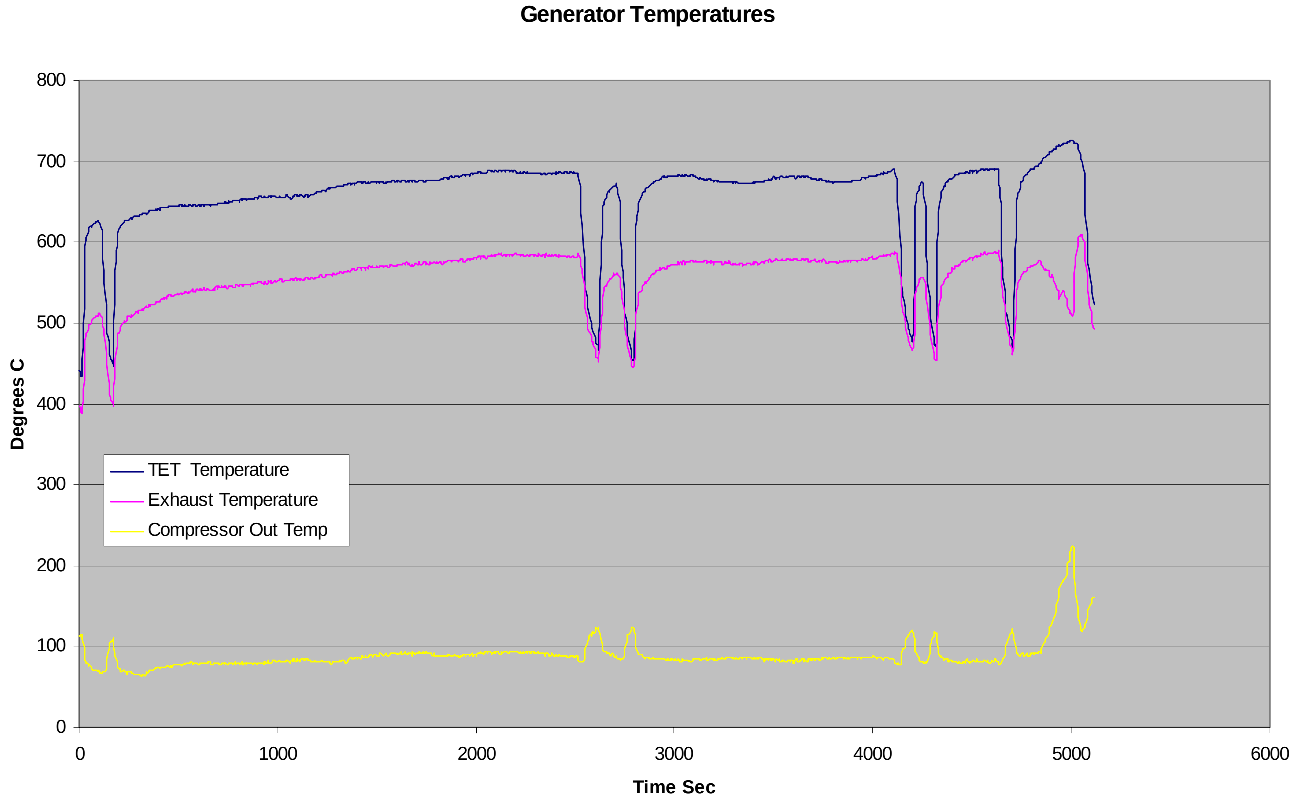


Figure 5.5.3

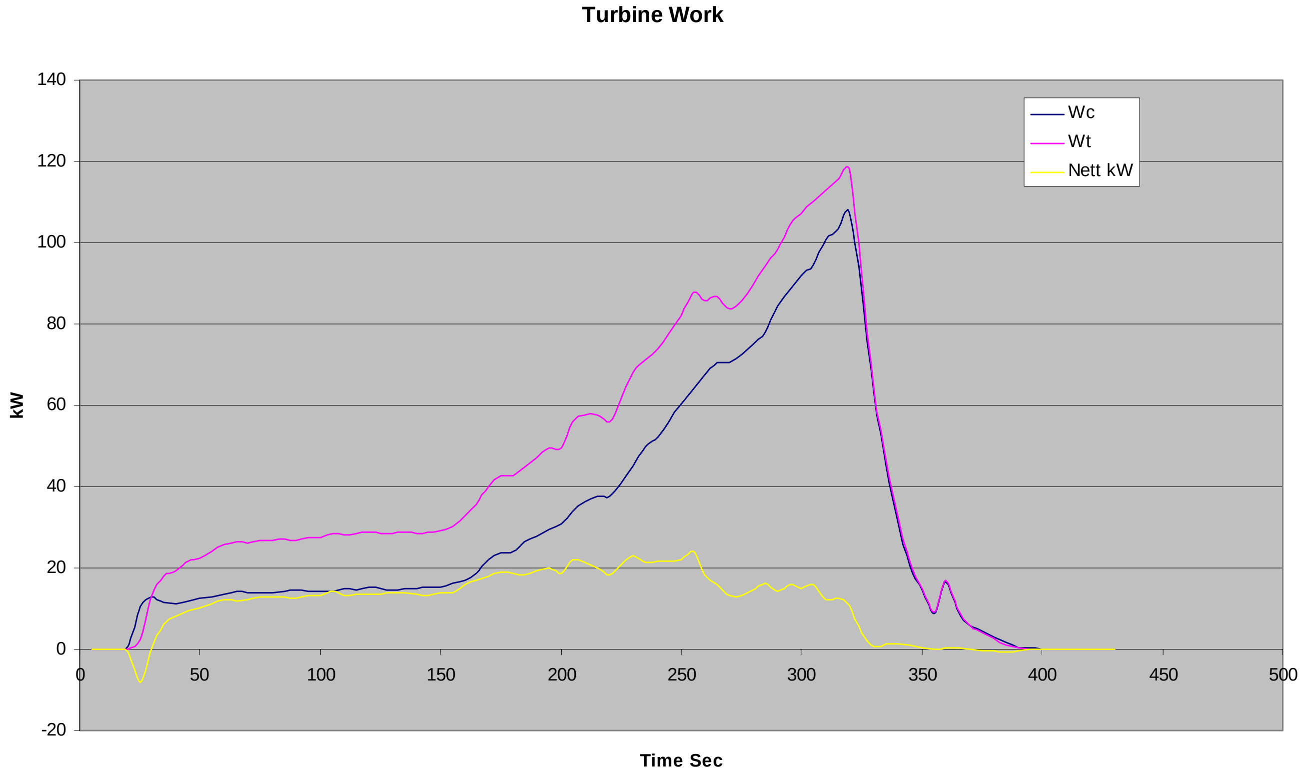


Figure 5.5.4

6.0 DISCUSSION

Test 1
From the temperature Vs time graph it can be seen that combustion temperature oscillation were high. However the effect on the air outlet temperature was damped considerably. This demonstrates the system is very slow to react to change.

It can also be seen that, the trend for the system is a slow continual rise in all temperatures. The highest air outlet temperature was 562^{o}C. It is very likely that the system had not reached steady state, before it was shut down. Therefore values recorded will not reflect the ultimate performance possible.

Thermal output from the C3 combustor was an average 500kW, the sum of the exhaust and utilised combustion thermal energy equates well to thermal output. The transferred thermal energy to the air stream slowly rose to 121kW. Heat transfer coefficient also slowly increased. This transferred thermal energy equates well to the original design of 120kW

From the energy balance it can be seen that more energy was going in the exchanger that coming out. The heat exchanger is heavily insulated and observations were made that the surface temperature of the exchanger was below 30^{o}C. Therefore thermal energy must still be going in to heating up the metal work.

Test 2

On this test the fuel feed systems rotary valve, jammed due to over sized fuel fragments. To continue the test fuel was loaded manually this however caused large oscillations in the combustion temperature.

It can be seen that, the trend for the system is a slow continual oscillatory rise in all temperatures. The highest air outlet temperature was 620^{o}C. Again it is very likely that the system had not reached steady state, before it was shut down. Therefore values recorded will not reflect the ultimate performance possible.

Thermal output from the C3 combustor was an average 540kW, the sum of the exhaust and utilised combustion thermal energy equates well to thermal output. The transferred thermal energy to the air stream slowly rose to 128kW. Heat transfer coefficient also slowly increased. This transferred thermal energy is slightly higher than the original design of 120kW.

Comparing with test one the heat exchanger modifications have yielded a small improvement in performance.

Test 3

The gas fired micro turbine was started, and run for and extended period to gain and understanding of its operation and data to compare with biomass generator system.

The turbine start sequence is very quick with some models being able to start within one minute, although very fast starts have a detrimental effect on turbine life due to thermal shock.

Gas fired micro-turbines receives a higher turbine entry temperature therefore power generated is higher, than can be expected from a biomass system.

The Bowman Power TG50 has software designed to protect and control the system from start up through generation and including shut down. This software has naturally been written for controlling a gas fired system, therefore some areas are not appropriate for a biomass-fired system.

Exhaust gas heat from this system was recovered in two stages. Stage one is a recuperator, this takes heat from the exhaust to preheat the compressed air before it enters the turbines combustor. This recuperator reduces the fuel consumption. Stage two takes the left over heat from the recuperator to produce hot water for heating. Excess heat from the oil is also used to heat the water.

Test 4
From the temperature Vs time graph it can be seen that combustion temperature oscillation were high. However the effect on the air outlet temperature was damped considerably. This demonstrates the system is very slow to react to change.

It can also be seen that, the trend for the hot air output temperature is similar to an inverse exponential, with a fast initial acceleration reducing to a slow continual rise. The highest air outlet temperature was 610^{o}C @ 34,000rpm with compressor preheat absent. Compressor work should add 150-200^{o}C onto this temperature. It is very likely that the system had not reached steady state, before it was shut down, although temperatures would not have risen much higher.

Thermal output from the C3 combustor was an average 500kW, the sum of the exhaust and utilised combustion thermal energy equates well to thermal output. The transferred thermal energy to the air stream slowly rose to 167kW. Heat transfer coefficient also slowly increased.

From the energy balance it can be seen that more energy was going in the exchanger that coming out. The heat exchanger is heavily insulated and observations were made that the surface temperature of the exchanger was below 30^{o}C. Therefore thermal energy must still be going in to heating up the metal work.

This test yielded an improvement in heat transfer over previous tests. This could be due to a longer warm up period, and hence less loss to metalwork.

System was unable to maintain speed without electrical assist. Electrical input accelerated the turbine to 15,000rpm. The heat from Biomass was provided part of the driving force slowly accelerating from 15,000rpm up to 34,000rpm. It was noted that zero power was needed at times with the current to the starter motor oscillating from 0-3A. Following the test, leaks were detected in some of the connections. These leaks are considered the major cause of the generator not self-sustaining. Compressor preheat was also absent, again this was due to leaks, no pressure increase equals no temperature increase.

From this experiment it can be seen that the system is very slow acting and cannot be expected to be operated in the same way as a gas turbine. Much longer acceleration times must be allowed for. This however causes issues with acceleration pasted resonant frequencies. Dwelling too long in a resonant frequency will cause damage to the turbine.

Test 5

The start up time for this test was not as long as the others, the combustor was lit at 9am. The combustor reached its operating temperature of 900^{o}C by 10am, we then waited till the exhaust temperature reached 200^{o}C before beginning the turbine start sequence.

The over speed protection pressure relief valve was initially open, bypassing the turbines power rotor. The starter motor/generator electrically accelerated the turbine to 31,000rpm until the heat exchanger heat the turbine entry air to 630^{o}C. This stage allows the heat exchanger to begin and build up its transfer of heat. Pipework is heated and the returned waste heat aids combustion. With the valve in the bypass position, the compressor generates no pressure, therefore power consumption is reduced.

On reaching this temperature, the over speed protection pressure relief valve was closed. This began the chemical acceleration phase. The mass of air now at a low pressure (due to lower running speed) is expanded over the power turbine. This provides mechanical energy to drive the turbine faster. The turbine should accelerate to a speed where the energy from power turbine is greater than the energy needed by the compressor. This is known as the self sustain speed. Electrical assist is removed after as the self sustain speed is reached, and the turbine is accelerated on chemical only to generation speed.

It was believed by the turbine manufacture that the system would self sustain at 40,000rpm. The turbine entry temperature slowly rose to 690^{o}C but the turbine only accelerated to just under 40,000rpm with no sign of further acceleration. Efforts were made to increase heat input, in order to push the turbine into is self sustain and acceleration modes. Combustion temperature was steady increased in an effort to accelerate the turbine. Even with combustion temperatures in excess of 1300^{o}C the turbine would not accelerate to self sustain speed. We were putting between 600kW and 700kW into the heat exchanger, yet only 180kW was being transferred. Our heat exchanger was in "stall" requiring more mass flow to transfer the energy. We could not further increase thermal input from the combustion as the high temperatures had already begun to affect some components of our system.

We requested a higher electrical starting speed, but the Bowman engineers were concerned that the alternator/starter motor may be damaged by electrically accelerating to a speed much higher than normal. Also the alternator/starter was not designed to run in motor mode for as long as we required. Our heat transfer system however requires a good mass flow, to transfer the energy by convection. With a low mass flow rate the velocity inside the heat exchanger is low therefore low heat transfer.

It was decided after a short discussion to increase the motor speed in small increments. Software changes were made to increase electrical acceleration on the turbine. After a few iterations in the software, the speed was increased via increasing the electrical assist current.
The turbine was restarted, on reaching 31,00rpm the pressure valve was closed. This caused a small drop in speed as compressor pressure rose. The turbine then

accelerated to 41,500rpm at this point electrical assist was removed. We had reached self sustain speed, the turbine acceleration slowed but continued to slowly accelerate. Heat transfer also rose with turbine inlet temperature increasing. On reaching approximately 45,000rpm the turbine acceleration increased dramatically. The turbine surged to 57,000rpm then slowed to 56,000rpm, this was close to a critical speed.

The heat exchanger then caught up with the sudden increase in mass flow rate and acceleration continued. This cycle of strong acceleration and heat exchanger recovery continued, past further critical speeds to be generation speed of 100,000rpm. Turbine inlet temperatures continued to rise up to 730^{o}C at the full mass flow, with the heat transferred racing up to 380kW. On reaching generation speed the unit was stopped by operating the over speed protection pressure relief valve.

As the turbine was not connected to the grid, net power is calculated from the measured temperatures, pressures and mass flow rates across the compressor and turbine. This ranged from 26-34kW over the recorded test data.

The turbine should now be connected to the grid to run the system long term. This grid connection required, controlling turbine speed in our system. Regulating the fuel rate can easily control gas turbines. Our Biomass turbine system reacts too slowly therefore would be unstable in this mode.

It was noted that return air from the turbine was over powering the induced draft fan. This caused the combustion chamber to become pressurised. This pressurisation forced combustion gasses to be vented though the inspection doors and back up the feed screw, all of which is a hazard. One of the reasons this occurred is that the return air was at a higher temperature than predicted therefore had a greater volume. Further work should be conducted to ensure return gasses are controlled safely.

From the turbine acceleration response shown, combustion temperatures should be able to be reduced to increase combustor life. This should also damp out surges in the turbine acceleration, further testing will confirm this. Heat exchanger performance could be further improved by adding surface area; higher temperatures at the turbine will increase the efficiency of the system. Higher temperature materials maybe needed to strengthen the system in certain areas.

Heat transferred graph (fig 5.51), illustrate the input energy from the fuel and turbine return, and its distribution into the "Combustion Used" and "Exhaust" lines. This shows the high levels of heat input to the system that were required to compensate for the lower heat exchanger performances. High exhaust temperatures also confirm the lower heat exchanger performance. Comparing the combustion heat input to the heat exchanger and the heat exchanged, it can be seen that there is a large amount of heat energy being absorbed into the metal work of the heat exchanger. For a realistic energy balance the system should be run for several days at generating speed, before capturing data.

Therefore results from actual thermal efficiency are misleading and will not represent the finished product. Further testing must be completed before publishing these figures. Theoretical efficiency based on recorded turbine temperatures was calculated to 17% overall. A larger heat exchanger and combustor air capacity will achieve these results.

On inspection of the biomass combustor after the system had cooled, it was noted that high temperatures caused distortion of heat exchangers metallic sections causing leakage paths to be created. These leakages and bypasses will reduce the directed flow, therefore reducing heat transfer. Combustor grates and feed system remained in good condition despite being subjected to temperatures far in excess of their specification. The firebricks showed signs of the high combustion temperature environment. A glazed appearance was evident in that silicon from within the bricks was brought to the surface. The glazed firebricks are still serviceable and are now harder than before.

Commercial analysis

Fuel costs were £100 per tonne, which equates to 2p per kW of heat input, this fuel is expensive but provides uniform results. Miscanthus grass is currently available in small quantities at £25 per tonne, which equates to 0.5p per kW of heat input. Other biomass fuels, such as MDF dust and wood working waste, cost around £50 per skip to be disposed of, therefore provide a free source of energy.

Although the system is still developing, and could benefit from an increase in both combustor and heat exchanger size, capital equipment costs for this 30kW_e prototype are around £2,500 per kW_e. Comparing this with our current steam based 50kW_e CHP system with an electrical efficiency of 8% and cost of £6,455 per kW_e. This represents a great leap forward at this size.

As the system size increases economies of scale improve, a 500kW_e system is expected to cost £1000 per kW_e and yield a higher efficiency. These costs are expected to fall with further development and mass production.

Commercial prospects for IFAT are good with our 3000 heat only installation, it is predicted approx. 1000 of these will be interested in producing electric from their existing waste fuel supplies. The 30kW_e unit could provide an offset to normal base load of 250kW_e average to wood working factories. With electric being purchased at 5p/kW this would save £1.50 per hr. operating this unit for 8000hrs per year will save £12,000, ROC will add another 3p/kW (although sold on the open market may yield 5p/kW), this add a further £7,200. Therefore the payback period can be calculated to 4years. Heat output will make system more attractive as well, by cutting gas or oil costs. Larger units with higher efficiencies and lower costs per kW will of course, provide much faster pay back, a 250kW_e system is expected to have a payback period of just 2 ½ years.

An area of concern is the reluctance of electricity supply companies to be interested in purchasing form small generators under 1MW_e this may dampen the market for distributed power. However they will accept a cluster of units to make up the 1$MW_{e,}$ this may require the setting up of a small generators selling group. Alternatively small generators will find uses in offsetting base loads.

7.0 CONCLUSIONS

From the above results and discussions the following conclusions may be drawn:

- Biomass Combustion Turbine project has successfully provided proof of principle
- Talbott's biomass combustor is capable of generating combustion gasses far in excess of specified 900^{o}C
- Heat exchanger performance improves with mass flow rate.
- Biomass Micro Gas Turbine acceleration is slower that conventional gas turbines.
- The system has demonstrated it is possible to produce 26-34kWe of electrical renewable energy plus recoverable thermal energy.
- Approximately 100-150kW of high-grade thermal energy is available for simple conversion to heating water or air.
- Low heat losses results in overall system efficiency between 80-85%
- Electrical efficiency is 17% and will rise with further development
- Positive pressure in the biomass combustion chamber is to be avoided.
- A partnership has being formed to further develop and investigate this system

We believe the project has been very successful and look forward to the future development of this exciting project.

8.0 RECOMMENDATIONS

It is recommended to run the system for a period of 1 year to provide data on the long-term effects on materials at the high temperatures. Maintenance and reliability issues should also be studied on this new system. The study should carefully condition monitor the system and seek to improve the overall cycle efficiency.

The biomass combustor should be redesigned to incorporate the higher air return temperatures and volume flow rates. This key area will reduce fuel consumption greatly and lead to the high efficiencies determined earlier. Other improvements could be incorporated inline with our experience of this system.

The heat exchanger could be improved with the knowledge gained through this project. High temperatures caused distortion of steel sections causing leakage paths to be created. Alternative material selections and construction details will reduce these leakages and bypasses, therefore improving heat transfer. Reducing the usage of metals will also lower the thermal mass of the system and hence speed up start up time. This factor may be of some concern to end-users. If the unit can start in 2hrs rather than 4hrs, 2hrs of fuel will be saved.

High LMTD's in the heat exchanger should be reduced by increasing the size of the surface area. The original design for the mass flow was much lower, therefore a new and larger heat exchanger would reduce combustion temperatures and increase efficiency.

The 50kWe gas turbine when adapted to run on biomass provides in the region of 26-34kW of electric power. It is also recommended to utilise waste heat from the system to be used in heating and cooling applications. Thus making an efficient biomass fired Co-gen and Tri-gen system. So instead of dumping heat in the summer, the energy maybe used to provide cooling. This cooling will offset electrically driven AC units, which will save fossil fuels that are combusted to generate electricity

The great strength of biomass is the ability to combust a wide variety of fuels. The suitability of this unit to alternative fuels should be studied in order to widen the market area for this system. Miscanthus in particular is being promoted as an energy crop and will almost certainly have a large renewables market share. Traditional rotary valve feed systems are not suitable for this fuel, therefore a miscanthus feed system should be developed for this growing market.

9.0 REFERENCES

Papers and Journal references

[1] *Title:* **Biomass and renewable fuels**
Authors: Chum, H. L.; Overend, R. P.
Author address: National Renewable Energy Laboratory, Golden, CO 80401, United States
Journal ref: Fuel Processing Technology, v 71, n 1-3, (June 2001 **2001**), p 187-195
ISSN: 0378-3820

[2] *Title:* **Re-appraisal of wood-fired combustion**
Authors: McIlveen-Wright, D. R.; Williams, B. C.; McMullan, J. T.
Author address: Univ of Ulster, Coleraine, UK
Journal ref: Bioresource Technology, v 76, n 3, (Feb **2001**), p 183-190
ISSN: 0960-8524 *Publisher:* Elsevier Science Ltd
Publisher address: Exeter, Engl

[3] *Title:* **Large-scale economic integration of electricity from short-rotation woody crops**
Authors: Varela, M.; Saez, R.; Audus, H.
Author address: CIEMAT, Madrid, Spain
Journal ref: Solar Energy, v 70, n 2, (**2001**), p 95-107
ISSN: 0038-092X
Publisher: Elsevier Science Ltd
Publisher address: Exeter, Engl

[4] *Title:* **Piston steam engine innovation**
Authors: Stokes, P. R.
Author address: Kew Bridge Steam Museum, Middlesex, Engl
Journal ref: Proceedings of the Institution of Mechanical Engineers, Part A: Journal of Power and Energy, v 210, 1, (**1996**), p 95-98
ISSN: 0957-6509 *Publisher:* MEP
Publisher address: Edmunds, Engl

[5] *Title:* **Thermophysical Properties In The Design Of Prime Movers.**
Authors: Kestin, J.
Author address: Inst for Advanced Study, Berlin, West Ger, Inst for Advanced Study, Berlin, West Ger
Journal ref: Journal of Non-Equilibrium Thermodynamics, v 11, n 1-2, (**1986**), p 1-21
ISSN: 0340-0204

[6] *Title:* **Gas turbines and spark ignition gas engines for industrial CHP**
Authors: Heap, C. C.
Journal ref: Power Engineer, v 5, n 5, (October 2001 **2001**), p 37-48
ISSN: 1367-191X

[7] *Title:* **Real-time measurement of combustion generated particles with photofragmentation-fluorescence**
Authors: Damm, C. J.; Lucas, D.; Sawyer, R. F.; Koshland, C. P.
Author address: Dept. of Mechanical Engineering University of California, Berkeley, CA, United States
Journal ref: Applied Spectroscopy, v 55, n 11, (November 2001 **2001**), p 1478-1482
ISSN: 0003-7028

[8] *Title:* **A study on a multi-stage hybrid gasifier-engine system**
Authors: Bhattacharya, S. C.; San Shwe Hla; Pham, H. -L.
Author address: Energy Program Sch. of Environment, Resources/Dvmt. Asian Institute of Technology, Klong Luang, Pathumthani 12120, Thailand
Journal ref: Biomass and Bioenergy, v 21, n 6, (**2001**), p 445-460 *ISSN:* 0961-9534

[9] *Title:* **Analysis of heat pipe system for waste energy recovery**
Authors: Mohamed, M. S.
Author address: Faculty of Engineering Cairo University, Giza, Egypt
Journal ref: Journal of Engineering and Applied Science, v 48, n 3, (June 2001 **2001**), p 525-544
ISSN: 1110-1903

[10] *Title:* **Combined cycle thermal performance diagnostic testing**
Authors: Schmitt, Thomas P.
Author address: GE Power Systems, Schenectady, NY, USA
Journal ref: American Society of Mechanical Engineers, Power Division (Publication) PWR, v 34 (2), (**1999**), p 535-545
Conference: The 1999 International Joint Power Generation Conference and Exhibition and ICOPE '99
(conference code **56952**) *Meeting date:* Jul 25-Jul 28 1999
Meeting location: Burlingame, CA, USA
Meeting sponsors: ASME, CSPE, JSME
Publisher: ASME *Publisher address:* Fairfield, NJ, USA

[11] LM2500 gas turbine fuel nozzle design and combustion test evaluation & emission results with simulated gasified wood product fuels
Authors: Neilson, Charles E.; Shafer, Douglas G.; Carpentieri, Eduardo;
Author address: General Electric Co, Cincinnati, OH, USA
Journal ref: American Society of Mechanical Engineers (Paper), n GT,(1998)
Paper No: 98-GT-337
Publisher: ASME
Publisher address: Fairfield, NJ, USA

[12] *Title:* **Mixing, modelling and measurements of incinerator bed combustion**
Authors: Goh, Y. R.; Lim, C. N.; Zakaria, R.; Chan, K. H.; Reynolds, G.; Yang, Y. B.; Siddall, R. G.; Nasserzadeh, V.; Swithenbank, J.
Author address: Univ of Sheffield, Sheffield, UK
Journal ref: Process Safety and Environmental Protection: Transactions of the Institution of Chemical Engineers, Part B, v 78, n 1, (**2000**), p 21-32
ISSN: 0957-5820 *Publisher:* IChemE
Publisher address: Rugby, Engl

[13] High efficiency microturbine technology
Authors: Doty, F. David; Miller, Brian L.; Hosford, Greg S.; Wilson, David Gordon; Huanbo, Wu; Jones, John Dewey.
Author address: Doty Scientific, Inc
Journal ref: Proceedings of the Intersociety Energy Conversion Engineering Conference , v 2
(1991) p 436-442
ISSN: 0146-955X
Publisher: IEEE
Publisher address: Piscataway, NJ, USA

[14] Micro gas turbine for combined heat and power in distributed generation
Authors: Carno, Johanna; Cavani, Adrin; Linanki, Leif of Vattenfall.
Author address: Vattenfall Utveckling AB, Stockholm
Journal ref: American Society of Mechanical Engineers (Paper), n GT,(1998)
Paper No: 98-GT-309
Publisher: ASME
Publisher address: Fairfield, NJ, USA

[15] Use of biomass for electric power generation in the South African and Zimbabwean sawmilling industry
Authors: Cochrane, E. D. D.
Journal ref: Journal of Energy in Southern Africa, v 11, n 2, (May 2000), p 191-202
ISSN: 1021-447X
Publisher: Energy Research Institute
Publisher address: Plumstead, South Africa

[16] Biomass combustion options for steam generation
Authors: Briggs, R. J.; Adams, J. M.
Journal ref: International Exhibition & Conference for the Power Generation Industries - Power-Gen, (1998), p 128
Conference: Proceedings of the 1997 International Conference on Power Generation, POWER-GEN (1998), p 128
(conference code 49202)
Meeting date: Dec 9-11 1997
Meeting location: Dallas, TX, USA
Publisher: PennWell Publ and Exhib
Publisher address: Houston, TX, USA

[17] Combined biomass and black liquor gasifier/gas turbine cogeneration at pulp and paper mills
Authors: Larson, E. D.; Kreutz, T. G.; Consonni, S.
Author address: Princeton Univ, Princeton, NJ, USA
Journal ref: Journal of Engineering for Gas Turbines and Power, Transactions of the ASME,
v 121, n 3, (1999), p 394-400
ISSN: 0742-4795
Publisher: ASME
Publisher address: Fairfield, NJ, USA

Books

[18] New and Renewable Energy,
Prospects for the 21st Century, The Renewables Obligation Preliminary Consultation.
Author: DTI (2000)

[19] Power plant Technology,
International Edition(1984),
Author: M.M.El-Wakil, McGraw-Hill
ISBN 0-07-066274-6

[20] Energy Efficiency for Engineers and Technologists(1990),
Author: T.D.Eastop & D.R.Croft, Longman Scientific & Technical.
ISBN: 0-582-03184-2

[21] Biomass Stoves: Engineering Design, Development, and Dissemination, Volunteers in Technical Assistance (VITA),
Author: Baldwin S.F, 1987
Arlington.

[22] Waste Incineration Handbook
Author: Cheremisnoff PN, 1992,
Butterworth-Heinemann, Oxford.

[23] Introduction to Combustion Phenomena
Author: Kanury AM, 1975
Gordon and Breach, New York.

[24] Experimental Evaluation of Biomass Burning Emissions: Nitrogen and Carbon Containing Compounds, in *Global Biomass Burning: Atmospheric, Climatic, and Biospheric Implications*
Author: Lobert J.M, Scharffe D.H, Hao W-M, Kuhlbusch T.A, Seuwen R, Warneck P, Crutzen P.J, 1991
Levine J.S (ed), MIT Press, Cambridge, MA, USA: 289-307.

[25] Basic Principles of Direct Combustion, in *Biomass Conversion Processes for Energy and Fuel*
Author: Shafizadeh F, 1981, *s*, Sofer S.S, Zaborsky O.R,
(Eds), Plenum Publishing Corporation, New York: 103-124.

[26] Chemistry of Pyrolysis and Combustion of Wood, in *Proceedings: 1981 International Conference on Residential Solid Fuels – Environmental Impacts and Solutions*
Author: Shafizadeh F, 1982, , Cooper JA, Malek D
eds, Oregon Graduate Center, Beverton: 746-771.

[27] Gas Processing*: Environmental Aspects and Methods*
Author: Speight JG, 1993
Butterworth-Heinemann, Oxford.

[28] Engine Exhaust Emissions, in *Engine Emissions: Pollutant Formation and Measurement*
Author: Myers P.S, Uyehora O.A, Newhall H.K, 1973,
Springer G.S, Patterson D.J (Eds.), Plenum Press, New York: 1-31.

[29] Coal Constitution — *typology, chemistry, physics*
van Krevelen DW, 1961
Elsevier Publishing Company, Gouda, The Netherlands.

Web based Links

http://www.capstoneturbine.com/
http://www.bowmanpower.com/
http://www.ingersoll-rand.com/energysystems/
http://www.parallon75.com/
http://www.britishbiogen.co.uk/
http://www.westbioenergy.org/biopub.htm
http://www.talbotts.co.uk

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APPENDICES

1.1 What is biomass?

Biomass is all plant and animal matter on the Earth's surface, such as trees, waste products and animal by-products. Biomass can be thought of as storage of solar energy in chemical form. Whether cultivated by man, or growing wild, plant matter represents a massive quantity of a renewable energy. Biomass can be combusted to provide heat then subsequently converted into other forms of energy.

Carbon dioxide from the atmosphere and water from the earth are combined in the photosynthesis process to produce carbohydrates (sugars) that form the building blocks of biomass. The solar energy that drives photosynthesis is stored in the chemical bonds of the structural components of biomass. When biomass is combusted efficiently, stored energy is released from the chemical bonds. Oxygen from the atmosphere combines with the carbon in the biomass to produce carbon dioxide and water. The carbon dioxide is then available to produce new biomass, therefore biomass is a renewable resource.

It is estimated that biomass production is about eight times the total annual world consumption of energy from all sources. At present the world population uses only about 7% of the estimated annual production of biomass.

The chemical composition of biomass varies among species, but biomass consists of about 25% lignin and 75% carbohydrates or sugars. Most species also contain around 5% of a smaller molecular fragments, called extractives. The carbohydrate fraction consists of many sugar molecules linked together in long chains or polymers. Two larger carbohydrate categories that have significant value are cellulose and hemi-cellulose. The lignin fraction consists of non-sugar type molecules linked together in large two dimensional mesh structures. The long cellulose polymers are used to build the fibers that give the plant its strength. The lignin binds the cellulose fibers together. It is this combination that gives plants their flexibility and structural strength.

In addition to the aesthetic value of plants, biomass represents a useful and valuable resource. Since the dawn of time we have exploited the solar energy stored in the chemical bonds by burning biomass as fuel and eating plants for the nutritional energy of their sugar and starch content. In the last few hundred years, we have exploited fossilized biomass in the form of coal. This fossil fuel is the result of a slow chemical transformations that convert the sugar polymer fraction into a chemical composition that resembles the lignin fraction. Thus, the additional chemical bonds in coal represent a more concentrated source of energy as fuel. Since it takes millions of years to convert biomass into coal, fossil fuels are not renewable in the time-frame over which we use them.
We have also exploited chemically useful biomass constituents. By dissolving the lignin from plants and discarding this part, cellulose fibres are recovered to make paper and textiles. After we have finished with this paper and textiles we can also convert them to energy.

The calorific value of biomass tends to lower than fossil fuel, hence the volume required for a given heat output is greater. This causes problems in economic transportation of biomass over long distances. Storage and drying are also areas of concern that need to be assessed in any evaluation.

Calorific Value comparison from published tables

Table 1: Calorific value of biomass and other fuels

Fuel	**GCV in MJ/kg**
Vegetables	6.7
Paper	14.6
Rags	16.3
Wood	17.6
Straw	18.0
Animal Waste	18.0
Rubber	34.7
Plastic	37.0
Coal	26.4
Oil	44.0
Natural Gas	52.4

Results from a our bomb calorimeter test for various biomass fuels are as follows:

Table 2 Calorific values from tests

Pencil Sharpening :	19.160 **MJ/kg**
Hard Wood chips :	18.075 **MJ/kg**
Miscanthus :	17.522 **MJ/kg**
Dried bread :	20.033 **MJ/kg**

The amount of heat that can be released from a given mass of fuel can be determined by:

Thermal Energy(Q) = Mass(M) x Calorific Value of the fuel(CV)

Biomass fuels are rarely free of water, the amount of water in fuels is know as moisture content. Moisture content is expressed a percentage of weight, 20-30% MC is acceptable in normal wood combustion. Water content lowers the calorific value of the fuel as the water must be evaporated first. This evaporation of water requires energy and causes an increase in fuel consumption for a given output.

1.2 Wood Fuel Analysis

Both the chemical and the physical composition of the fuel are important determining factors in the characteristics of combustion.
Wood can be analysed by breaking it down into structural components (called proximate analysis) or into chemical elements (ultimate analysis). Below are given average results of the two types of analyses.

Table 3 Ultimate analysis of wood ([22]Cheremisinoff 1992)

Species	Ash (%)	Extractives (%)	Lignin (%)	Hemicellulose (%)	Cellulose (%)
Softwood	0.4	2	27.8	24	41
Hardwood	0.3	3.1	19.5	35	39

Table 4 Proximate analysis of wood ([25]Shafizadeh 1981)

Carbon (%)	Hydrogen (%)	Oxygen (%)	Nitrogen (%)	Sulphur (%)	Non-combustibles
50.5	6.0	42.4	0.2	0.05	1.0

Moisture
Moisture in wood (and other biofuels) is stored in spaces within the dead cells and within the cell walls. When the fuel is dried the stored moisture equilibrates with the ambient relative humidity. Equilibrium is usually about 20% in air dried fuel.
The moisture in the fuel acts as a heat sink, lowering the combustion efficiency. It also promotes the formation of carbonaceous char (that is char, water and carbon dioxide) ([25]Shafizadeh 1981).

Inorganic Materials (Ash)
Inorganic materials in plants depend on the type of the plant and the soil contamination in which the plant grows. On average wood contains about 0.5% ash. Insoluble compounds act as a heat sink in the same way as moisture, lowering combustion efficiency, but soluble ionic compounds can have a catalytic effect on the pyrolysis and combustion of the fuel. The presence of inorganic compounds favours the formation of char ([25]Shafizadeh 1981).
The ash is typically composed of the following compounds:
CaO (±50%), K_2O (±20%), Na_2O, MgO, SiO, Fe_2O_3, P_2O_5, and SO_3 ([21]Baldwin 1987). The SO_3, as reported in an ash analysis, is in the form of a sulphate (for example, $CaSO_4$ or K_2SO_4).

Organic Materials
The organic compounds in wood are principally in the form of cellulose, hemicellulose, lignin. A small proportion are solvent soluble extractives (lipids and terpenes).

a) Cellulose
Cellulose is the same in all types of biofuel - except for the degree of polymerisation. It is composed of D-glucopyranose units linked linearly with b -(1 4) glycosidic links. Cellulose is the main component of wood ([26]Shafizadeh 1982).

b) Hemicellulose
Acetyl-4-O-methylglucuronoxylan forms the main hemicellulose of hardwoods. Glucomannan forms the main hemicellulose of softwoods. These carbohydrates easily break down at the glycosidic link. They have a lower heat of combustion than cellulose because they contain several molecules of water in their chemical composition ($[C_6(H_2O)_5]_n$ or $[C_5(H_2O)_4]_n$). Hardwoods have a greater proportion of acetyl and methoxyl groups than do softwoods. This explains why they are used in distillation to obtain acetic acid and methanol.

c) Lignin
Lignins have an approximate analysis of $C_{10}H_{11}O_2$ for both softwoods and hardwoods (Shafizadeh 1981:112). In addition to the same compounds as hemicellulose, hardwoods have syringyl propane units, and softwoods contain guaiacyl propane units. On combustion, lignin yields mainly char ([25]Shafizadeh 1981) although the quantity of char depends strongly on the burning conditions.

1.3 Biomass Combustion

Biomass fuels never burn directly: Solid biomass fuels are thermally degradable and under the influence of a sufficiently strong energy source they break down into a mixture of volatiles and carbonaceous char. The two modes of combustion (solid char, and gaseous volatiles) have completely different chemical mechanisms and kinetics. The thermal degradation of the fuel is called pyrolysis.

Pyrolysis
Cellulose and hemicellulose form mainly volatile products on heating due to the thermal cleavage of the sugar units. The lignin forms mainly char since it is not readily cleaved to lower molecular weight fragments ([25]Shafizadeh 1981).
The discussions of pyrolysis below focus primarily on cellulose because it is the largest component of the wood forming volatiles ([26]Shafizadeh 1982).

Ratio of volatiles to char
The proportion of volatiles to char is a complex interaction of temperature, heating rate, particle size and the effect of catalysts. Yield is highly dependent on the pyrolysis temperature (the heat treatment temperature), the nature of the substrate, and the presence of incombustible materials ([26]Shafizadeh 1982).
Addition of an acidic catalyst or slow heating promotes the dehydration and charring reactions. It follows that a higher temperature and smaller particle size, leading to increased heating rates, promotes the production of volatiles, whereas lower temperatures and larger pieces of wood promotes the formation of char (as well as water and CO_2). The presence of water, and inorganic materials also promote the formation of char because of the ‘heat sink’ they provide.

Initiation reactions
Cellulose and hemicellulose initially break into compounds of lower molecular weight. This forms an ‘activated cellulose’ which decomposes by two competitive reactions – one forming volatiles (anhydrosugars) and the other char and gases.

Pyrolysis reactions
The thermal degradation of the activated cellulose and hemicellulose to form volatiles and char can be divided into categories depending on the reaction temperature. Within a fire all these reactions take place concurrently and consecutively. The reactions are highly complex - they are briefly described in Table 5.

Table 5 Pyrolysis reactions at different temperatures

Condition	Processes	Products
Below 300°C	free radical initiation, elimination of water and depolymerisation	formation of carbonyl and carboxyl, evolution of CO and CO_2 and mainly a charred residue
300-450°C	breaking of glycosidic linkages of polysaccharide by substitution	mixture of levoglucosan, anhydrides and oligosaccharides largely in the form of a tar fraction
Above 450°C	dehydration, rearrangement and fission of sugar units	variety of carbonyl compounds such as acetaldehyde, glyoxal and acrolein which evaporate easily
Above 500°C	a mixture of all above processes	a mixture of all above products
Condensation	unsaturated products condense and cleave to the char	a highly reactive char residue containing trapped free radicals

Volatiles
The yield of gases within two different temperature regimes is given in Table 6

Table 6 Yield of volatile gases ([27]Speight 1993)

	<400°C	**>1000°C**
Yield of gas (m^3/ton)	**125**	**>550**
Composition (vol. %)		
CO_2	**30**	**20**
CO	**25**	**25**
CH_4	**14**	**12**
C_nH_m	**4**	**2**
H_2	**20**	**35**
N_2	**7**	**6**

The organic compounds in the biofuel break down and evaporate into ([25]Shafizadeh 1981).

1. a gas fraction containing:
    1. CO
    2. CO_2
    3. some hydrocarbons
    4. H_2
2. a condensable fraction containing:
    1. H_2O
    2. low molecular weight organic compounds
        1. - aldehydes
        2. - acids
        3. - ketones
        4. - alcohols
3. a tar fraction containing:
    1. higher molecular weight sugar residues
    2. furan derivatives
    3. phenolic compounds
    4. airborne particles of tar and charred material which form the smoke.

Gaseous emissions are predominantly a product of pyrolitic cracking of the fuel. If flames are present, fire temperatures are high, and more oxygen is available from thermally induced convection. The lower temperatures of the smouldering stage results in a lower oxygen supply from diffusion into the fuel bed – gases in this phase which leave the fuel bed are not oxidised further ([24]Lobert et al 1995).

Char

The char which is formed is highly reactive because of the trapped free radicals, and porous. This means a large surface area which has a large absorptive capacity. The properties of the char are related to the pyrolysis conditions as well as the physical and chemical properties of the fuel. Cellulose chars are most reactive and have the greatest surface area when formed at 550°C ([25]Shafizadeh 1981).

Combustion

The pyrolysed products, volatiles and char, burn with completely different characteristics. The burning of the active carbon (the char) to form CO_2 in the presence of sufficient oxygen and high enough temperatures is known as glowing combustion. Where temperatures are too low, or where there is insufficient oxygen for complete combustion smouldering occurs (characterised by smoking or emission of unoxidised pyrolysis products). The burning of the volatiles is known as flaming combustion. Flaming dominates at higher temperatures and smouldering at lower temperatures.

Reaction Rate

At lower temperatures, that is for glowing and smouldering combustion, the rate of combustion is controlled by the pyrolysis and combustion kinetics. At higher temperatures, for flaming combustion, the reactions take place very rapidly and heat and mass transfer factors dominate ([25]Shafizadeh 1981).

Determining the reaction rate of even a simple solid (such as a carbon sphere) is complex and involves at least the following five steps ([23]Kanury 1975):

1. Oxygen has to diffuse to the fuel surface,
2. Diffused oxygen has to be absorbed by the surface,
3. Absorbed oxygen has to react with the solid to form absorbed products,
4. Absorbed products have to be desorbed from the surface, and
5. Desorbed products have to diffuse away from the surface.

These steps, which must occur in order, determine the reaction rate. The slowest step, under the reaction conditions determines the burning rate. For carbon combustion steps 2 and 4 are known to be extremely rapid. Step 3 is the slowest when the particle temperature is low, and hence kinetic processes dominate. In this kinetic regime, the burning rate is exponentially related to temperature.

When temperature is high step 3 is faster than steps 1 and 5 and therefore the reaction rate is controlled by mass transfer processes. In this regime, burning rate depends weakly on temperature and strongly on particle size.

The formation of the volatiles at temperatures greater than 400°C is a highly endothermic process and consequently higher temperatures do not necessarily lead to greater reaction temperatures ([26]Shafizadeh 1982:757).

The formation of an ash layer on the char slows combustion appreciably (Baldwin 1987:183).

Combustion of Activated Carbon

Char is very different from pure carbon compounds like graphite. Cellulose chars have a formula of approximately $C_6.7H_3.3O$ ([26]Shafizadeh 1982). It is useful however to consider the burning of a pure carbon particle because it can give us an indication of what can be expected from complex compounds.

The burning of a carbon particle has been studied in some detail. Carbon can oxidise to form either carbon monoxide or carbon dioxide according to the following two equations:

$C + O_2 \quad CO_2$

$C + ½O_2 \quad CO$

At lower temperatures, and in the presence of sufficient oxygen ([25]Shafizadeh 1981) the formation of CO_2 dominates. At higher temperatures CO is formed preferentially, and either escapes or burns later, well away from the solid carbon.

The ratio of CO to CO_2 is influenced by various anions and cations ([25]Shafizadeh 1981). Phosphates and borates (anions) increase the formation of CO and decrease CO_2. Sodium and potassium ions (cations) reduce CO and increase CO_2 and therefore promote smouldering combustion.

At very high temperatures no oxygen reaches the carbon and it therefore 'burns' in carbon dioxide according to the following reaction equation:

$C + CO_2 \quad 2CO$

The flame produced by this reaction is pale blue and envelopes the char particle. Oxygen diffuses inwards, and CO or CO_2 move in the opposite direction.
Outgoing pyrolysis products also react with the carbonaceous layer ([23]Kanury 1975).
Frequently inert gases are absorbed on the char surface (gases such as N_2 and CO_2). These reactions are rapid and reversible.

Combustion of Volatiles
The volatiles burn with flames. Many free radical reactions are involved in all phases of this process. The transformations are extremely complicated and include further fission, dehydration, and disproportionation of the sugar units ([25]Shafizadeh 1981). Further pyrolysis of the volatiles can cause char particles to form in the gas. This is the cause of soot. Soot is also formed at temperatures of around 700°C according to the following reaction: $2CO \quad CO_2 + C$.

Sulphur Dioxide
Whereas CO is a product of incomplete combustion, SO_2 is the result of perfect combustion. Sulphates are frequently emitted as particulates ([28]Myers et al 1973). Sulphur can exist in the fuel in an organic or inorganic form. In coal there is usually more organic sulphur than inorganic ([27]Speight 1993). Organic sulphur groups are usually oxidised to sulphates ([29]van Krevelen 1961) whereas inorganic sulphur is oxidised to pyritic sulphur FeS_2 ([27]Speight 1993).

1.4 Biomass Conversion

Talbott's Heating Ltd manufacture a range of biomass-fired combustors from 25kW to 12,000kW, which have been designed to form the key component in renewable energy schemes, sustainable waste management programmes and distributed energy supplies. The system employs direct combustion of biomass/wood based material to produce hot water, hot air or steam. An additional benefit is a zero-cost waste disposal. Biomass combustion equipment represents a carbon dioxide neutral, renewable energy source using equipment, which can be sited within the industrial environment, unrestricted by geographical limitations. Talbott's have been manufacturing wood combustion units for twenty years and have 3000 referenced installations world-wide.

The fuel handling system consists of, an air-locked screw auger passing fuel slowly beneath the ceramic combustion zone extension so that even fuels with moisture content as high as 50% arrive at the grate area in a dry, pre-heated condition. Combustion primary air circulates around the outside of the ceramic firewalls lowering the temperature differential across the unit's outer casing and minimising losses. This also preheats the combustion air, which is then admitted to the fire zone at three injection levels. Primary air is fed to the base of the grate maintaining an even temperature distribution throughout the length of the firebed and ensuring a steady evolution of combustion gases. Secondary air is injected through ceramic nozzles above the firebed creating a turbulent atmosphere where combustion of the gases generated from the fuel primarily takes place. Finally, a tertiary air supply enters at a higher level still to provide an excess of oxygen for the remainder of the combustion zone. Air is admitted into this region in sufficient quantity to limit the temperature to around 1000°C. After leaving the firebed, the combustion gases, thoroughly mixed with the tertiary air, pass into the combustion zone extension. This ceramic labyrinth maintains the air and gases in an oxidising atmosphere at high temperature for up to two seconds to complete the 'total combustion' process.

1.5 Emissions

Talbott's wood combustors
All Talbott's combustors are compliant with EPA regulations. Biomass Combustor units will combust waste and produce less carbon monoxide and particulate matter than your average gas boiler. This helps our environment by using a non fossil fuel that is CO_2 neutral and produces no harmful sulphur emissions

The CO, HC and particulate, components of the exhaust gas stream are minimised by the total combustion effect of the extended ceramic zone. This provides up to two seconds dwell in a high temperature oxidising atmosphere before cooling in the heat exchanger. NOx : The combustion temperature is limited to 1000°C by careful control of gas evolution from the fuel (primary air) and the introduction of optimised volume of secondary and tertiary air to dilute the heat of combustion. The conditions thus created are far from ideal for the oxidation of nitrogen, which occurs, so readily in the high temperature/pressure rapid burning atmosphere of the internal combustion engine. Direct combustion of solid wood fuel also results in the combination of molecular hydrogen and oxygen in the wood’s chemical structure to yield water at the heart of the fire (even with zero moisture content fuel). As this water is vaporised heat is withdrawn locally from the firebed which inherently results in a lower peak combustion temperature than that achieved by fossil solid fuels, again helping to give low levels of NOx formation

Measured emissions
- Particulate Emission concentration 50mg/m^3 avg
- Formaldehyde and phenol blanks showed zero
- VOC 1.3 mg/m^3 carbon

Gas Emissions

Natural gas creates significantly smaller environmental impacts than coal. On a kJ basis, natural gas combustion generates about half as much carbon dioxide, or CO_2, as coal, less particulate matter, and very little sulphur dioxide or toxic air emissions. Natural gas combustion may, however, produce nitrogen oxides and carbon monoxide in quantities comparable to coal burning

Table 8 Composition of Natural Gas, percent by volume:

Methane (CH_4):	98.90%
Ethane (C_2H_6):	00.16%
Propane (C_3H_8):	0.02%
Nitrogen (N_2):	0.87%
Carbon dioxide (CO_2):	00.02%
Oxygen (O_2):	00.02%

Combustion Reaction Details:
Combining *Methane*, *Ethane*, *Propane*, and *Oxygen* produces: *Carbon Monoxide*, *Carbon Dioxide*, and *Water* (*Oxygen Dihydrate*).
$CH_4 + C_2H_6 + C_3H_8 + O_2 \rightarrow CO + CO_2 + H_2O$

This is the primary reaction. It is an exothermic combustion reaction (it produces heat), Carbon Monoxide is only formed in trace quantities (less than 1 ppm). The Water vapour, which sounds fairly harmless, actually often carries particles of soot from burnt dust, or foreign materials in the Natural Gas into the air. As such it can be a problem, it can be dealt with through the usage of an ionizer, which would attract all the droplets to its surface, getting them out of the air, but would also tend to increase the amount of Carbon Monoxide produced.

Combining *Nitrogen* and *Oxygen* produces *Nitrogen Dioxide*, *Nitrogen Trioxide* and related compounds. $N_2 + O_2 \rightarrow NO_2 + NO_3$ etc.

This is a secondary reaction, it is an endothermic combination reaction (it requires an outside energy source to take place, it is not self sustaining). The products are all usually referred to as Nitrogen Oxide (NO_X). Nitrogen Dioxide is particularly dangerous since when combined with water it produces Nitric Acid ($NO_2 + H_2O \rightarrow H_2NO_3$)Since human lungs contain a considerable amount of moisture, inhaling Nitrogen Oxide is considered to be extremely dangerous.
This reaction is actually independent of the Nitrogen in the natural gas itself, since there is an ample supply of Nitrogen in ordinary air. The reaction will take place wherever conditions permit (it requires an appropriate energy source to sustain the reaction, heat or ultra-violet light are both effective).

Most of the research found in the area of gas combustion, is based on NOx reduction.

Measured emissions

- O_2 = 18.2 %
- CO = 50 ppm @ 15% O_2
- NO_x = 48 ppm @ 15% O_2

1.6 Heat Exchangers

The combustion of biomass generates heat, for a system to be efficient it is essential to use as much of this heat as possible. Biomass heat exchangers need to be efficient whilst resistant to corrosion and fouling. Heat exchangers may be classed in three categories:

- Recuperator
  In this continuous process, heat is transferred from one fluid to another via a heat conductive separating wall.

- Regenerators
  In this cyclic process, heat is transferred by passing the fluid into a matrix container. By alternating the fluids the matrix becomes a heat source then a heat sink.

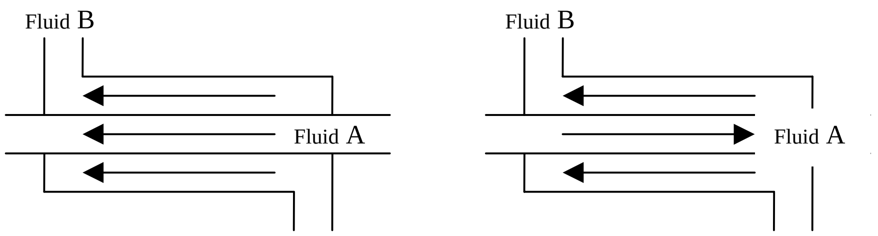
- Evaporators
  Liquid is cooled continuously by evaporation, in the same space as the coolant.

As our process is a continuous one, the recuperator is the best definition for our application.

<u>Parallel and counter flow recuperators</u>

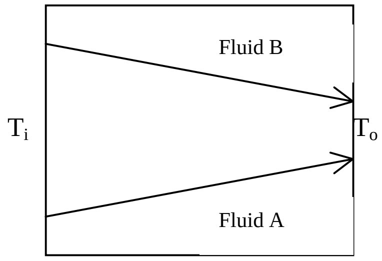
In a simple double pipe heat exchanger, heat is transferred from a fluid flowing inside a pipe to second fluid flowing around the outside of the same pipe. The second flow stream is directed by a second larger pipe, thus creating an annulus which controls the streams movement.

If the fluids flow in the same direction, it is said to be parallel flow, but if the flow are opposite, then it is counter flow.



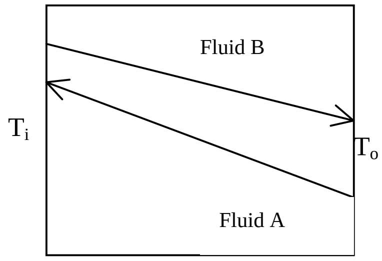
Parallel Flow Counter Flow

Temperature difference profiles for parallel flow and counter flow heat exchangers are detailed below.



Length

Parallel Flow



Length

Counter Flow

From the above it can be seen that the temperature difference is not constant over the length of the heat exchanger, therefore this temperature distribution must be taken into account in any evaluation. The Logarithmic Mean Temperature Difference method is used to achieve this.

The Logarithmic Mean Temperature Difference can be calculated by considering the heat transfer dQ across any short section of the heat exchanger tube: -

$$dQ = U\,dA\ \ T \qquad (1)$$

where U is the overall heat transfer co-efficient

A is the Surface area

T is the temperature difference

and $T = T_A - T_B$

$$d(\ T) = dT_A - dT_B \qquad (2)$$

The heat lost from fluid A must equal that gained by fluid B.

i.e., $dQ = -m_A C_{pA} dT_A = m_B C_{pB} dT_B$

where m is mass flow

C_p is specific heat at constant pressure

T is temperature

$$dT_A = \frac{-dQ}{m_A \times C_{p_A}}; \qquad dT_B = \frac{dQ}{m_B \times C_{p_B}}$$

Now $$d[\ T] = -\left[\frac{1}{m_A \times C_{p_A}} + \frac{1}{m_B \times C_{p_B}}\right] dQ \qquad (3)$$

Integrating (3) between inlet and outlet: -

$$T_o - \ T_i = -\left[\frac{1}{m_A \times C_{p_A}} + \frac{1}{m_B \times C_{p_B}}\right] Q \qquad (4)$$

Substituting for dQ in (3) from (1): -

$$\frac{d[\Delta T]}{\Delta T} = -\left[\frac{1}{m_A \times C_{p_A}} + \frac{1}{m_B \times C_{p_A}}\right] UdA$$

Integrating between inlet and outlet: -

$$\ell n\left[\frac{\Delta T_o}{\Delta T_i}\right] = -\left[\frac{1}{m_A \times C_{p_A}} + \frac{1}{m_B \times C_{p_B}}\right] UA \qquad (5)$$

From (4) and (5): -

$$Q = UA \left[\frac{\Delta T_o - \Delta T_i}{\ell n\left[\frac{\Delta T_o}{\Delta T_i}\right]}\right] \qquad (6)$$

From equation (6) it can be seen that the required mean temperature difference Tm is given by: -

$$\Delta Tm = \frac{\Delta T_o - \Delta T_i}{\ell n\left[\frac{\Delta T_o}{\Delta T_i}\right]}$$

In counter flow the temperature range is greater because the heated fluid exit temperature can be higher than the heating fluid exit temperature. For any given temperatures LMTD is always high in counter flow, meaning lower heat transfer surface area. When M_a x Cp_a = M_b x Cp_b then temperature difference is constant over its length. If one of the fluids is condensing or boiling then its temperature is constant.

If the heat exchanger is more than a single pass on either side, then LMTD correction factors maybe needed.

To determine rate of heat transfer(Q) we need to know the amount of surface area(A) and the overall heat transfer co-efficient(U). The surface area is simply the geometric area in contact fluids. The overall heat transfer co-efficient however is more complicated.

<u>Overall heat transfer co-efficient (U)</u>

Overall heat transfer co-efficient is the summation of all the heat transfer co-efficient's. These include outside and inside surface co-efficient's, wall conductivity and fouling factors(R). These factors can be thought of, as resistance's to heat. Therefore to add these factors we must invert them first.

$$U = \frac{1}{\frac{R_{fi}}{A_I} + \frac{1}{h_i A_i} + \frac{\ln(r_o/r_i)}{2\pi kL} + \frac{1}{h_o A_o} + \frac{R_{fo}}{A_o}}$$

Where r is tube radius

k is thermal conductivity

Overall heat transfer co-efficient may also be determined experimentally for a fixed geometry. This is often the only sure way to predict performance accurately with new heat exchanger types.

Now that we have our LMTD, surface area and overall heat transfer co-efficient, we can calculate the rate of heat transfer(Q) from the following expression:

$$Q = A.U.LMTD$$

1.7 Turbines

Converting biomass heat energy in to electrical energy is currently limited. The traditional method involves producing steam, by passing the hot biomass combustion gasses through a waste heat steam boiler. This steam is then used to drive a steam engine or turbine generator. Steam based systems provide low fuel to electrical output efficiency's, due to the dumping of heat in the condensation phase, as a result fuel feed rates high and the electric outputs are low. The economics of steam are hard to justify under 1MW due to low efficiencies and high capital costs. Steam is easier to justify when there is a use for the excess heat (CHP), but this relies on a constant demand for heat when the system is generating.
Steam engine tends to have higher efficiencies, but costs are high compared to turbines. Manufactures of steam engines have reduced dramatically, due to the low demand.
Alternative methods are currently being studied these include Gasification and direct firing of gas turbines.

Steam Turbines
Steam turbines are powered by either saturated or super heated steam. The volume of steam is expanded over the turbine blades and discharged to the steam or condensate recovery system. The discharge of the steam turbine can be low pressure saturated steam or condensate or a mixture of both. It is common practice to used the left over thermal energy for heating, this two fold approach is known as combined heat and power (CHP). Steam turbines like steam engines require a steam boiler and condensate handling equipment. All of these ancillary pieces of equipment dramatically add to the initial costs. This has had the effect of making units under 5MW commercially unattractive and difficult to economically justify.

Gas Turbines

Gas turbine in their simplest form consists of a compressor, combustion chamber and a power turbine. Air is drawn into the compressor and compressed, this compression forces the air into the combustion chamber. In the combustion chamber fuel is burnt to provide heat. This heat causes a rapid increase in temperature (between 800^{o}C and 1300^{o}C). The volume of air is increased substantially thus providing a high flow through the power turbine blades, which is converted in to shaft power to drive an alternator. The gases exhausting from the power turbine can be very high (500^{o}C) so methods have be developed to recover this heat to improve efficiency.
For power generation it was not economically viable to build turbines under 1MWe. Recent developments have introduced smaller, cost effective miniature gas turbines, known as "Micro-Turbines".

Micro-Turbines are gas turbines in the 30-1000kWe range. They are designed to be simple, low cost power generators for distributed power.

Photo 1.7.1
Bowman Power TG80

An Internet search yields manufactures of micro-turbines and research discussions. Manufactures include Capstone (30kWe) , Honeywell (Parallon 75kWe), Elliot and ABB.
Bowman produce Combined Heat and Power (CHP) turbogenerator sets with 80% system efficiency with 50 kW(e) electrical output power and 275 kW(th) thermal output power.
Ingersol Rand produce IPowerWorks, Small Gas Turbines, with low-NOx, recuperated gas turbine engines directly driving a low speed induction generator. Sizes range from 30 to 250 kW_e. Other manufactures are rapidly developing turbines, it to which is seen to be a big growth area.