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EXPERIMENTAL CHARACTERIZATION OF A BATCH FEED BIOMASS GASIFIER SYSTEM FOR INTERNAL COMBUSTION ENGINES

Z. A. Zainal, S. Bari and M. Z. Abdullah
School of Mechanical Engineering, Universiti Sains Malaysia
Engineering Campus, 14300 Nibong Tebal
Seberang Perai, Penang, Malaysia

Abstract Biomass is a very important source of energy in many parts of the world, especially for areas remote from supplies of high quality fossil fuels. Through the process of gasification, biomass can be converted into producer gas, which mainly comprises of H_2 and CO. The process biomass gasification treats the solid feed material in a gasifier such that virtually all of it is converted into gas, which has a calorific value of 3-5 MJ/Nm^3. One of the processes of gasification is the batch feed system, whereby material is added in batch every now or so. After cleaning, this gas can be used to run small internal combustion (IC) engines, boilers, process heaters etc. This system is cheaper compared to continuous flow system. However, there are many problems associated with such batch system of producing uniform gas-flow with constant calorific value. This study is involved in designing an air blown downdraft gasifier system and characterizing the process occurring within it (e.g. temperature) as well as the output (e.g. gas calorific value and flow rate). Application of the gas, produced from the gasifier, on a 20 kW reciprocating IC engine is studied with an electrical load of 12 kWe. Particular attention is given on cooling and cleaning system to provide quality of gas for such application.

Keywords: Biomass, Gasification, IC engine, power generation

INTRODUCTION

The use of producer gas, generated from biomass gasification, to run internal combustion engines dates to the Second World War. Many studies have been conducted on producer gas as a fuel for internal combustion engines for transportation as well as for electricity generation. These studies reveal that the producer gas can be used in four types of engines: i) Spark ignition petrol engine, ii) Compression ignition diesel engine operated on dual fuel with producer gas as the main and diesel as the pilot fuels, iii) Converted diesel engines into spark ignition gas engine, and iv) Gas turbines with either compressed producer gas and air mixture or with compressed air (Shashikantha, 1994). The first type has low overall efficiency and the second type needs both producer gas and diesel fuel. The third type seems to be the most suitable choice both in terms of efficiency and independence from diesel fuel. The fourth type, gas turbine requires the pressurization of the producer gas or a pressurized gasifier. Such techniques are capital intensive and are not directly applicable to smaller scale systems as discussed in this paper.

SPECIFICATION OF THE ENGINE

Email: mezainal@yahoo.com

Table 1 shows the specification of the engine used in this study. The original engine was a diesel engine, which was converted into a spark ignition engine to be run on natural gas. The engine had a compression ratio of about 13:1, which is higher than a similar capacity petrol engine (about 9:1). Because of higher anti-knock value of the producer gas, a much higher compression ratio of up to 15:1 could be used if available. The ignition system used was the transistorized coil ignition system or the high-energy electronic-ignition system, which consisted of a magnetic pulse generating system. The ignition timing was set at about 30^o BTDC (before top dead center). This was determined by the position of the magnet relative to the position of the piston at TDC.

Table- 1: Specification of the engine

Model	Lister HR2
No. of cylinder:	2
No. of stroke	4
Maximum speed	1800 rpm
Maximum power output	20 kW
Maximum capacity	2.2 liters
Cooling system	Air cooled
Compression ratio	13
Maximum motored pressure	14.50 bar

AIR FUEL MIXTURE

In most SI gas engines, air and fuel are mixed in a venturi or carburetor. However, in the present work a Tee junction was used to feed the air and the producer gas. Both the air and the producer gas mixtures were sucked into the system by the engine. Their flow rates were controlled by ball valves, and their values were read from variable-area flow meters (rotameters). The rotameter for measuring the amount of wood gas (producer gas) was calibrated based on the ratio of the densities of air and wood gas. The average mass air-fuel ratio of the engine was found to be 2.95. The theoretical mass air-fuel ratio for the combustion of producer gas (wood gas) was found to be 1.11; therefore the engine was running in a lean condition. The average flow rate of air was found to be 1441 liters/min whilst the flow rate of the fuel gas was found to be 556 liters/min. The average total flow rate of the air and fuel mixture was found to be 1997 liters/min. Although the rating of the engine is 20 kW at 1800 rpm, it had to match the generator that was rated at a maximum power output of 12.5 kW at 1500 rpm.

PROCEDURE FOR OPERATING THE NGINE

Fig 1 shows the piping system and the valves used for controlling the flow of wood gas, air and natural gas. Natural gas was used as the initial startup fuel for the engine. Valve 4 was initially closed. As the engine was being cranked, 40 liters/min of natural gas was passed into the Tee junction via valve 5 where it mixed with the incoming air. When the engine had fired, the flow rate of the natural gas was increased to 70 liters/min with the air supply maintained at about 700 liters/min via valve 3. The engine was allowed to warm up for about 15-20 minutes before running it on wood gas.

Prior to this or while the engine was running, the gasifier was started up with an air supply of 400 liters/min and biomass material of about 30 kg. The gasifier was initially filled with 1.5 kg of charcoal in the reduction zone to reduce the time for introduction of the wood gas into the engine. The wood gas was first vented by closing valve 2 and opening valve 1 fully, because it was observed that the wood gas initially contained dense smoke. This was due to the temperature of the combustion zone not raised above 900° C to crack down the tar formed as a result of pyrolysis process. After about 10 minutes, the wood gas was observed to be relatively clean and was ready to be introduced into the engine via the gas clean up system. Valves 4 was opened fully whilst valve 1 and 2 were gradually closed and opened respectively thus allowing the wood gas to enter the Tee junction and mixing it with air before being drawn into the engine. As this happened, the engine speed increased gradually. At the same time, valve 5, the natural gas inlet valve was turned down to maintain the speed of the engine. The change over from natural gas to wood gas was gradual to allow the engine to stabilize with the new fuel. After a few minutes, the engine was run entirely on wood gas without any support from the natural gas. An important procedure to be highlighted here is that the speed of the engine should be about 1500 rpm to sustain the momentum required for the changeover from natural gas to wood gas. A test performed at a lower speed of 1100 rpm failed during the changeover. When the engine was running on no load condition, some of the wood gas was vented off by opening valve 1 partially. The flow rate of the wood gas flowing into the Tee-junction was about 550 liters/min at a temperature of about 45° C whilst the air supply was maintained at about 1300 liters/min. At these flow rates the engine speed was maintained at 1500 rpm.

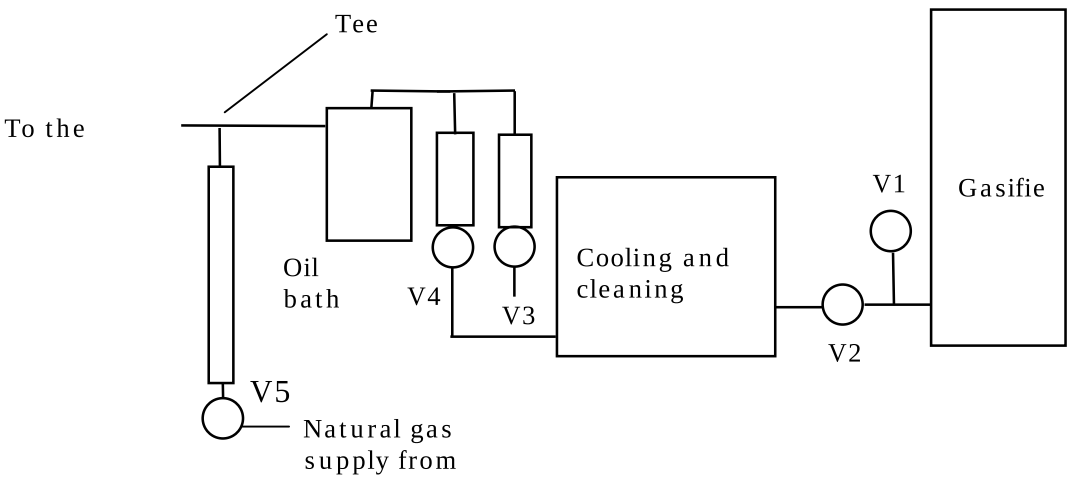

Fig. 1 Schematic diagram of the system showing the valves for controlling the air, the fuel gas and the natural gas into the engine.

When it was observed that due to insufficient supply of wood gas the speed of the engine decreased, valve 4 was slowly turned on to allow the natural gas to take over from the wood gas. Valve 1 was fully opened whilst valves 2 and 3 were fully closed. The engine was allowed to run on the natural gas for about 15 minutes before the whole system was shut down to minimize deposition of tar in the engine and the valves. The air supply into the gasifier was continued to ensure that all of the biomass material was thoroughly burned.

GAS CLEAN UP

Before introduction into the engine the wood gas was passed through the gas clean up system consisting of a cyclone, multitube cooler, box condenser, spiral condenser and an oil bath filter. The amount of water vapor condensed in the system was 8-10% of the feed. The amount of particulates trapped in the cyclone was about 0.03% of the feed. The amount of tar from the gasifier was found to be about 0.09% of the feed and after passing through the oil bath filter 77.0 % of the tar was trapped in the filter. The use of the oil bath filter proved to be very effective.

COMBUSTION CHARACTERISTICS OF WOOD GAS IN ENGINES

Wood gas, which mainly consists of hydrogen, carbon monoxide and methane, was ignited in the combustion chamber by spark plug. The ignition timing was set at about 30° BTDC to allow extra time for the slow burning of carbon monoxide and methane. The overall flame speed of wood gas is much lower than petrol, and together with high dilution with nitrogen reduces the chance of detonation of the end gas. Hence a diesel engine converted into spark ignition engine whilst maintaining the high compression ratio can achieve a higher thermal efficiency than a normal spark ignition engine. Throughout the experiment conducted detonation was not observed. However, Hollingdale (1983) reported otherwise.

TAR AND PARTICULATE IN THE ENGINE

The cumulative running time of the engine on wood gas was about 35 hours. One of the engine cylinder heads was removed to investigate the accumulation of tar and particulate. Observation of the piston crown and the cylinder head showed that the amount of tar was negligible because most of it burned in the combustion chamber. However, the tar and particulate were scraped from the surface of the piston crown and the cylinder head and weighed. The amount of tar and particulate were found to be 1.1 g. With the amount of wood gas entrained into both cylinders at about 550 liters/min, the total amount of particulate accumulated in the 35 hours of cumulative running was found to be 1.9 mg/m^3 or 63 mg/hr. It was also found that 0.7 g of tar and particulate were deposited behind one of the inlet valves. That would amount to a total of about 1.21 mg/m^3 or 40 mg/hr. The tar had caked up due to the high temperature inside the cylinder. The outlet valve was found to be free of any tar deposit because most of the tar is burnt inside the combustion chamber.

The engine was also subjected to regular treatment with Redex (a chemical treatment) after 6 hours of cumulative running to remove some of the tar and to avoid the valves from sticking. With the engine regularly treated with Redex and running on natural gas at the end of the test, overhauling period could be extend to about 200 hours where the amount of tar behind the inlet valve would be 4.0 g. Parikh (1991) reported that continuous running of 400 hours with dual-fuel (producer gas as the main fuel and diesel as the pilot fuel), the performance of the diesel engine was not affected.

EMISSION OF NOx FROM THE ENGINE

The emission of NOx was found to be in the range of 40-45 ppm. Most of the NOx is produced as a result of combustion of the wood gas and air in the engine.

Table-2: Overall performance of the biomass gasifier system.

Run	Power input into the gasifier (kW)	Power output from the gasifier (kW)	Power output from the generator (kW)	Efficiency of the engine / generator (%)	Overall efficiency of the biomass gasifier system (%)	Mass/kWh
1	75.79	38.41	8.06	21.0	10.63	2.00
2	71.44	46.73	5.81	12.4	8.13	2.58
3	57.35	39.12	6.10	15.6	10.64	1.97
4	59.35	40.2	6.25	15.6	10.54	2.02
5	39.13	29.33	6.05	20.6	15.46	1.49
6	48.93	41.39	6.04	14.6	12.34	1.74
7	58.53	46.54	6.02	12.9	10.28	2.06
Average	58.65	40.2	6.3	16.1	11.15	1.98

EFFICIENCIES OF THE ENGINE AND THE OVERALL SYSTEM

Table 2 shows the overall performance of the biomass gasifier system. The average power output from the gasifier was found to be 40.2 kW and the average power output from the generator was found to be 6.3 kW. Therefore, the average efficiency of the engine and the generator together was found to be 16.1%. Three convector heaters (maximum power output of 9.0 kW) were used to consume the power output from the generator. The overall efficiency of the system was found to be 11.15%. The average specific consumption of the biomass fuel is 1.98 kg/kWh.

During the experiment it was not possible to increase the power output to its maximum capacity of 9.0 kW due to the heaters tripping caused by overheating of the electrical heating element. Therefore, the convectors were set at 70.0% of full load to avoid tripping. The rated power output of 12.5 kW of the generator was therefore not fully utilized.

CONCLUSION

The paper highlighted the experiences and problems associated with running an internal combustion engine with producer gas. Using diesel converted spark ignition engine (DCSI) the average efficiency of the system was found to be 11.15% and the specific consumption was found to be 1.98 kg/kWh. The NO_X emission from the engine was found to be in the range of 40-45 ppm.

The use of oil bath filter was found to be essential, removing about 77.0% of the tar. The remaining tar was found to be deposited at the inlet valve, and would require regular maintenance.

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