

One Dimensional Modeling of a Turbogenerating Spark Ignition Engine Operating on Biogas

2011-01-1144

Published
04/12/2011

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doi:10.4271/2011-01-1144

ABSTRACT

Turbocompounding is generally regarded as the process of recovering a proportion of the exhaust gas energy from a reciprocating engine and applying it to the output power of the crankshaft. In conventional turbocompounding, the power turbine has been mechanically connected to the crankshaft but now a new method has emerged. Recent advances in high speed electrical machines have enabled the power turbine to be coupled to an electric generator. Decoupling the power turbine from the crankshaft and coupling it to a generator allows the power electronics to control the turbine speed independently in order to optimize the turbine efficiency for different engine operating conditions.

Some renewable electricity is presently being generated from compression ignition engines fuelled primarily on biogas using a small proportion of injected palm oil to initiate combustion. Spark ignition engines are being considered as an alternative lower cost option. This paper considers the challenges of implementing turbogenerating on a spark ignition engine operating on biogas. The 1D engine simulation work reported in the paper shows the required air/fuel ratio, compression ratio, ignition timing, valve timing and turbomachinery specifications in order for the engine to operate at maximum efficiency. The predicted performance of the turbogenerated spark ignition engine is compared with that of an equivalent turbocharged spark ignition engine also operating on biogas.

1. INTRODUCTION

Biogas is a renewable fuel that is produced by the anaerobic fermentation of organic material. A common way of utilizing this fuel is to use it in a spark ignition engine. These engines ingest a premixed gaseous air/fuel mixture which is ignited by a spark in a similar way to an ordinary gasoline spark ignition engine. However, when running a spark ignition engine on biogas, there are some drawbacks when compared to a similar engine running on gasoline. Reference [1] reports that the presence of carbon dioxide reduces the in-cylinder flame velocity which in turn reduces engine performance. Similarly, the presence of carbon dioxide lowers the average efficiency which in turn corresponds to an increase in the length of the combustion period and ignition delay. To counteract this loss in efficiency, the carbon dioxide gas content can be reduced by a methanation process [2]. This process can be costly and complicated and it also requires an energy input. One other possible way of counteracting this decrease in average efficiency is to install a Turbogenerator.

1.1. WHAT IS TURBOGENERATING?

Turbogenerating is a form of turbocompounding. [Figure 1](#) shows the typical layout of a turbogenerating engine system. The engine itself is turbocharged with the Turbogenerator turbine being located downstream of the turbocharger turbine. This downstream turbine is connected to a high speed electric generator which can supply electricity to auxiliary systems or compliment the main generator if the engine is being used in a 'genset' application.

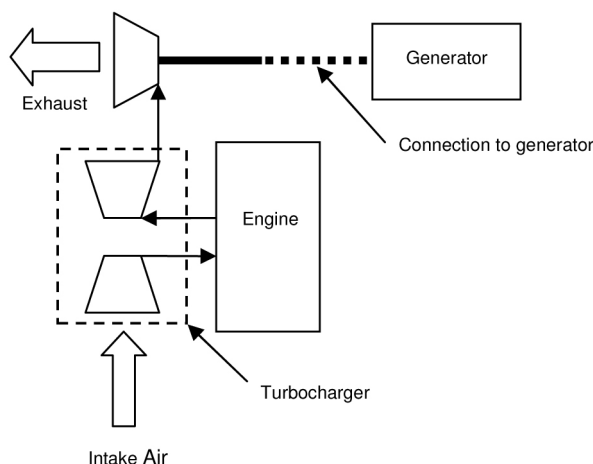


Figure 1. Typical layout of a Turbogenerating engine

Turbogenerating has been shown to reduce specific fuel consumption within internal combustion engines. Previous studies [3, 4, 5] have shown that the application of a Turbogenerator to an engine system has the potential to reduce the specific fuel consumption (S.F.C.) by up to 10% (depending on previous system setup).

Reference [4] pertains to an engine system which operates on biogas. This engine system has its combustion initiated by the injection of palm oil into the cylinder. Investigations into this engine system with the use of a validated model indicate a possible 9% reduction in S.F.C. when compared to a similar non-turbogenerated engine. It is thought however, that it may be possible to operate this engine as a spark ignition system, thus removing the need for the palm oil and all the extra associated costs.

Therefore, the purpose of this paper is to investigate the effects that a Turbogenerator has on a spark ignition engine operating on biogas. This turbogenerated engine's performance will then be compared against a similar turbocharged but non-turbogenerated engine operating on the same fuel.

2. MODELS USED

The models used for this investigation are based heavily on the model validated in reference [4]. However, its combustion sub model was altered to make it fully predictive. The engine being modeled is an 11 litre, straight 6 turbocharged, (turbogenerated) spark ignition engine, which is used to drive an electric generator at full engine load using biogas as its fuel source. The models used for the investigations are 1-dimensional time based solutions built using the commercially available software GT Power.

Within this paper, there are two engine models being looked at and compared. Model A is a spark ignition turbocharged engine operating on biogas and Model B is the same engine

with the addition of a Turbogenerator. These models are representative of typical turbocharged spark ignition engines and their important specifications are shown in Table 1. An important point to note is that when the Turbogenerator is installed; there are alterations that need to be made to the turbocharger turbine. The turbocharger turbine is reduced in size and an alternative housing is used, however, the compressor remains the same.

Table 1. Engine model A's specifications

Engine Speed	1500	rpm
Bore	127	mm
Stroke	154	mm
Connecting Rod Length	255	mm
Compression Ratio	8	
Swept Volume	11.7	litres
Cylinders	6	
Layout	Inline	
EVO	127	°A.T.D.C. _f
EPLP	241	°A.T.D.C. _f
EVC	363	°A.T.D.C. _f
IVO	357	°A.T.D.C. _f
IPLP	467	°A.T.D.C. _f
IVC	585	°A.T.D.C. _f

2.1. COMBUSTION SUB-MODELS

The aspect that makes these models different from other engine models is the fuel they are using. These engines run on biogas which is a product of fermentation of organic material and has its own unique combustion characteristic.

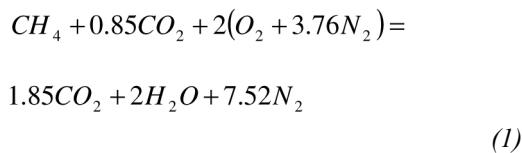
In order to model this biogas, a predictive combustion sub-model was used. The sub model is contained in the larger engine model and is a function within Gamma Technology's GT Power software. It is well-suited for homogeneous fuel-to-air mixtures and calculates the in cylinder burn rate assuming near 100% (0.9998) combustion whilst taking into account factors such as, cylinder geometry and fuel make-up. For this sub-model to accurately predict the energy released from the fuel, it needed to know the gas composition and the stoichiometric air to fuel ratio (A.F.R.). Whilst carrying out these calculations, it was found that the software being used to model the engines did not have the ability to allow the user to specify the laminar flame speed for a blend of hydrocarbons, hydrogen and nitrogen. It was at this point that the decision was made to remove the oxygen, nitrogen and hydrogen from the simulated biogas within the model. Table 2 shows the actual biogas composition and the simulated biogas within the model. It can be noted that the methane content of this biogas is on the low side of 'normal', however, these are the values used by the partner company and are considered correct.

Table 2. Biogas constituents

Constituent	Actual % by mass	Model % by mass
Methane	30.93319	30.00
Carbon Dioxide	68.38990	70.00
Oxygen	0.11276	0.00
Nitrogen	0.56380	0.00
Hydrogen	0.00035	0.00

The decision to remove the oxygen, nitrogen and hydrogen was deemed acceptable as they occur in very small quantities. Also, the models are being compared to each other as opposed to being used to accurately represent a specific engine system.

Equation 1 shows the modelled biogas's theoretical chemical reaction with air.



This produces a stoichiometric ratio of 4.98 kg of air to every 1kg of biogas. These new values for the biogas constituents were then input into the model and a check was done to verify that the model predicted the correct AFR.

2.2. MODEL A OPTIMISATION

As a final exercise before the investigations were performed, model A (turbocharged non-turbogenerating engine) was optimized to produce maximum efficiency at the rated conditions. This optimization process involved matching the turbomachinery to the engine, the compressor and turbine were sized accordingly to ensure that they operated in the most efficient regions of their respective maps. The valve timing was altered to minimize the amount of fuel entering the exhaust manifold during the valve overlap and the ignition timing was set with minimum advance for best efficiency. The compression ratio was set just below the point at which knock occurred. Table 5 gives key performance figures of model A at its rated conditions.

3. GENERALISED THEORETICAL INVESTIGATIONS

Before detailed investigations of the turbogenerating system were carried out, a more general investigation into boost pressures was performed. These general investigations were carried out on the above mentioned models with the turbomachinery being replaced by inlet and outlet conditions representing compressors and turbines with constant polytropic efficiencies of 77 and 80% respectively. The engine speed was fixed at 1500rpm with boost pressures being varied from 1.5 to 2.2 bar, these pressures are representative

of the range that the installed compressor can produce. The compression ratio was also varied as necessary to prevent knocking as the boost pressure was increased.

The reason for using fixed inlet and outlet conditions removes the need for matching the turbomachinery and provides a basis for comparison between turbocharged and turbogenerating engines.

Figure 2 shows generalised power and overall efficiency trends for the turbocharged and turbogenerated systems for varying boost pressures. It can be seen that power rises with increasing boost pressure for both systems but begins to drop off for the turbogenerated system after a peak of about 200kW. This drop in power is due mainly to the increase in back pressure caused by the addition of the turbogenerator.

The turbocharged and turbogenerated systems reach peak efficiencies of 35.9% and 40.5% respectively with the turbogenerated system's efficiency dropping off quicker than that of the turbocharged system. This quicker drop off is due to the reduction in compression ratio which is necessary in order to prevent knock and is not entirely offset by the addition of the Turbogenerator.

These generalized findings, albeit with lower efficiencies, are similar to the findings in [6] which pertain to compression ignition engines.

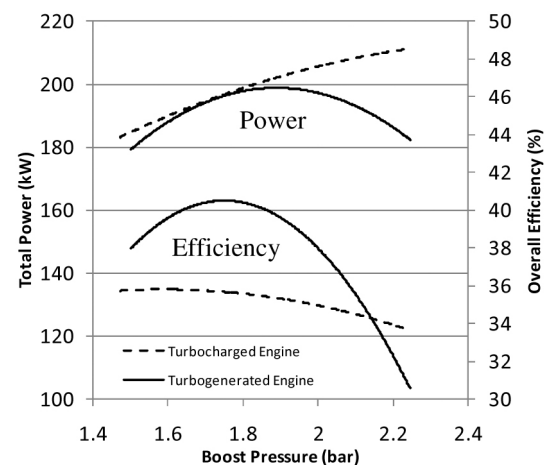


Figure 2. Generalised performance figures

4. DETAILED INVESTIGATIONS

The investigations in this paper pertain to the effects of placing a Turbogenerator in the exhaust stream of an optimized spark ignition engine operating on biogas (model A). After adding a Turbogenerator to the engine (now model B), the turbomachinery needed to be re-matched before any investigations could take place. Placing a Turbogenerator in the exhaust stream increases the exhaust manifold pressure.

Table 3. Key performance figures for the Turbocharged system

Boost Pressure (bar)		1.5	1.6	1.7	1.8	1.9	2.2
Engine Power	kW	182.4	187.1	194.3	197.8	201.8	211.8
TG Power	kW	-	-	-	-	-	-
Max. cylinder pressure	bar	51.2	52.3	52.6	51.7	48.4	46.4
System efficiency	%	35.6	35.8	36.0	35.8	35.0	33.8
Ignition delay	°CA	25.4	25.7	26.9	27.9	30.3	33.6
Engine backpressure	bar	1.06	1.09	1.14	1.18	1.24	1.40
BMEP*	bar	12.47	12.80	13.28	13.52	13.80	14.48

Table 4. Key performance figures for the Turbogenerated system

Boost Pressure (bar)		1.5	1.6	1.7	1.8	1.9	2.2
Engine Power	kW	155.1	156.5	162.3	166.0	160.9	136.7
TG Power	kW	26.4	29.1	31.3	33.4	38.3	45.2
Total System Power		181.5	185.6	193.6	199.4	199.2	181.9
Max. cylinder pressure	bar	47.5	46.0	48.6	49.7	43.8	28.7
System efficiency	%	38.7	38.9	39.9	40.5	39.3	30.3
Ignition delay	°CA	31.9	33.7	34.8	36.1	40.0	42.9
Engine backpressure	bar	1.98	2.14	2.29	2.44	2.71	3.53
BMEP*	bar	12.41	12.70	13.24	13.64	13.62	12.44

*BMEP is system BMEP calculated using 'Total Power' where:

$$BMEP = \frac{P_{total} n_c}{VN}$$

This in turn means that the turbocharger turbine needs to be of a smaller design. In order to compare the turbomachinery between the two models, a notional mass flow scale factor of 1 has been applied to the turbocharger turbine and compressor from model A. The same maps are then used in model B with the appropriate scale factors applied to these maps.

This matching process is not discussed here as it is detailed in many other pieces of literature [7]. For the purpose of these investigations; the turbomachinery was considered 'adequately matched' at this stage.

The model being investigated in this part is the turbogenerating model (model B). Effects into, boost pressure, turbogenerator speed and capacity, valve timing, and ignition timing were investigated with respect to total power output and overall engine efficiency. Once these effects had been established, an optimization exercise was performed to find the engine setup that produced the best overall system efficiency. These results were then compared against a similar turbocharged engine (Table 5).

4.1. BOOST PRESSURE

Boost pressure was varied from 1.5-2.2 bar while at the same time reducing the compression ratio to prevent knock. It is

known that retarding the ignition timing is a valid way of reducing knock, but for these investigations it was decided to hold this constant so that it was only the boost pressure and the geometric compression ratio that was affecting engine performance. Turbogenerator speed and capacity were kept constant at 49krpm and 1.275 respectively for these initial investigations.

Trends similar to those found in section 3 and displayed in Figure 2 were obtained when investigating boost pressure with the matched turbomachinery. Tables 3 and 4 show the effect that varying boost pressure had on key engine performance figures for the turbocharged and turbogenerated systems respectively.

When comparing tables 3 and 4 it can be seen that the maximum cylinder pressure for the turbogenerated engine is less than that for the turbocharged system. This is due to a higher quantity of trapped residuals in the cylinder causing a smaller amount of fuel to be burned. This lower cylinder pressure indicates that the ignition timing needs advancing as the flame front will be slower than in the turbocharged engine, this is looked at later in the paper. The higher quantity of trapped residuals is caused by the addition of the Turbogenerator which increases the engine's backpressure, this in turn leads to a reduction in power output from the engine crankshaft. However, in certain cases (boost pressures