Proceedings of ASME Turbo Expo 2011
GT2011
June 6-10, 2011, Vancouver, British Columbia, Canada

GT2011-45317

INVESTIGATIONS INTO THE PERFORMANCE OF A TURBOGENERATED BIOGAS ENGINE DURING SPEED TRANSIENTS

Ian Thompson
Queen's University Belfast
Belfast, N. Ireland

Stephen Spence
Queen's University Belfast
Belfast, N. Ireland

Charles McCartan
Queen's University Belfast
Belfast, N. Ireland

David Thornhill
Queen's University Belfast
Belfast, N. Ireland

Jonathan Talbot-Weiss
6 Tintara Street
Carseldine
Queensland
4034
Australia

ABSTRACT

Turbogenerating is a form of turbocompounding whereby a Turbogenerator is placed in the exhaust stream of an internal combustion engine. The Turbogenerator converts a portion of the expelled energy in the exhaust gas into electricity which can then be used to supplement the crankshaft power. Previous investigations have shown how the addition of a Turbogenerator can increase the system efficiency by up to 9%. However, these investigations pertain to the engine system operating at one fixed engine speed. The purpose of this paper is to investigate how the system and in particular the Turbogenerator operate during engine speed transients.

On turbocharged engines, turbocharger lag is an issue. With the addition of a Turbogenerator, these issues can be somewhat alleviated. This is done by altering the speed at which the Turbogenerator operates during the engine's speed transient. During the transients, the Turbogenerator can be thought to act in a similar manner to a variable geometry turbine where its speed can cause a change in the turbocharger turbine's pressure ratio.

This paper shows that by adding a Turbogenerator to a turbocharged engine the transient performance can be enhanced. This enhancement is shown by comparing the turbogenerated engine to a similar turbocharged engine. When comparing the two engines, it can be seen that the addition of a Turbogenerator can reduce the time taken to reach full power by up to 7% whilst at the same time, improve overall efficiency by 7.1% during the engine speed transient.

INTRODUCTION

A Turbogenerator is an exhaust driven electrical generator. To produce a turbogenerated engine system, the Turbogenerator is placed in the exhaust stream of an internal combustion engine as shown in Figure 1. The Turbogenerator is exposed to the full exhaust flow enabling it to recover as much energy as possible.

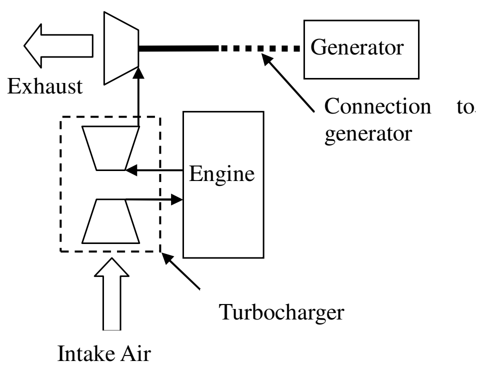

Figure 1. Typical layout of a Turbogenerated engine

The electrical energy produced by the Turbogenerator is used as and when required by the engine system or the vehicle it is in. The Turbogenerator's main benefit is that it can reduce fuel consumption by up to 10% [1-4]. References [1-4] pertain mainly to engines operating at fixed speeds and loads Some research has been done on the benefits of transient operation [5,6] but little is published with respect to turbogenerated biogas engines.

MODEL DEVELOPMENT

There are two models being used in these investigations, model C and model D. Model C is of a turbocharged engine operating on biogas and model D is the same engine with the addition of a Turbogenerator. These models are based heavily on the validated model from [2] except this time they are being used to investigate transient operation where the engine speed varies from 1200rpm to 2500rpm. Table 1 gives an overview of the engines being modelled.

Table 1. ENGINE SPECIFICATIONS

Engine Speed	1500	rpm
Bore	127	mm
Stroke	154	mm
Connecting Rod Length	255	mm
Swept Volume	11.7	litres
Cylinders	6	
Layout	Inline	
EVO	136	°A.T.D.C.$_f$
EPLP	250	°A.T.D.C.$_f$
EVC	363	°A.T.D.C.$_f$
IVO	354	°A.T.D.C.$_f$
IPLP	464	°A.T.D.C.$_f$
IVC	582	°A.T.D.C.$_f$

The model that models C and D are based on was built using the 1 dimensional engine modelling software GT Power [8]. The model has been set up in the conventional manner with the turbomachinery components having their own maps for reference. The electrical components of the model were given their respective efficiencies and a biogas mass flow valve that was controlled by a PID controller was installed. The model was then run and the results were compared to actual test data. Figures 2 and 3 show how well the model predicts performance when compared to actual test data.

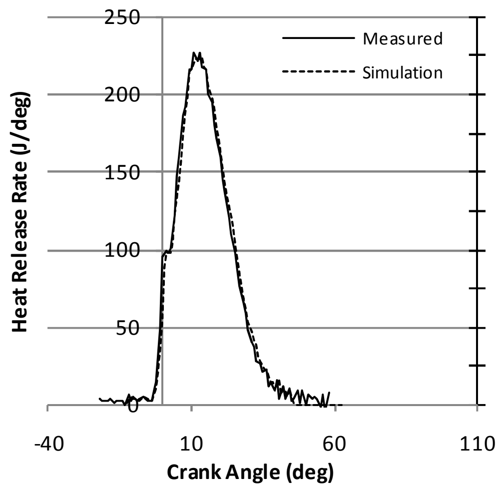

Figure 2. COMPARISION OF MEASURED AND SIMULATED HEAT RELEASE RATES

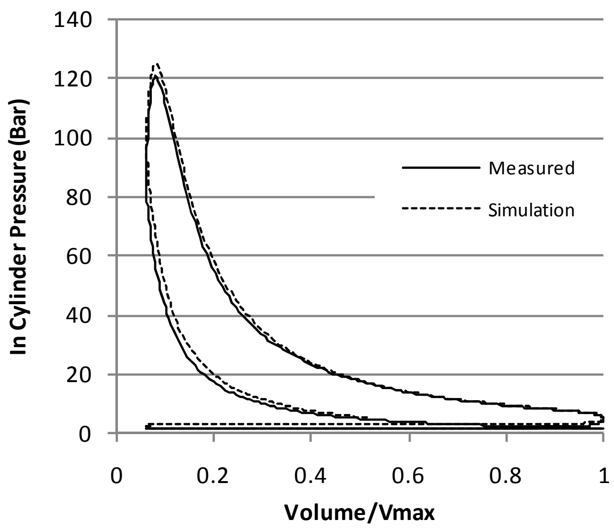

Figure 3. COMPARISION OF MEASURED AND SIMULATED P-V DIAGRAMS

A full validation exercise was carried out on the model and is discussed in more detail in [2].

The model in [2] has been validated at one operating point, 1500rpm, full load. In order to use this model for transient investigations correct fuel injection timing and biogas mass flow rate as the engine speed is varied had to be determined. From measured test data it was known that the correct injection timing was 22.5° B.T.D.C.$_f$ with an ignition delay of 9.5º CA. At 1500rpm, the 9.5º CA corresponded to a time value of 0.00106 seconds. For these investigations it was assumed that the ignition delay time remained constant regardless of the speed of the engine and that combustion is initiated 12º

B.T.D.C.$_{f}$. The reason for this assumption is the combustion sub model has a combustion profile from measured data which indicates such timings. With this in mind, new 'start of injection' times were calculated and input into the models. The values are shown in Table 2.

The models used in these investigations operate mainly on biogas and use a small amount of palm oil injected directly into the cylinder to initiate combustion. Another assumption that was made was that the air to fuel ratios were constant throughout the transient speed range. These AFRs were determined from the test data obtained from the engine tests and were input into the model.

Table 2. START OF INJECTING TIMINGS FOR TRANSIENT OPERATION

Engine Speed (rpm)	SOI °B.T.D.C.$_{f}$.	ID ° B.T.D.C.$_{f}$.	SOC ° B.T.D.C.$_{f}$.	Time for ID (s)
900	17.7	5.7	12	0.00106
1000	18.3	6.3	12	0.00106
1100	19.0	7.0	12	0.00106
1200	19.6	7.6	12	0.00106
1300	20.2	8.2	12	0.00106
1400	20.9	8.9	12	0.00106
1500	21.5	9.5	12	0.00106
1600	22.1	10.1	12	0.00106
1700	22.8	10.8	12	0.00106
1800	23.4	11.4	12	0.00106
1900	24.0	12.0	12	0.00106
2000	24.7	12.7	12	0.00106
2100	25.3	13.3	12	0.00106
2200	25.9	13.9	12	0.00106

One final model input that was determined from simulations was the profile of the transient. Simulations were run using AVL's Cruise software [9], a drive cycle simulation package, to obtain a realistic transient profile. The engine specifications were entered into AVL's Cruise along with vehicle specifications which represented a 10,450kg articulated truck. The engine speed profile which was obtained and input into GT Power is shown in Figure 4.

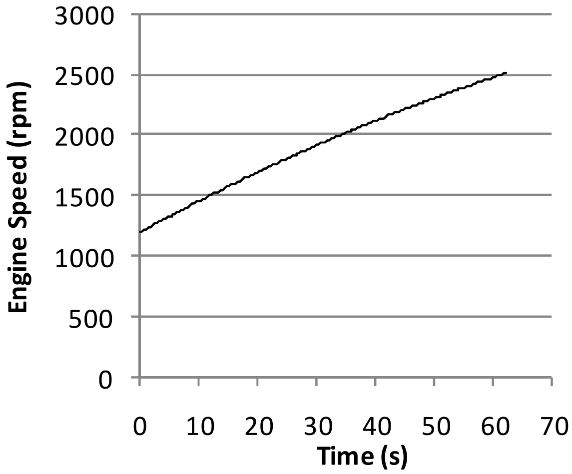

Figure 4. IMPOSED TRANSIENT PROFILE

To start the comparison between the two engines, the turbomachinery was matched to provide similar boost pressures throughout the speed transients of each engine. The size and geometry of the turbine wheels for these investigations has been fixed, the same is true for the nozzle guide vanes, it is assumed that the wheel sizes are of optimum specifications. Figure 5 shows how the boost pressure varies with engine speed for models C and D with Figures 6 and 7 showing how power and torque vary with engine speed.

It should be noted that the engines being modelled here had previously been used for electricity generation at one fixed speed as detailed in [2]. The powers displayed in Figure 6 are explained as follows:

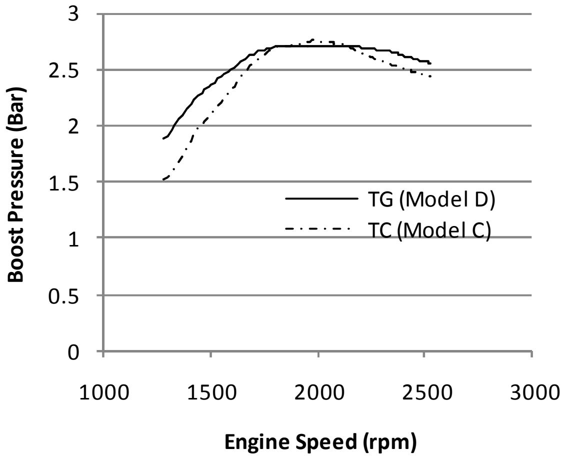

Figure 5. BOOST PRESSURE VARIATION

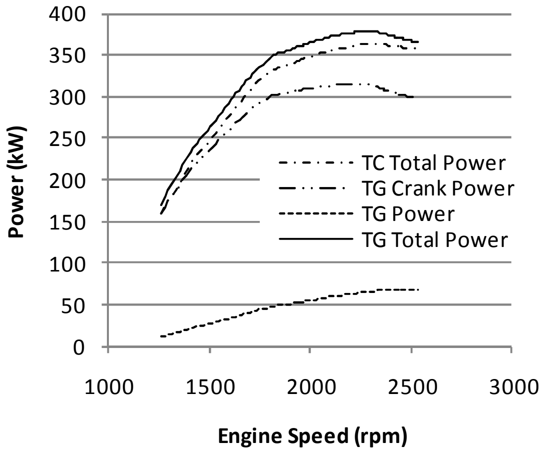

Figure 6. POWER VARIATION

TC Total Power – This is the total power output from the turbocharged engine, in this case, crank power equals total power.
TG Total Power – This is the sum of the crank power and the power produced by the Turbogenerator.
TG Crank Power – This is the power output from the turbogenerated engine's crank.
TG Power – This is the power produced by the Turbogenerator itself.

These investigations focus on comparing the total output from the engine systems, i.e. total powers and overall efficiencies. It is not the purpose of these investigations to determine how these power outputs may be used within a vehicle but merely demonstrate the power and efficiencies available.

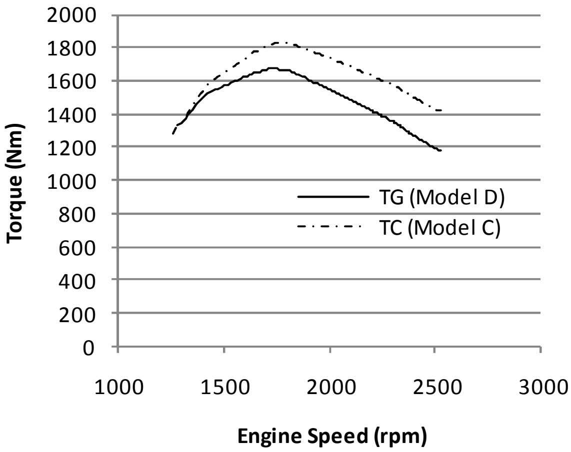

Figure 7. TORQUE VARIATION

IMPOSED TRANSIENT INVESTIGATIONS

One major advantage that turbogenerating has over mechanical turbocompounding is that the speed of the Turbogenerator's turbine wheel can be controlled independently of the engine's speed. This means that the pressure ratio across it and thus the pressure ratio across the turbocharger turbine can be controlled. This enables the user to control the boost pressure and the power output during a transient phase.

Looking at Figure 5, it can be seen that running an engine with a Turbogenerator at one fixed speed as opposed to a turbocharger alone during a transient produces a flatter, more constant boost pressure as well as increasing overall efficiency. However, one disadvantage that the addition of a Turbogenerator produces is that it causes the engine crankshaft to output a lower amount of torque; this is mainly due to the increase in engine backpressure caused by the addition of the Turbogenerator which reduces the engine's expansion ratio, thus resulting in less power being produced by the crankshaft.

Looking at Figure 8 it can be seen that the improvement in efficiency that a Turbogenerator operating at a constant speed produces is not constant throughout the engine speed transient. This will be addressed by varying the speed of the Turbogenerator with the results being shown later in the paper.

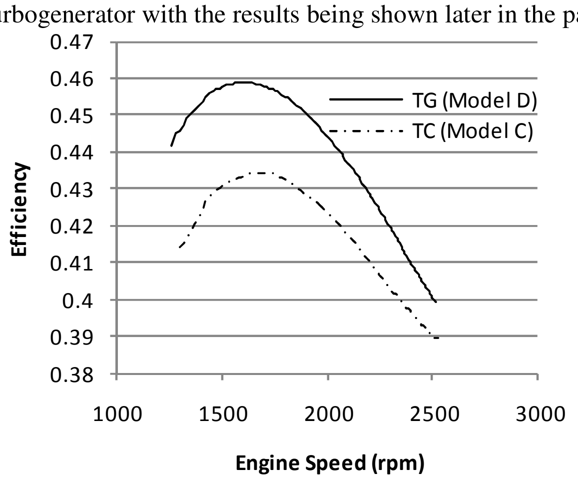

Figure 8. OVERALL EFFICIENCY

The first part of this paper deals with investigations under imposed realistic transient conditions. The model from [2] operates at a fixed engine speed of 1500rpm with a Turbogenerator speed of 49krpm. This is the speed at which the Turbogenerator was run at during the entire transient for the above investigations. However, it is known that altering the Turbogenerator speed has an effect on engine performance. Therefore, the next step was to investigate how engine performance is affected by Turbogenerator speed during an engine speed transient.

Turbogenerator Speed

The same engine speed transient profile from before was used and the Turbogenerator speed in the models was varied from 40krpm to 70krpm, this is representative of what can be done in reality by the Turbogenerator's electronics.

Figure 9 shows how overall system efficiency varies during the transient with differing Turbogenerator speeds.

From Figure 9 it can be seen that a benefit can be achieved from varying the Turbogenerator speed during the transient. It appears that the greater the engine speed and the greater the Turbogenerator speed, the greater the benefit. The data from these Turbogenerator speed investigations was analysed and the Turbogenerator speed that corresponded to the highest efficiency at each engine speed was determined. This Turbogenerator speed profile was drawn up as shown in Table 3 and input into the simulation software, GT Power.

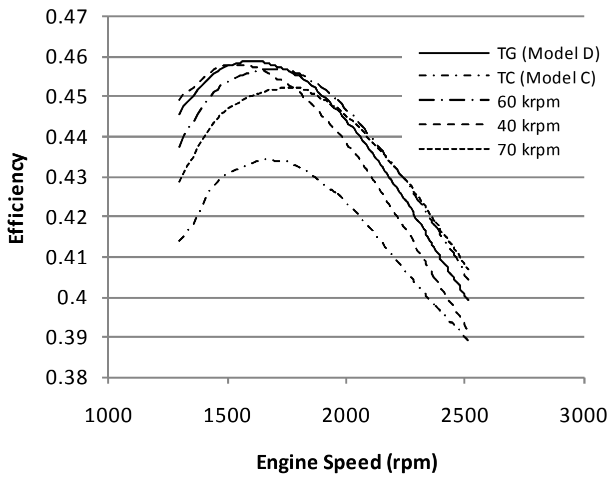

Figure 9. OVERALL SYSTEM EFFICIENCY FOR DIFFERENT TURBOGENERATOR SPEEDS DURING ENGINE SPEED TRANSIENT

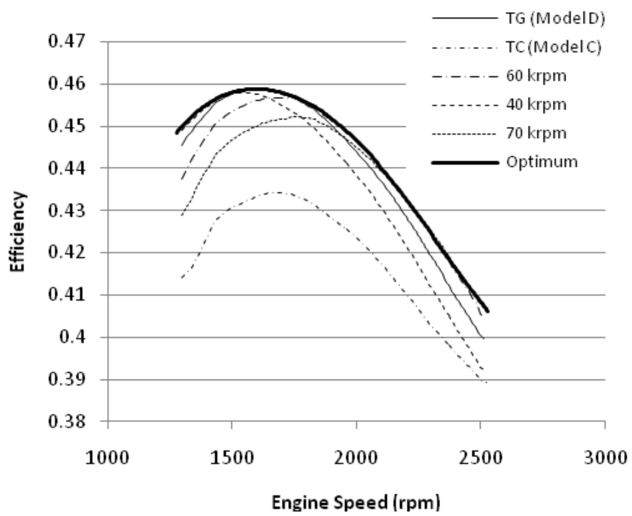

Figure 10. OPTIMUM SYSTEM EFFICIENCY

Figure 10 shows how overall system efficiency is improved when the Turbogenerator speed is altered during the engine transient. The optimum (thickest solid black line) bounds all the other lines showing that maximum efficiency is improved by varying the Turbogenerator speed during an engine speed transient. When comparing Figure 10 to Figure 8 it can be seen that varying the Turbogenerator speed, the efficiency improvement is more constant over the engine speed range.

Figure 11 compares this new overall system efficiency (varying TG speed) to the overall system efficiency when the Turbogenerator is held at a constant speed during the engine transient. The area under the thicker of the two solid lines (varying Turbogenerator speed) is larger than the area under the thinner solid line (constant Turbogenerator speed) indicating an increase in overall efficiency throughout the transient with the biggest increases occurring at low and high engine speeds.

Table 3. TURBOGENERATOR SPEED PROFILE FOR OPTIMUM SYSTEM EFFICIENCY.

Engine Speed (rpm)	TG Speed krpm
1200	37
1300	37.8
1400	40.9
1500	43.3
1600	48.3
1700	49.9
1800	60
1900	60
2000	60
2100	60
2200	60
2300	65
2400	70
2500	70

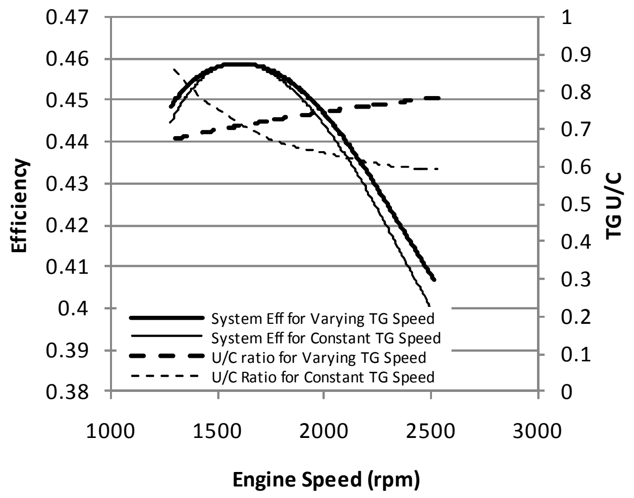

Figure 11. SYSTEM EFFICIENCY AND TG VELOCITY RATIO

The reason that varying the Turbogenerator speed during the engine speed transient improves efficiency is that the Turbogenerator turbine can be operated at its most efficient point. To determine where this was, efficiency plots as a function of velocity ratio were plotted; these are shown in Figure 12. From Figure 12 it can be seen that the maximum efficiency point moves from a velocity ratio of around 0.7 to 0.8 as the Turbogenerator's speed increases. This is confirmed by the thicker dashed line in Figure 11 which shows the Turbogenerator's velocity ratio during the engine speed transient increasing from roughly 0.7 to 0.8 as the Turbogenerator's speed also increases. Compare this to the thinner dashed line in Figure 11 which shows how the velocity ratio of the Turbogenerator changes during an engine speed transient as the Turbogenerator speed remains constant at 49 krpm.

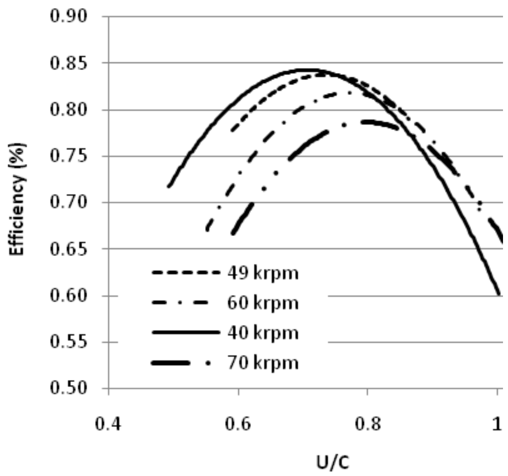

Figure 12. TURBOGENERATOR TURBINE EFFICIENCY AS FUNCTION OF VELOCITY RATIO

Valve timing

As has been noticed in previous experiments [7], the benefit of turbocompounding can be further enhanced by altering the valve timing. Intake and exhaust valve timing was investigated separately by varying the crank angle at which peak lift occurs (CTA) being varied.

Inlet Valve Inlet valve closure timing mainly affects volumetric efficiency. To investigate how this affected engine performance during an engine transient (TG speed was also varied) the intake valve CTA (duration was kept constant) was varied from 446 to 470°A.T.D.C.$_{f}$. Figure 13 shows how this parameter affects overall system efficiency.

Figure 13 shows that different inlet valve timings produced different efficiencies at different engine speeds. At low speeds later inlet valve opening (and thus closing) improves efficiency. This is in contrast to the higher engine speeds where an earlier inlet valve opening (and thus closing) improves efficiency.

Exhaust Valve The same investigations were performed on the exhaust valve with its CTA being varied from 235 to 262°A.T.D.C.$_{f}$. Figure 14 shows how the exhaust valve's CTA affects overall efficiency. When comparing the different efficiencies produced it can be seen that there is very little to be gained by varying the exhaust valve timing whenever the Turbogenerator speed is varied during the engine speed transient. One reason for this providing no benefit is that varying the Turbogenerator's speed causes the back pressure to change which in turn alters the rate at which the gasses leave

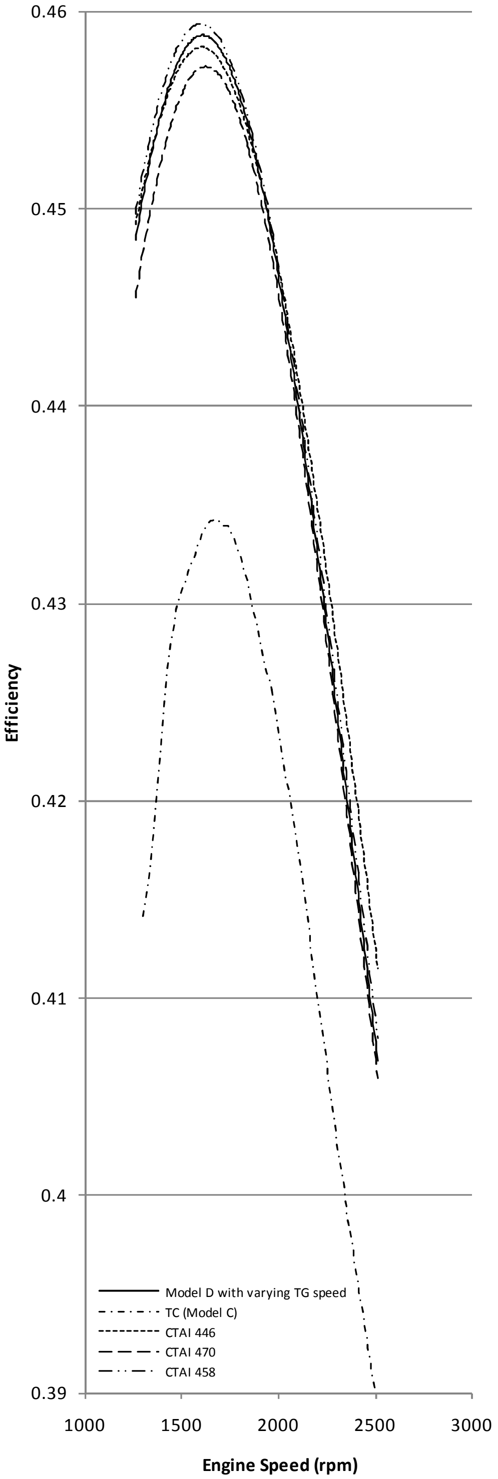

Figure 13. HOW OVERALL EFFICIENCY IS AFFECTED BY INLET VALVE TIMING

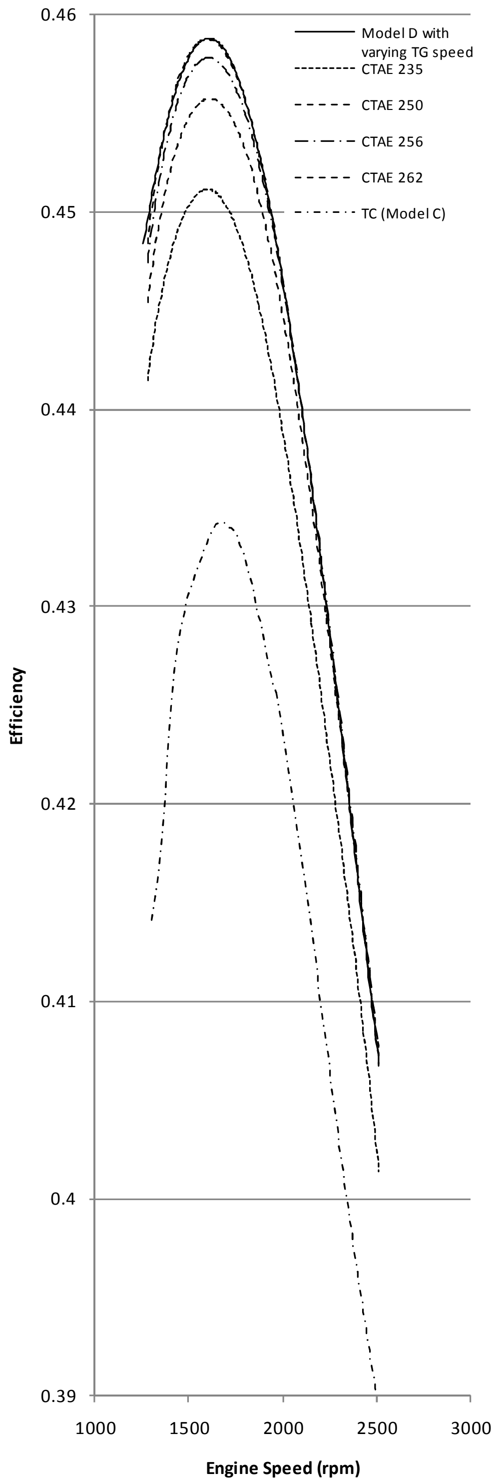

Figure 14. HOW OVERALL EFFICIENCY IS AFFECTED BY EXHAUST VALVE TIMING

(and to a certain extent, enter) the cylinder. In short, varying the Turbogenerator speed during an engine transient produces similar results as varying the exhaust valve timing.

On a turbogenerated engine system it may be beneficial to implement variable inlet valve timing but there is little to be gained from implementing variable exhaust valve timing. Even at that, the gains to be had are marginal. These gains would be greater should the Turbogenerator operate at one fixed speed during the engine speed transient but it may be easier to vary the Turbogenerator's speed than re-fit and engine with a variable valve system.

To understand just how much effect variable valve timing has on overall efficiency, an engine speed transient simulation was run where the TG speed was varied along with the inlet valve CTA being varied from 458°A.T.D.C.$_f$ at low engine speeds to 446°A.T.D.C.$_f$ at high engine speeds. These new timings were compared to the maximum efficiency produced from the previous set of investigations (Model D where TG speed varied only). Figure 15 shows that by varying the TG speed and the inlet valve timing, overall efficiency can be improved by around 0.5%

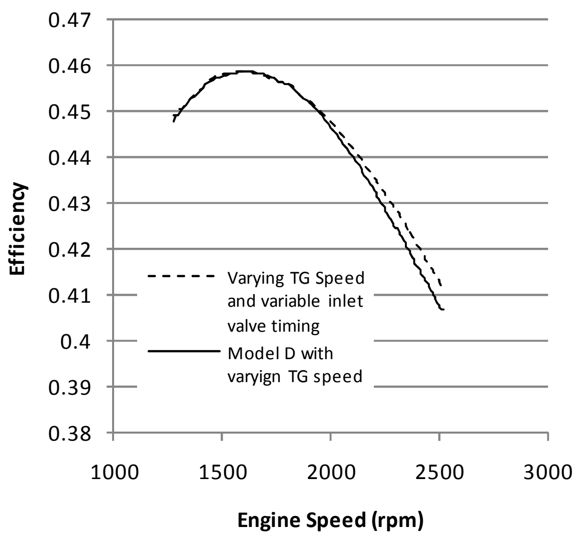

Figure 15. IMPROVEMENT GAINED BY VARYING TURBOGENERATOR SPEED AS WELL AS VARYING THE INLET VALVE TIMING

NON IMPOSED TRANSIENT INVESTIGATIONS

So far, the engine transient has been imposed, i.e. the engine speed has been dictated according to time. These next set of investigations have a load applied to the engine with a step change to this load occurring at some stage during the transient. This step change in load was applied to model C and the optimum model D (Turbogenerator speed varies with engine speed using TG speed profile from Table 3) in the same way.

Adding a Turbogenerator to an engine system not only increases overall efficiency, it also helps improve boost levels which in turn improves the engine's response to load input changes. Figure 16 shows how boost pressure varies in a turbocharged engine, a Turbogenerated engine with the Turbogenerator operating at a fixed speed (49 krpm) and an engine where the Turbogenerator's speed varies with engine speed.

Figures 16 and 17 show that by allowing the Turbogenerator speed to vary during the engine speed transient, not only can a more constant boost pressure be maintained but the time taken for the engine to reach full power can be decreased. This implies that a turbogenerated engine can react more quickly to changes in load than a turbocharged engine. To demonstrate this, model C and the optimum model D (Turbogenerator speed varies with engine speed using TG speed profile from Table 3) were run from 1300 rpm to 2500 rpm with step change in load occurring at the same point. Figure 17 shows how the two models differ in the time taken to reach maximum power. The time taken for the turbogenerated engine system to reach its maximum power is 7% less than the time taken for the turbocharged model. This is due to the fact that as the Turbogenerator's speed changes so too does the work required to overcome the centrifugal force imparted on the flow by its turbine's rotational speed, this in turn causes the pressure ratio across the Turbogenerator and thus the turbocharger turbine to change. This means that at low engine speeds it is beneficial to have the Turbogenerator rotating at a low speed to allow the turbocharger turbine to make as much use of the larger pressure ratio across it as at low Turbogenerator speeds; the TG's pressure ratio is small. A further reason for the turbogenerated engine reaching maximum power sooner is that the turbocharger turbine is smaller than the one on the turbocharged engine, this means the compressor on the turbogenerated engine can reach full boost more quickly.

In short, varying the Turbogenerator's speed allows the pressure ratio across the turbocharger turbine to be held at its optimum operating point.

One point that should be made is that it is the total (crank plus TG power) output power from the turbogenerated engine that is achieved quicker. The crank power in the turbogenerated engine is actually developed slower but the power recovered by the TG makes up for this loss.

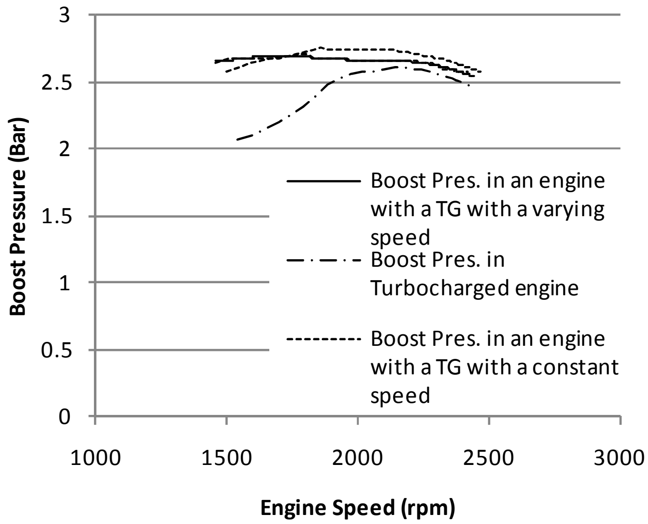

Figure 16. BOOST PRESSURE AS A FUNCTION OF ENGINE SPEED

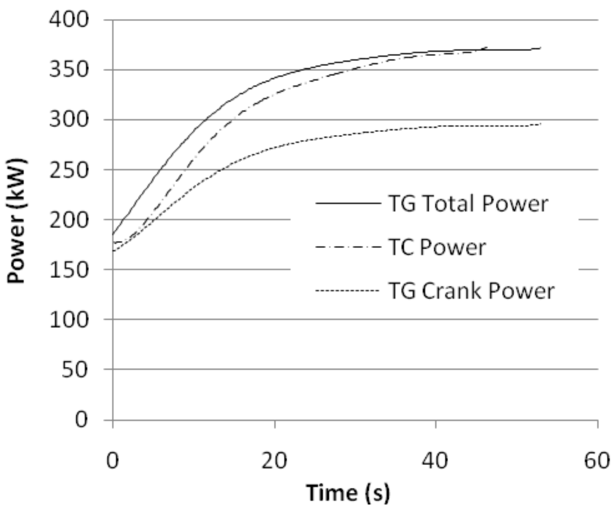

Figure 17. POWER OUTPUT AGAINST TIME DURING THE ENGINE SPEED TRANSIENT

CONCLUSIONS

These investigations have shown that by adding a Turbogenerator to a turbocharged engine system, overall system efficiency can be improved by 2.8% at 1300rpm and 1% at 2500rpm which equates to a 5.1% increase over the entire full load engine speed transient.

When the Turbogenerator speed is varied during the transient these figures can be further improved to an increase of 3.5% at 1300rpm and 1.8% at 2500rpm which equates to a 6.5% increase over the entire full load engine speed transient.

Using variable inlet valve timing in conjunction with variable Turbogenerator speed also yields and improvement in overall efficiency. This improvement equates to a 7.1% increase from the turbocharged engine over the entire full load engine speed transient.

There is little to be gained from implementing variable exhaust valve timing.

By adding a Turbogenerator to a turbocharged engine system the engine's response time to a step input change in load can be reduced by up to 7%.

FURTHER WORK

Figure 6 implies that the addition of a Turbogenerator has the potential to improve overall efficiency further at lower engine speeds. It would be interesting to see how a Turbogenerator affects system efficiency at lower speeds, e.g. engine speeds below 1300rpm.

Altering intake valve timing has been shown to improve overall efficiency, the next step would be to investigate valve duration of both inlet and exhaust valves as there may be further gains to be had.

Investigations into part load transients would be interesting as well as emissions analysis.

The engine being modelled here does not have an exhaust gas recirculation system. As emissions legislation becomes ever more stringent, future engines are likely to use some type of EGR aftertreatment. Investigations into the impact that this would have on engine operation would be interesting as the addition of a Turbogenerator increases the engine's backpressure which aids high pressure EGR.

ACKNOWLEDGMENTS

The authors would like to thank Bowman Power Group Ltd for their technical support and provision of test data with which to validate the models. Thanks is also given to Mr Jason Walkingshaw for his assistance in analysing a proportion of the turbine data.

NOMENCLATURE

AFR	Air to fuel ratio
B.T.D.C.	Before Top Dead Centre
f	Firing
CA	Crank Angle
CTA	Crank Timing Angle
SOI	Start of injection
ID	Ignition delay
SOC	Start of combustion
TG	Turbogenerator
U/C	Velocity ratio

REFERENCES

[1] **Talbot-Weiss, J.,** “SFC Improvements from Turbo-Generating Heavy-Duty Diesel Engines”, PhD Thesis, University of Sussex, 2009

[2] **Thompson, I.G.M., Spence, S.W., McCartan, C.D. and Talbot Weiss, J.,** “Design, validation and performance results of a turbocharged turbogenerating biogas engine model”, 9th International Turbocharger Conference, London, England, May 19th and 20th, 2010, Paper No 101219

[3] **Hopmann, U. and Algrain, C,** “Diesel Engine Electric Turbo Compound Technology”, Future Transportation Technology Conference, Costa Mesa, California, June 23-25, 2003 SAE Paper 2003-01-2294

[4] **Vuk, C.,** “John Deere - Electric Turbo Compounding technology update”, 13th Diesel Engine-Efficiency and Emissions Research Conference, Detroit, Michigan, 2007

[5] **Panting, J., Pullen, K. R. and Martinez-Botas, R. F.,** “Turbocharger Motor-Generator for Improvement of Transient Performance in an Internal Combustion Engine”, Proc Instn Mech Engrs Vol 215 Part D, Paper D04500, 2001

[6] **Wei, W., Zhuge, W., Zhang, Y. and He, Y.,** “Comparative study on electric turbo-compounding systems for gasoline engine exhaust energy recovery”, Proceedings of ASME Turbo Expo 2010, June 14-18, 2010, Glasgow, UK, Paper no. GT2010-23204

[7] **Wallace, F.J. and Cox, A.,** “The Ultimate performance potential of compounded diesel engines for heavy vehicles”, 6th International Conference on Turbocharging and air management systems”, IMechE HQ, London, UK, 3-5 November 1998, C554/015/98

[8] **GT Power v 7**, Gamma Technologies Inc, Westmont, IL, USA

[9] **AVL Cruise**, AVL United Kingdom Limited, Avon House, Hartlebury Trading Estate, Worcestershire, DY10 4JB, UK