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

GT2011-46599

A One-Dimensional Investigation of the Effect of an Active Control Turbocharger on Internal Combustion Engine Performance

Apostolos Pesiridis, Benjamin Dubois and Ricardo F. Martinez-Botas

Mechanical Engineering Department
Imperial College London
United Kingdom

ABSTRACT

The present paper discusses the impact of a new type of turbocharger, namely, the Active Control Turbocharger (ACT). The aim of this work was to prove the advantage of this type of turbocharger over the current state-of-the-art: the Variable Geometry Turbocharger (VGT). This was achieved by carrying out a comparison between two commercial Diesel engine models (through the use of a commercial engine simulation software), which belong to the same family: one 10 litre engine equipped with VGT (originally) was consecutively compared to the same model of engine modified for ACT operation and through the integration of the ACT into the 8l version of the same engine in order to demonstrate the ACT's downsizing capability.

The study has been carried out for speeds between 800 and 2000 rpm, and a fuel-air ratio range of between 0.017 and 0.057. The results showed that the actuation of the turbine in ACT mode (through the sinusoidal regulation of the turbine inlet area with each incoming exhaust gas pressure pulse) increases greatly the energy available at the turbine inlet. This leads to an increase of the boost pressure at the intake of the engine by an average 30%. The specific fuel consumption was found to be similar throughout engine operating range with a penalty of up to 10% for the ACT engine of the same size (10 litre). A comparison was then carried out between the 10 litre VGT engine and the 8 litre ACT engine. The 8 litre has been found to produce up to 37% more torque and horse power under 1400 rpm and obtained very similar performance to the 10 litre VGT engine at higher speeds. At constant power output between the 8 and 10 litre engines, it has been found that the fuel consumption was decreased by a maximum of 9% when using the 8 litre engine.

The results of the present study were encouraging with respect to the potential of ACT to downsize the internal combustion engine.

KEYWORDS: flow control, downsizing, turbocharger, variable geometry

INTRODUCTION

Downsizing is already an established fuel economy solution and has already been adopted by a number of vehicle manufacturers. The driving force behind the capability of engine designers to downsize their engines is the boosting system. This may involve various forms of turbocharging or supercharging. The current state-of-the-art in turbocharging is the Variable Geometry Turbocharger (VGT) which has already displaced the majority of earlier fixed (turbine inlet) geometry turbochargers in Diesel engine applications. VGTs are an important progression in the evolution towards ever cleaner internal combustion (IC) engines and adherence to ever stricter emissions regulations.

The Active Control Turbocharger (ACT) is a technology which was developed at Imperial College with the view to offer a new option in meeting with these ever increasing demands required of turbocharging. The Active Control Turbocharger (ACT) is a turbocharger system which consists of a system and method of operation, which regulate the inlet area to a turbocharger inlet, throughout each engine exhaust gas period, thereby actively adapting to the characteristics of the high frequency, highly dynamic flow. This so far untapped energy source lends itself to exploitation and the following definition of the Active type of Control for turbochargers and of the Active Control Turbocharger as a system can be given as follows:

Active Control of an exhaust gas flow into a turbocharger turbine is defined as a ***method of operation*** *of a turbine inlet mechanism in which a periodic turbine inlet area regulation* *can be provided by the mechanism at the exact rate, if so required, at which the exhaust gas is emitted - in the form of periodic flow - by an internal combustion engine and at a geometrical amplitude* ***proportional*** *to the amount of emitted exhaust energy, with the intent to raise the cycle-averaged energy content flowing through a turbocharger turbine thereby increasing the power output of the turbine.*

An Active Control Turbocharger is defined as a ***turbocharger system*** *that is equipped with a turbine inlet area-regulating mechanism* ***capable*** *of applying an active method of control of the exhaust flow entering the turbine.*

The aim of this work was to prove the downsizing capabilities of the Active Control Turbocharger (ACT) at the simulated level. In the following sections, details of the operation of the ACT are introduced, followed by a description of the construction of the model of the 6-cylinder CI engine and its validation. Model results were obtained for a 10l Variable Geometry Turbocharger (VGT) engine which was then used as the basis for comparison with the 10l ACT engine model and the 8l ACT engine model results. In this way a comparison of ACT against VGT for the same was possible as well as the possibility to replace a 10l engine with the 8l version in the same series of engines while assessing the level of performance to ensure it stayed within at least comparable limits with the larger engine.

NOMENCLATURE

General Symbol	Definition
a	Vane Amplitude (°)
A	Turbine Inlet Area (m^2)
A*	Critical Turbine Inlet area (m^2)
A/F	Air Fuel Ratio
ACT	Active Control Turbocharger
CI	Compression Ignition (Engine)
C_p	Specific Heat (J/kg.K)
f	Frequency (Hz)
FGT	Fixed Geometry Turbocharger
IC	Internal Combustion (Engine)
M	Mach Number
$\dot{m}$	Mass Flow Rate (kg/s)
n	Number of Strokes
N	Speed (rpm)
P	Pressure (Pa)
rpm	Revolution per Minute
sfc	Specific Fuel Consumption (kg/kW/hr)
T	Temperature (K)
t	Time (s)
VGT	Variable Geometry Turbocharger
$\dot{W}$	Power (W)
η_{tt}	Total-to-Static Efficiency
θ	Nozzle Vane Angle (°)
Φ	Phase (rd)
ω	Angular velocity (rd/s)

Subscripts Symbol	Definition
0	Total
A	Air
Atm	Atmospheric
c	Compressor
e	Exhaust
F	Fuel
m	Mechanical
t	Turbine

THEORY

In a typical IC engine only an approximate 40% is converted into brake power from the amount of chemical energy available prior to combustion. Of the 60% of unutilised energy, approximately 35% is available in the exhaust gas stream. Turbochargers exist as the devices that capitalise upon this 35% of energy loss by helping to recover as much of this amount of energy as possible. Generally, it may be stated that the proportion recovered out of this 35% of energy lost during one cycle of combustion in an engine in a typical efficient turbocharger of today does not exceed 40%, Dye [1], although variations are significant between engine types.

The power of the turbocharger turbine is mainly derived from the pressure drop that can be achieved across it as we can see in the power equation of a turbine, Watson and Janota [2], coming from the Euler equation [1]:

$$\dot{W}_t = \dot{m}_e . C_{p,e} . \eta_{t-s,t} . T_{0e} . \left[1 - \left(\frac{P_{s,atm}}{P_{0e}} \right)^{\frac{\gamma_e - 1}{\gamma_e}} \right] \qquad [1]$$

Where $\dot{W}_t$ is the turbine power, $\dot{m}_e$ the mass flow rate of the exhaust gas stream, $\eta_{t-s,t}$ is the turbine total-to-static efficiency, T_{0e} is the total turbine inlet temperature and $\left(\frac{P_{0atm}}{P_{0e}} \right)$ is the inverse expansion ratio.

As the compressor and the turbine are mechanically linked it the following equation of the turbocharger power balance may be derived after accounting for the mechanical efficiency:

$$\dot{W}_c = \eta_m . \dot{W}_t \qquad [2]$$

From equations 1 and 2, above, it may be observed that the higher the pressure drop across the turbine the higher the boost pressure (and hence compressor power) will be. The purpose of

ACT, therefore, much like VGTs before was to raise the pressure levels at the turbine inlet by adjusting appropriately its area. Unlike VGT operation, however, the ACT can do so for every incoming pressure wave while the VGT relies on engine conditions to change thus altering the operation from one constant bulk average quantity of flow to another before sensing this change and providing the appropriate turbine inlet area for it.

The definition of ACT given in the Introduction points out to a number of minimum parameters required to effect Active Control of the exhaust gas flow. These parameters are in addition to any thermodynamic parameters that would need to be monitored and used to control these ACT-specific parameters. These are: (1) Waveform, (2) Frequency, (3) Amplitude, (4) Null Point and (5) Phase.

Figure 1(a) illustrates a typical pressure profile as it arrives at the turbine inlet from the exhaust valves *before* being affected by the turbine inlet area-regulating schedule of operation of either a VGT or ACT actuator (in (b))

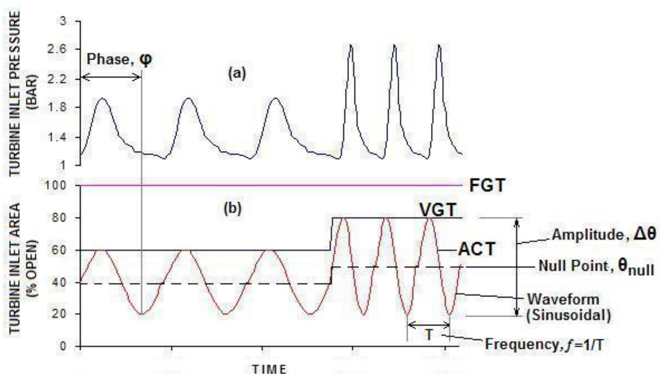


Figure 1 – Comparison of the operating profile of a non-regulated turbine inlet area (FGT) to VGT and ACT turbine inlet area regulation schedules in (b), in response to a typical pressure profile (a) as it arrives at the turbine inlet of a standard FGT from the exhaust valves. The important parameters affecting ACT operation are illustrated as well.

The important parameters affecting ACT operation are illustrated in Figure 1 as well. The pressure profile in the exhaust manifold is illustrated over a period of three pulses with the same characteristics and a different period of three pulses with the same characteristics between them but different to the previous set of pulses. The two sets are joined by a sharp transient change. The second set of pulses exhibits a higher peak pressure and, therefore, illustrates, ostensibly (and not necessarily only that in practice), an engine load change while the second set of pulses exhibit shorter durations (periods) which suggest a higher engine speed. This arrangement of pulses of instantaneous, transient response is only used to illustrate the way in which a typical VGT actuator would respond (which by definition would only provide area regulation when an engine operating point change occurs).

Once the maximum inlet area has been found (through the VGT simulation in this case) other parameters have to be optimized in order to improve the performance of the ACT-equipped engine in relation to the reference VGT-equipped engine. The instantaneous position of the flow restrictor is evaluated from Equations 3 and 4, below:

$$\theta_{ACT} = \theta_{ACT,\min} + a\left[1 - \left(\sin(\omega t)/\sin(\omega t)_{\min}\right)\right] \quad [3]$$

Where,

$$a = \theta_{ACT,\max} - \theta_{ACT,\min} = \theta_{VGT} - \theta_{ACT,\min} \quad [4]$$

θ is the rack position, ω the angular velocity and t the time.

The frequency of emitted pulses into a turbocharger is dependent on the engine speed N_{eng} (rpm), the number of strokes, n (4 or 2-stroke), the number of groups connected to a turbine entry, *G*, and the number of cylinders in a group (3 cylinders), *C*, and is given by Equation 5, below:

$$f = \frac{2N_{eng}CG}{n} \quad [5]$$

For the engine used, C=3, G=2 and n=4. For the range of speeds in this case, the frequency of the wave is between 40 and 100Hz.

The choice of flow restrictor phase (ωt) is primary importance to the efficient operation of the ACT. The original idea for ACT was to match the incoming pressure wave peak with the peak of the sinusoidal area modulation (see Figure 1). The way the phasing was carried out is presented in the following sections on model constructrion.

THE MODEL

MODEL CONSTRUCTION

Simulation of the engines was carried out by modelling and equivalent engine for the turbocharger size of interest (i.e., similar in size to the ones typically tested experimentally in the test facility of Imperial College). The engine specification used as a reference for the creation of the engine model was that of a commercial 10 litre turbo-diesel engine used in conjunction with a suitable Variable Geometry Turbocharger (VGT) for which performance maps were available. The 8 litre engine model used later for the evaluation of the ACT potential for downsizing was very similar to the 10 litre model (taken from the same commercial family of engines) with only the geometrical features of the engine changed as required. The engine models were constructed using a commercial one-dimensional software package for engine simulation with a fixed geometry turbocharger (FGT) which was prepared for as a

VGT but instead of providing different turbine inlet areas at different engine speed and load conditions, only the fully open (100% open) turbine inlet area case map was provided throughout the engine operating range.

The engine geometrical and other specification data were input and the model was run in order to obtain the difference in performance between the known VGT-equipped, engine performance data and the simulated FGT-equipped, engine data. The engine principal inputs are indicated in Table 1, along with the constraints in limiting exhaust flow and boost pressure and exhaust and boost temperature.

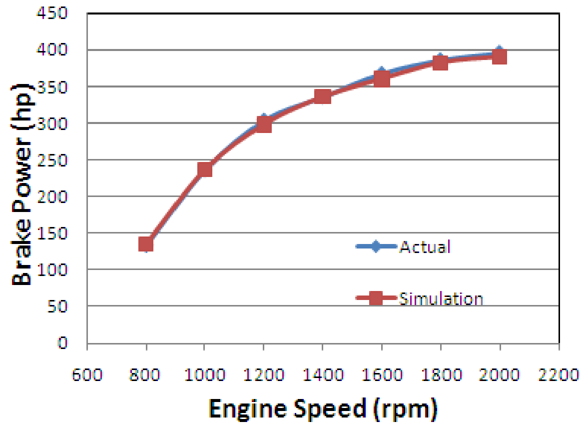
Table 1 - Principal characteristics of engine and turbocharger in the simulation program

Engine Parameter	8 l	10 l
No of turbine entries	2	2
No. of groups connected to a turbine entry, G	1	1
No. of cylinders per group (G), C	3	3
No. of strokes, n	4	4
Speed Range N_{eng} (rpm)	800-2000	800-2000
Total Displacement (l)	7.79	10.3
Compression Ratio	16:1	16.5:1
Max.Air Requirement (kg/s)	0.4444	0.4444
Max. allowable boost pressure (bara)	2.60	2.75
Max. allowable boost temperature (K)	323.15	333.15
Max.exhaust temperature (K)	783.15	793.15
Exhaust Flow at Max Output (kg/s)	0.3694	0.4625
Engine mass (kg)	681	932

MODEL MATCHING

The FGT performance data were found to be inferior to the actual engine data as expected since the real engine was equipped with a VGT instead of a FGT. The next step then in the building up of a correct model of the commercial engine in question equipped with a VGT was to try and match the performance specifications (i.e., the company plots available for bmep and power) by modifying the turbocharger in the model to operate as a VGT instead. Other performance parameters applied such as torque and bmep but the way to achieve this match was to adjust the FGT model with the appropriate turbine inlet areas throughout the engine operating range (i.e., the speed range since the available data defined a maximum fuelling – load – condition at each speed in the speed range from 800rpm to 2000rpm).

The resultant performance curves obtained in the simulation matched well with the actual engine performance curves available as indicated in Figure 3 (for both brake power and bmep).



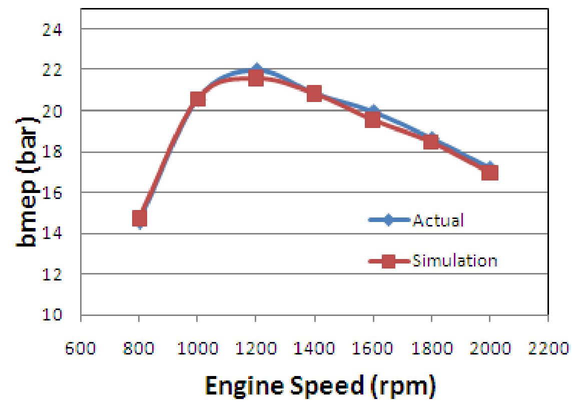


Figure 3 - (a) Brake power correlation between actual and simulated data of the VGT-equipped engine and (b) bmep correlation of the same

THE ACT MODEL

The turbine performance maps of the most efficient VGT/ACT turbocharger tested so far by Rajoo and Martinez-Botas [3] were used for five VGT (turbine inlet area) rack positions which were then normalised from fully open (1 or 100% open to 0.29 or 29% inlet open area) as in Table 2:

Table 2 - Relationship between blade angle and rack position (normalized open area)

Blade angle (deg)	Rack Position (-)
40	1
50	0.77
60	0.53
65	0.41
70	0.29

In the model an actuator was attached to the turbine for ACT operation. The actuator was triggered by either a sensor or a user-defined functions. In the case of the ACT, it seemed very convenient to employ a user-defined function to drive the actuator, in order to control the turbine inlet area as a function of a sinusoidal wave input.

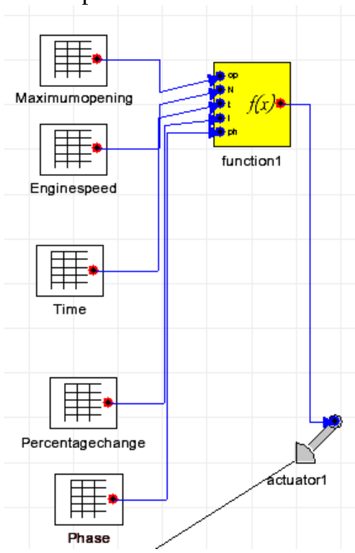


Figure 4 - ACT actuation model

In the actuation system illustrated in Figure 4 the input of the user defined function are the parameters required to calculate the value of the function. These input were the maximum opening, which is equal to the equivalent VGT rack position for the same engine condition, the rotational speed of the engine, the time of the simulation, the amplitude and phase of the motion of the actuator.

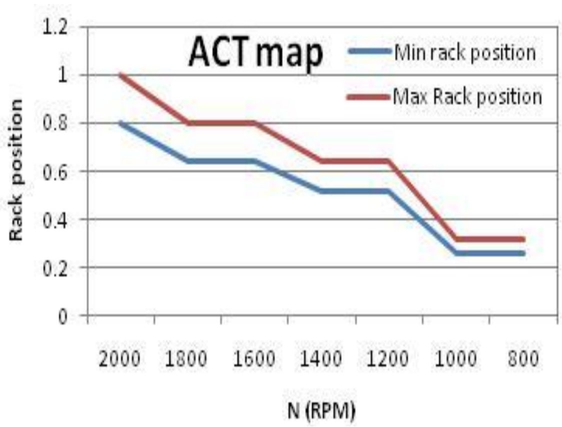


Figure 5 - ACT Rack position map. Red or maximum position coincides with VGT rack position.

The other parameter that was taken into account was the percentage of the most restricted area for the given cycle with respect to the maximum rack position. This maximum restriction was accurately chosen, in order not to overcome (through excessive turbine inlet pressure rise) the cylinder pressure limits. A study of these parameters has been done in order to find the optimum, a compromise between performance and material stress.

After having optimised this from the parametric study the map of the opening for the ACT for the reference power (maximum F/A) is provided in Figure 5.

Two ways were considered in the calculation of the minimum area, $\theta_{ACT,min}$ (see Equations 3 and 4). The first is based on the flow characteristics and was the imposed choke limit (at which the Mach number, M, equalled is 1). The ratio of the actual opening of the turbine, to the choked area, A* is given below:

$$\frac{A}{A^*} = \frac{1}{M}\left[\left(\frac{2}{\gamma+1}\right)\left(1+\frac{\gamma-1}{2}M^2\right)\right]^{\frac{\gamma+1}{2(\gamma-1)}} \quad [6]$$

For the turbine of interest, and for the rotor speed experienced in our case, the Mach number is averaged at 0.45, and therefore the ratio is around 1.5. This was in terms of averaged values and the minimum area has certainly to be tweaked in order to have the best results, but that was a very good start as it was physically relevant.

The option used for this calculation in the model and which was focussed on the engine performance, was to run a simulation with the parameter varying. In this case, the software provided the power and the torque for all the values of the parameters tested, and the one chosen was the one which provided the best performance (Figure 5).

The last parameter to be matched was the phase of the sinusoidal change of the ACT rack position to the incoming exhaust pulse. This was in order to match the peak of the sine wave, and therefore the maximum opening with the pressure peak of the pressure pulse in order not to exceed any nominal engine limits such as cylinder pressure and to create excessive backpressure in the system

Due to pressure and bulk flow travel in the exhaust manifold due to the time required for the pulse to propagate along the duct. In order to know how big the difference is, it is important to know the bulk flow velocity at the entry of the duct, assuming that the flow is fully developed already in the duct at this point, which for the range of engine speeds tested was found (after running the basic model) to lie between 36 - 85 m/s. This means that the time required for the pulse to get to the inlet of the turbine (180º past the turbine inlet casing 'tongue') is between 0.5 and 1ms.

ACT PARAMETRIC MODELLING

The primary parameter for the choice of the opening at a given load and speed is the net torque delivered to the shaft by the engine. For a truck engine, this is even more important than the horse power and is the reason for the selection of an opening based on maximising torque for a given speed. The effects of different rack positions (0.29, 0.4975, 0.645, 0.8225 and 1) have been studied throughout the speed range. The results are shown on Figure 6, below:

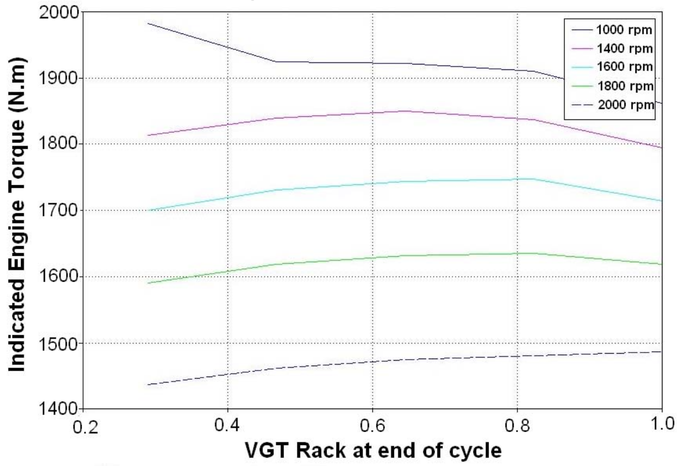


Figure 6 - Effect of rack position on torque at constant speed and a fuel-air ratio of 0.057.

The rack position, as expected, for the maximum torque, increased with speed. This is due to the fact that at a higher speed, there is an increase in the average turbine inlet pressure, and thus requiring a higher rack position in order not to restrict the inlet area of the turbine excessively and create undesirably high backpressure and corresponding pumping losses as well exceeding of the cylinder pressure limits as the boost pressure is increased.

The map of the rack position is, therefore, plotted according to the results obtained in Figure 7, below:

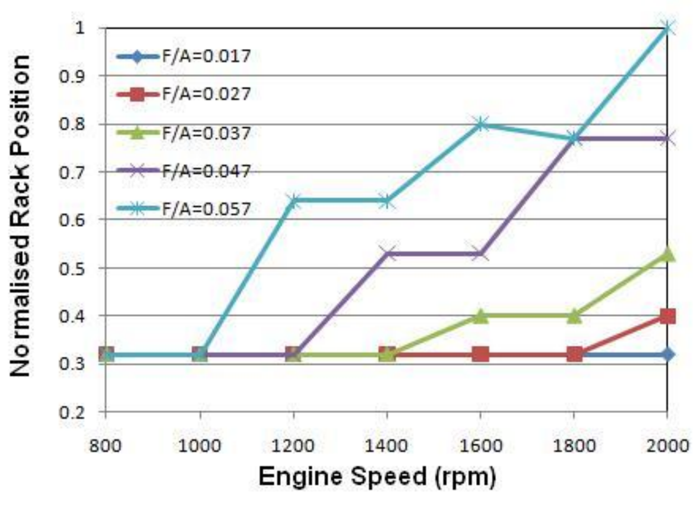


Figure 7 - Rack position map for the entire speed and load range

VGT - ACT PERFORMANCE COMPARISON ON THE SAME 10L MODEL

With the ACT parameters and their functionality identified and sufficiently optimised, the purpose of the first series of simulations was to compare the effect of ACT in relation to VGT on the same basic engine – the 10l model.

In an IC engine, as explained by Watson and Janota [2] and Heywood [4], the combustion, and therefore the power delivered by the engine are related to the amount of fuel available for the combustion. In the software, there were two different, practical ways to simulate the variation of the load of the engine. The first was to act directly on the fuel mass flow rate, with the F/A ratio being kept constant; it was thus possible to build a map of the engine load range to be simulated. The other method involved the variation of the A/F ratio. This variation changes the composition of the mixture and the heat release by the combustion and, therefore, the peak pressure in the combustion will change. The second option was chosen to simulate the load change. For a diesel engine, the F/A ratio would typically lie between 0.014 and 0.056 as and for this range five different values ratios were chosen between 0.017 and 0.057 (adjusted slightly to the characteristics of the particular engine), to give a wide enough range of values for the evaluation of the performance of the engine.

For a typical part-load case (F/A of 0.037) the effect of ACT on boost pressure may be appreciated in Figure 8, where it can be contrasted to the equivalent VGT performance.

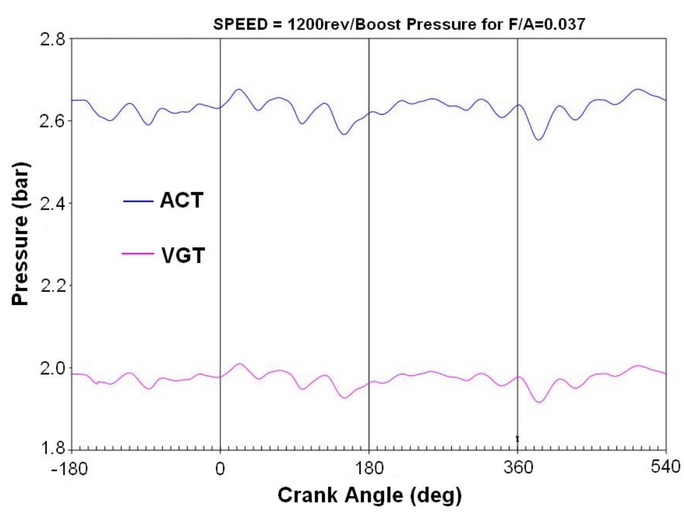


Figure 8 - Comparison of VGT and ACT Boost Pressure over a cycle for F/A=0.037

The boost pressure for the ACT is on average nearly 33% higher than the equivalent VGT value throughout the cycle.

In terms of overall results, Figure 9 provides the cumulative results for torque at all loads for both ACT and VGT:

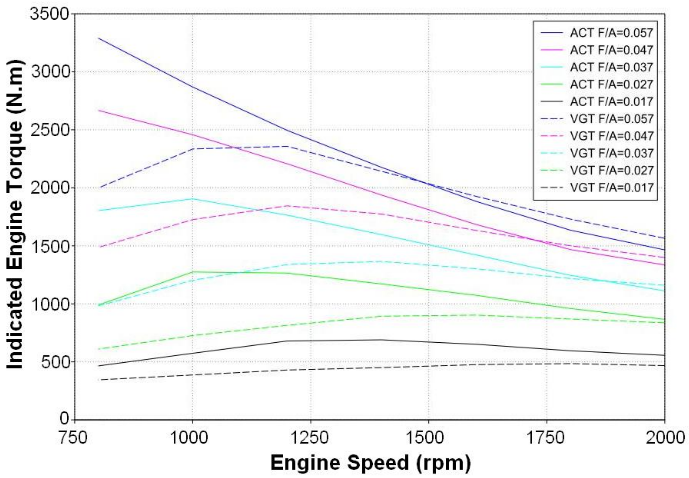


Figure 9 - Comparison of torque between ACT and VGT 10 l engine for constant load lines

As a general observation from Figure 9 it may be stated that at low engine speed, the ACT provides the greatest increase in torque. This increase is significant for all loads at low speed but is reduces as the pressure drop margins are reduced at the higher engine speeds (see Figure 5, where the area reduction is kept small so as not to exceed the cylinder pressure and boost pressure limits of the engine as per Table 1).So for this range of loads, form F/A=0.017 to F/A=0.047, the engine really benefits from the ACT with an increase of between 10 and 83%, depending on the load considered. The greatest improvement occurs between 800 rpm and 1000 rpm which is noticeable when compared to the more usual high torque region above 1200 rpm. Figure 9 would suggest, therefore, a significantly higher torque backup at the critical low speed region of engine operation.

In a similar manner, the power output of the engine is greatly enhanced particularly at low speeds (since power is directly proportional to torque), Figure 10:

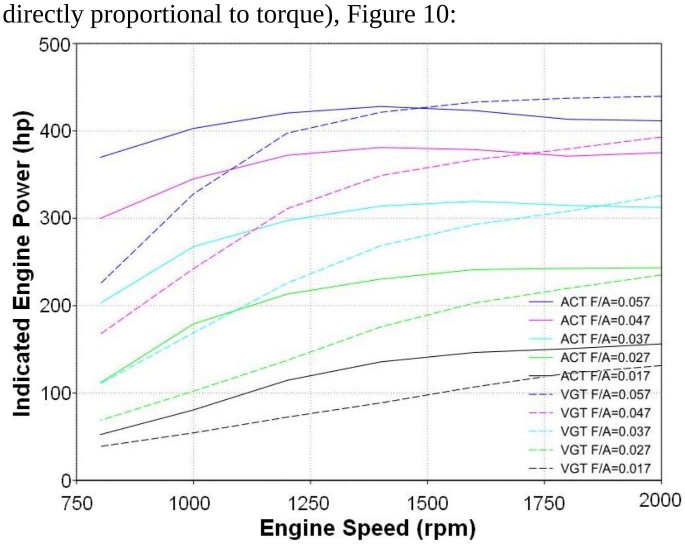


Figure 10 - Comparison of engine power between ACT and VGT engine for constant load lines

The same comments are still relevant when looking at Figure 10, above. The increase in horse power, independent of load, is very significant at the lower speeds. This is true of the entire range of speeds for the lower loads, between F/A=0.017 and F/A=0.037. For the two highest loads, F/A of 0.047 and 0.057, the improvements are significant up to 1600 rpm. It is also possible to see that the curves are much flatter, the difference in horse power between the highest and lowest speed being much lower. The engine, when coupled to an ACT is able to deliver a large amount of power even at lower speeds, and the driveability would be enhanced when accelerating at a constant rate. In the case of the VGT engine, the horse power at 800 rpm is usually approximately 50% lower that the horse power at 2000 rpm, whereas it is much higher (at least 80% of the maximum at the higher speeds) for the ACT engine.

The equivalent specific fuel consumption (sfc) trends may be observed in Figure 11:

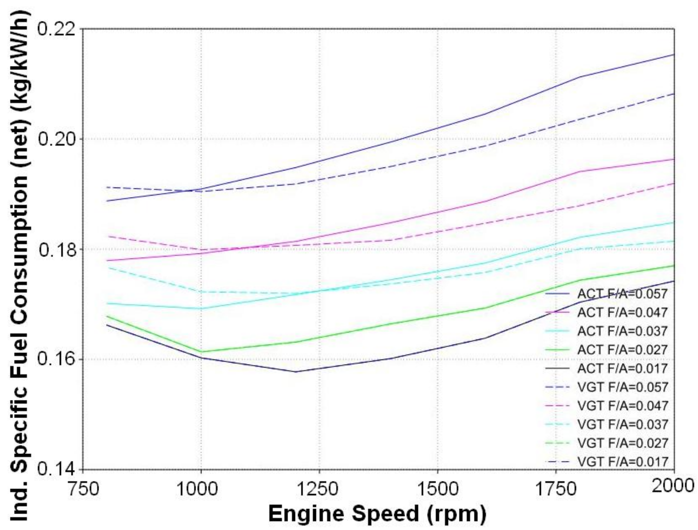


Figure 11 - Comparison of Specific Fuel Consumption between ACT and VGT engine for constant load lines

Two main trends arise from Figure 11; the first, is that at the lower loads, under F/A=0.037, the sfc between the ACT and the VGT engine is almost the same. The second trend is that at the higher loads the sfc for the ACT engine is higher throughout compared to VGT.The sfc is calculated as:

$$sfc = m_f / P \qquad [7]$$

where P is the power output of the engine. This means that the actual fuel consumption is higher throughout and in particular at the highest loads. This is to be expected however since the basic engine was of the same capacity while the boost pressure delivered by ACT in any given condition was, generally, significantly higher, hence the amount of fuel that could be introduced in the combustion chamber was higher (for the given F/A input in the program) and therefore the fuel consumption is higher (which allows the power to increase substantially).

COMPARISON OF THE 10L VGT WITH THE 8L ACT ENGINE

The other component of this study was the comparison of the performance of the ACT coupled in the model to a smaller engine (8 l) than the reference 10 l model coupled with VGT. This was chosen so as to be significantly smaller than the reference engine while still being part of the same family of engines.

The same load and speeds ranges were run for immediate comparisons to be drawn. In Figure 12, the torque levels for the 8l ACT, the 10l VGT and the 10l ACT from the previous section are presented at full load (F/A of 0057):

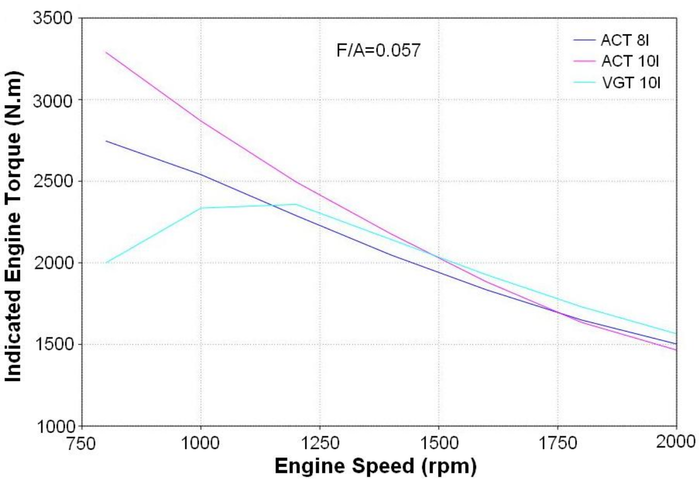


Figure 12 - Comparison of torque between 10l ACT, 8l ACT and 10l VGT engine

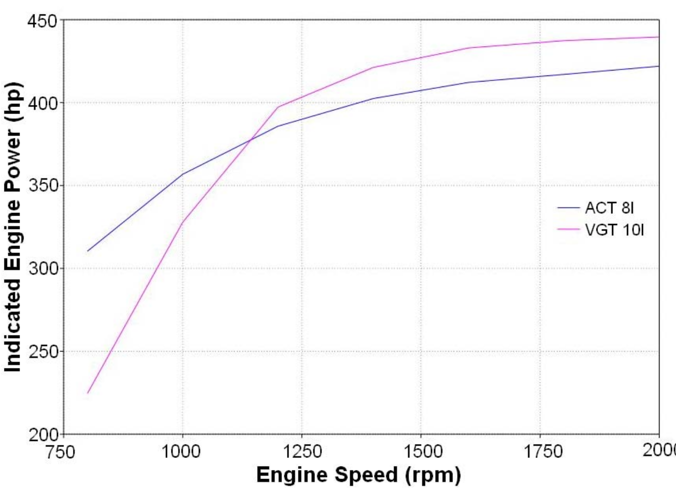


Figure 13 - Comparison of horse Power between 8l ACT and 10l VGT engine

In Figure 12, the torque delivered by the 8 litre engine, is either clearly above or at a slightly lower level than the 10l VGT engine. As for the 10l ACT, it is interesting to note the increase in the torque at low speeds, under 1200 rpm. It is interesting to see the increased torque backup exhibited by the 10l ACT retained for the 8l case (although obviously reduced in comparison to the 10l ACT case). This results in a downsized engine with up to 37% higher torque at low speed (up to 1200rpm where the performance is matched), while having only 4% to 5% less torque in the 1200rpm to 2000rpm region.

Given the dependence of power on torque for the same speeds modeled, the same conclusions can be drawn (increase of up to 37% at up to 1200rpm then a minimal drop) from the power comparison between the two engines in Figure 13.

A most important aspect in this downsizing study was the comparison in terms of fuel consumption between ACT and VGT. Given that the 8l is able to deliver the same or more power than the 10l VGT with only a minimal drop above 1200rpm, it was interesting to calculate the effect of changing the F/A ratio in order to match the power output from the two engines. The results are presented in Figure 14.

Figure 14 - Comparison between the fuel mass flow rate for the 8l ACT and 10l VGT for the same power output

The shape of the two curves is very similar. The fuel mass flow rate, however, is higher in the case of the VGT throughout by up to approximately 5% at the highest speed (2000rpm). This means that the amount of fuel required to have the same power output is higher in the case of the 10 litre engine. The reason is certainly in part due to that the combustion chamber of the 10 litre VGT engine being larger (more friction and heat losses) than the one in the 8 litre ACT engine; therefore, in order to have a mixture as rich as the one for the 8l, more fuel was required but with the air mass flow being only slightly higher for the 10l case (in proportion to the increase in the fuel mass flow required in the 10l case, for the constant F/A ratio provided in both cases) the boost pressure stayed correspondingly lower for the 10l indicating the effect of the significant boost pressure rise effected by ACT (see Figure 8).

As one might expect from Figure 14, the efficiency superiority of the ACT 8l engine can be demonstrated in terms of the specific fuel consumption performance. Figure 15 shows this difference in comparison to the VGT engine.

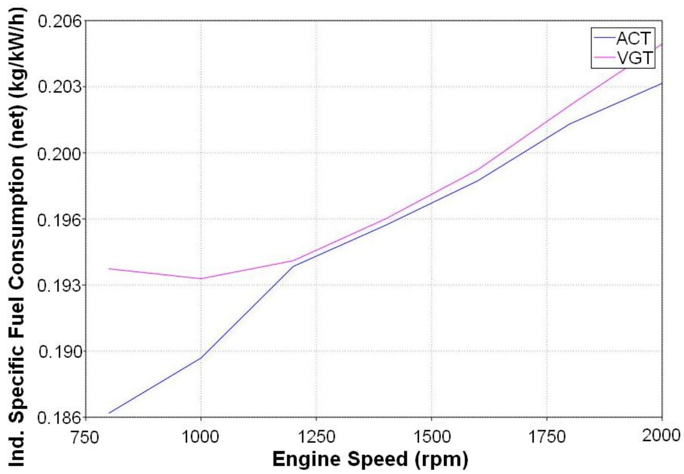


Figure 15 - Comparison between sfc for 8l ACT and 10l VGT for same torque output

The significant increase in power at the lower speeds is in evidence in Figure through a highly improved sfc (a maximum of 8.3% improvement). Beyond 1200rpm the ACT performance is more evenly matched, with a slightly higher improvement at the highest speeds (6% at 2000rpm).

CONCLUSIONS

In the present study commercial engine simulation software was used to construct and calibrate two engine models, one of 8l and the other of 10l capacity, which represented the characteristics of two consecutive modern engine models of the same family, in use today. The goal of this exercise was to confirm the positive effect of ACT on engine performance in comparison to the current state-of-the art – the VGT - to increase the power output of the same basic engine. In addition, the 8l model was coupled to the ACT and compared against the same basic 10l VGT configuration. The goal in the second instance was to investigate the ability of ACT to match the performance of the bigger VGT engine as much as possible, thereby achieving downsizing of the larger engine through the replacement of the turbocharger on a unit-for unit basis.

In the first case, it was proven that an engine coupled to ACT is able to deliver significantly more power, in comparison to the same 10l VGT engine, at the lower end of the speed range while the power difference is smoothed out at the higher end of the speed range so as not to exceed nominal engine cylinder pressure limits.

In the second case, the 8l was matched reasonably well and would be in a position to replace the 10l in practice. Two consistent results emerged: a clear ACT superiority at speeds of up to 1200rpm and matched results to slightly inferior above that speed to the maximum speed of 2000 rpm. Given the relative size and mass of the two engine (932 kg to 681 kg for the 10l and 8l models, respectively) a saving of 27% in terms of mass could be achieved with an engine at least as good overall (in the 8l case) to the larger 10l engine equipped with the production VGT. In addition a benefit in terms of size from the smaller 8l would emerge thus offering improvement in packaging terms. In terms of driveability, this would improve in the smaller 8l ACT since the torque backup at the lower end increased, perhaps making this even better suited for off-highway applications.

Applications of this device are in following with traditional VGT application areas in both gasoline and diesel applications and across the automotive, marine and industrial sectors. Suitable control strategies and real world hardware requirements are as of this writing under development.

REFERENCES

[1] Dye, A., 2006, "A novel method of high efficiency pressure charging", Conference proceedings of the 8th International Conference on Turbocharging and Turbochargers, IMechE., pp 305-316.

[2] Watson, N., and Janota, M.S., 1982, "*Turbocharging the Internal Combustion Engine*", Wiley Interscience, New York.

[3] Rajoo, S., and Martinez-Botas, R., "Experimental Study On The Performance Of Variable Geometry Mixed Flow Turbine For Automotive Turbocharger", 8^{th} International Conference on Turbocharging and Turbochargers, IMechE., 17-18 May 2006.

[4] Heywood, J., 1988, "*Internal combustion engine fundamentals*", McGraw Hill Inc.