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Direct Comparison of an Engine Working under Otto, Miller and Diesel cycles: Thermodynamic Analysis and Real Engine Performance

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ABSTRACT

One of the ways to improve thermodynamic efficiency of Spark Ignition engines is by the optimisation of valve timing and lift and compression ratio. The throttleless engine and the Miller cycle engine are proven concepts for efficiency improvements of such engines.

This paper reports on an engine with variable valve timing (VVT) and variable compression ratio (VCR) in order to fulfill such an enhancement of efficiency. Engine load is controlled by the valve opening period (enabling throttleless operation and Miller cycle), while the variable compression ratio keeps the efficiency high throughout all speed and load conditions.

A computer model is used to simulate such an engine and evaluate its improvement potential, while a single cylinder engine demonstrates these results.

The same base engine was run on the test bench under the Diesel cycle, Otto cycle and Miller cycle conditions, enabling direct thermodynamic comparisons under a wide variety of conditions of speed and load.

The results show a significant improvement of the Miller cycle over the Otto cycle engine. Comparisons of the Miller engine with the Diesel engine shown that it is possible to have a SI engine with better efficiency than a similar Diesel engine for most of the working conditions.

INTRODUCTION

Engine research for automotive purposes focuses its action on solving problems related to energy use and pollution. Analyzing how energy is spent in car engines shows that the conventional Otto cycle engine is used with reduced efficiency for a significant part of the driving time. During most of the time the engine works at part load with low efficiency, decreasing even more as the load is further reduced [1,2]. Usually, engines are designed to have the best performance at a certain working conditions. When they are operated at different conditions the performance is poorer. Variable configuration engines must be used to improve efficiency throughout the working range of the engine. Variation of valve timing and lift and compression ratio are significant parameters that can contribute to engine performance improvement.

Valve event variation is one of the research fields more exploited in engine technology. Several variable valve actuating systems have been proposed [3-5] and results reported. The variation of the intake valve closure timing on itself can produce variations of the amount of air/fuel mixture trapped within the cylinder in each engine cycle. The substitution of the throttle valve, which is one of the main causes of the reduced cycle efficiency at reduced loads, by a variable valve timing (VVT) system for load control has been presented widely [6,7], with favorable results.

The thermal efficiency of Otto cycle is theoretically expressed as:

$$\eta = 1 - \frac{1}{\varepsilon^{\gamma-1}} \quad (1)$$

It can be seen from (1) that, at least theoretically, the main variable that influences the thermal efficiency is the compression ratio. Its increase is a benefit for the thermodynamic performance of the engine, but a limit is imposed by engine knock. If compression ratio could be adjusted to the cycle conditions, setting it to the maximum value before knock occurrence, significant improvement would be achieved. Several systems for compression ratio variation were reported and implemented [8-11], and benefits were quantified.

In a previous work [2] this team developed a computer model to predict engine improvements using VVT and variable compression ratio (VCR) simultaneously. Then a single cylinder engine was modified to work under these conditions and tests were conducted. In order to produce different compression ratios several pistons were used enabling results similar to a variable compression ratio engine. Also, different camshafts were used to perform intake valve timing variation similar to a VVT engine.

COMPUTER MODEL

A computer model was used to simulate the engine performance and preview some preliminary results of the performance with different cam shapes and different compression ratios. The program presented elsewhere [12], is a one zone model, with the combustion following a Wiebe function. It includes Annand heat transfer coefficient [13], where average constant temperatures are considered for the cylinder head, cylinder walls and piston head. It includes a model for friction mep calculation [14,15]. Mass flow entering and leaving the cylinder is simulated using a compressible flow model [16]. The model also considers variable gas properties as a function of in-cylinder temperature [17,18].

The model calculates instant cylinder pressure and temperature, from which it is capable of calculating several engine performance parameters, like power (work), thermal efficiency, specific fuel consumption, indicated and break mean effective pressure. The model includes the calculation of engine friction, so bmep can be calculated.

For the simulations an engine configuration was set, similar to the real engine on the test bench. The engine characteristics are shown in table 1 for the Otto engine. In the real engine the combustion chamber has a deep toroidal bowl combustion chamber shape, while in the computer model a simplification was introduced and a cylindrical combustion chamber shape was used.

Table 1: Engine configuration data.

Number of cylinders	1
Bore [mm]	70
Stroke [mm]	55
Connecting rod length [mm]	90
Combustion chamber height [mm]	5.3
Total displacement [cc]	211
Compression Ratio	11.5
Intake Valve Diameter [mm]	30
Exhaust Valve Diameter [mm]	25
IVO [CA]	20 BTDC
IVC [CA]	32 ABDC
EVO [CA]	40 BBDC
EVC [CA]	12 ATDC

COMPUTATION RESULTS

The preliminary computational simulations were made considering the following assumptions:

Combustion starts at 20 BTDC at takes 40 CA to finish and is considered as a complete and stoichiometric combustion. Gasoline was the fuel selected for the performed simulations.

Temperatures from the cylinder walls are considered constant during all the cycle.

For the Otto cycle at WOT the maximum temperature of the cycle is 2082 K and peak pressure is 75 bar. This temperature and pressure was used as a reference for the compression ratio adjustment when intake valve closure (IVC) time was changed. As presented in another work [1] if only the intake valve closing time changed, an increase of the efficiency is expected in relation to the Otto cycle, but the efficiency still decreases with load reduction. If compression ratio is adjusted then an improvement is possible in terms of thermal efficiency with the decrease of load, which is the main aim of this research.

Figure 1 shows the specific fuel consumption versus engine load for the three SI engine versions: Otto, Miller and Miller VCR. It can be seen that there is an improvement just by the use of a variable valve timing system to control the load via late intake valve closure (Late IVC). When compression ratio adjustment is used, the benefit in terms of specific fuel consumption becomes much more important. It is relevant to refer that with the Miller VCR engine the specific fuel consumption decreases as load is reduced from full throttle down to approximately 7 bar bmep, from 249 g/kWh down to 237 g/kWh.

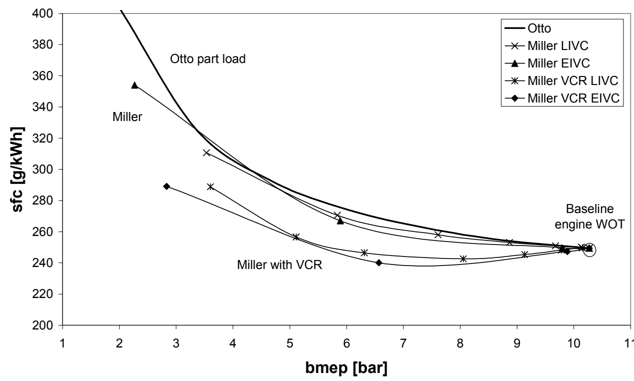

Figure 1 – Specific fuel consumption as a function of load for 2500 rpm simulations.

At very low loads or at idle, the compression ratio required to keep the Miller VCR engine strategy working might be so high that it becomes physically impossible to realize it, because there is no space between piston and poppet valves. In the case of idle the use of the throttle valve may be required when the Late IVC strategy is used, as the delay required to the intake valve to close is so high that the ignition would occur with the intake valve still open [3].

In the case of Late IVC the minimum load presented in figure 1 is close to the minimum possible due to the referred problem of ignition during open intake valve. If early intake valve closure (Early IVC) is used for load control it is possible to achieve much lower loads.

The engine, either working as simple Miller or as Miller with VCR has better efficiency if Early IVC is used instead of Late IVC. This is mainly caused by reduced pumping losses. In the case of Late IVC the air/fuel mixture in inducted into the cylinder and after the BDC it is blown-back again to the intake manifold. This effect is more intense as longer IVC delays are used.

ENGINE MODIFICATION

To perform a direct comparison between a Diesel engine and a SI engine it was decided to use a Diesel engine and after measuring its performance, modify it to work as a spark ignition engine. With this modification the engine is capable to work as a SI engine, but several design characteristics are still optimized to work under the Diesel.

The Diesel engine used was a Yanmar L48 AE (figure 2). The commercial Diesel engine specifications are presented in table 2.

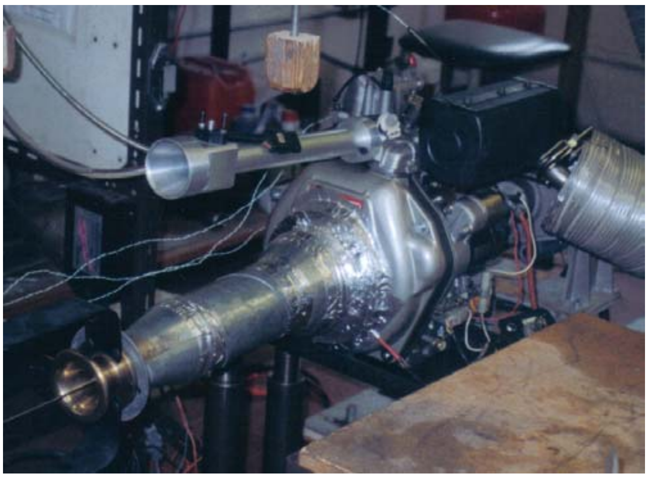

Figure 2: Yanmar engine on the test bench.

In order to use the same DI Diesel engine as a SI engine several modifications were made. At the engine head the direct injection fuel injector was removed and a spark plug was placed at the same position. A long intake manifold with a throttle body was placed upstream of the intake duct. The intake manifold also included a fuel injector positioned upstream of the intake valve. The intake manifold volume was sized large enough to avoid fuel loss during the blow-back when using the Late IVC strategy. The first piston was modified to have a wide combustion chamber and a compression ratio of 11.5:1.

An electronic control unit was used to control spark timing and fuel injection and a pressurized fuel circuit was connected to the low pressure petrol injection. An ignition coil and an ignition module were necessary for the ignition system operation.

Table 2: Yanmar engine commercial specifications.

Model	L48AE	
Type	Single-cylinder, vertical 4-cycle diesel	
Combustion system	Direct injection	
Displacement [cc]	211	
Engine speed [rpm]	3000	3600
Maximum output [hp]	4.2	4.7
Continuous output [hp]	3.8	4.2
Cooling	Forced air	
Lubrication	Oil pressure	

The first attempt to run the engine in SI mode was not completely successful due to the excessive swirl generated by the original (Diesel) intake duct in the engine head. The engine run without burning all the fuel, emitting large quantities of HC. As it can be observed in the spark plug (figure 3), the burning zone was only on one side of the spark plug showing the direction of the extremely high swirl movement inside the cylinder.

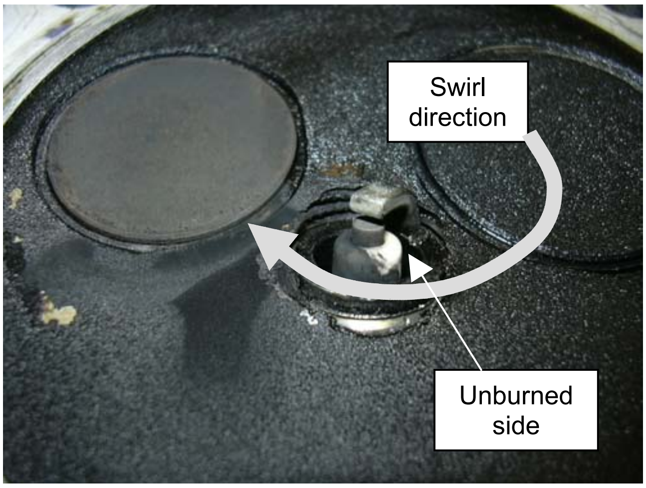

Figure 3: Burning zones on the spark plug, after first running attempt.

So the engine head was mounted on a visualization rig, described in figure 4, to enable the study of the generated swirl within the engine cylinder. Attached to the head was a glass tube with the diameter of the engine cylinder, but much longer. It was possible to notice the helix forming by the inlet air, when injecting smoke within the air stream. A paddle wheel was also used to measure the swirl index, by measuring the rotational speed at several distances from the engine head (56 mm, 76 mm, 96 mm).

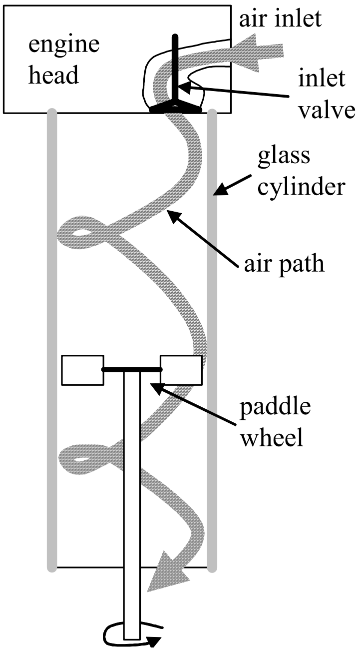

Figure 4: Testing rig for swirl measurement.

The swirl reduction was achieved by means of intake duct modification. The original shape had a channel for swirl induction in the air flow and a deflector at the channel end. The first modification made to the intake port was to create a deflector in the opposite side of the existing one. The existing deflector was eliminated and the added deflector was built in such a way that it interrupted the air flow at the entrance of the swirl induct channel. This channel was also filled in order to reduce swirl movement of the incoming air. As described above the swirl was measured at several distances from the engine head and the results are presented in Figure 5. Swirl index was calculated using the definition of Annand and Roe [19]. The reduction of the swirl, depending of the distance at which it is measured and the air flow, goes from 60% up to 80%.

The discharge coefficient of the induction port changed and the modification of the intake port was optimized in order to minimize the increase of the pressure drop in that passage. In fact the discharge coefficient increased slightly but the difference is acceptable for the case of an intake port. Figure 6 shows the discharge coefficient with and without the modification and its evolution as a function of the air mass flow.

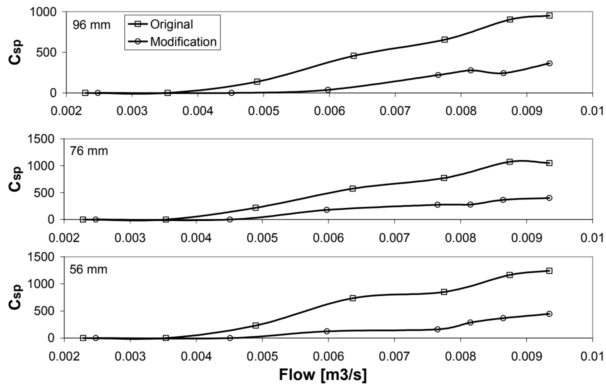

Figure 5: Swirl results after and before the intake duct modification.

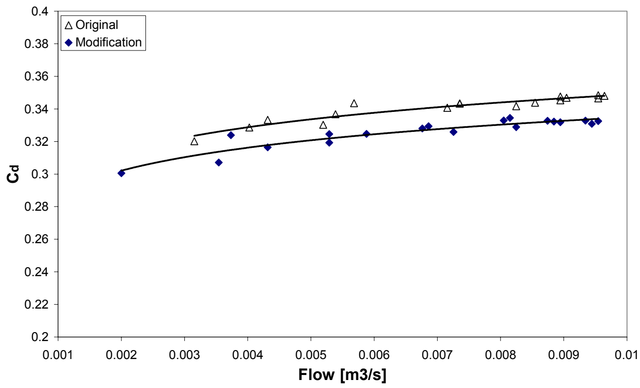

Figure 6: Original and modified discharge coefficient of the intake duct and valve.

ENGINE TESTS

The different cams that were used in the engine had either Late IVC with different dwell angles or Early IVC with different opening periods. The Late IVC cams had dwell angles of 20, 40, 60 CA. Two Early IVC cams were tested to evaluate and compare the ability of each load control strategy to reduce the engine load, as shown in figure 7.

The exhaust timing and the intake valve opening position was kept always the same. Figure 7 shows the lift of the valves during the cycle completion for the different camshafts used to achieve different loads. Figure 8 presents the volumetric efficiency of each camshaft, calculated with the test results performed at stoichiometric conditions. As it can be seen the Early IVC strategy allowed much lower loads than Late IVC.

To achieve the variation of the compression ratio, different pistons were used. As the original engine is a DI Diesel engine, the original pistons were used, and the piston bowl of standard Diesel piston was enlarged to create different combustion chamber volume and therefore different compression ratio. Table 3 reports the piston sizes and compression ratios, including the original Diesel engine piston. The original configuration of the engine, as a Diesel, had a compression ratio of 19.9:1. The piston had a deep toroidal bowl combustion chamber. With this kind of design it is possible to enlarge the combustion chamber in the piston to a cylindrical bowl chamber (Figure 9). This combustion chamber shape is not the ideal for SI engines, but this was the possible chamber to be manufactured from the original combustion chamber (Diesel) of the engine. Figure 10 shows the several pistons used in the SI engine tests, from the Otto engine piston with a compression ratio of 11.5:1 up to a 17.5:1. Also, for comparison, the standard Diesel piston with a compression ratio of 19.9:1 is included.

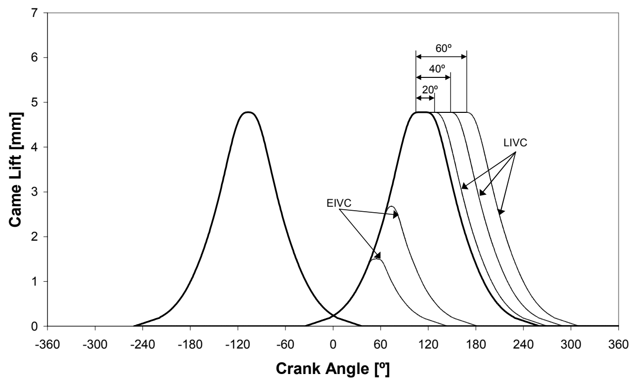

Figure 7 – Different cam profiles tested.

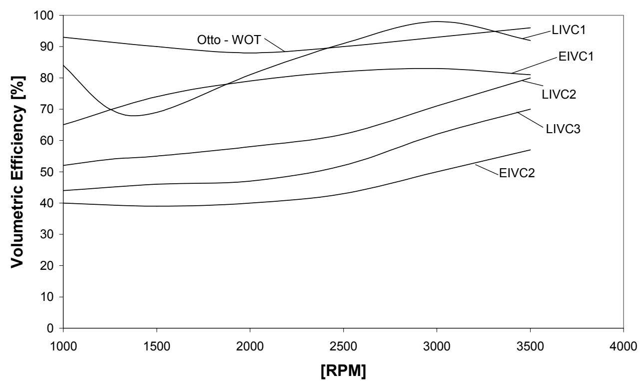

Figure 8: Volumetric efficiency of each camshaft.

Table 3: Pistons specifications.

Piston	Combustion chamber diameter [mm]	Compression Ratio
1	48.4	11.5:1
2	46.0	12.5:1
3	43.8	13.5:1
4	41.8	14.5:1
5	40.1	15.5:1
6	38.5	16.5:1
7	37.1	17.5:1
Diesel	-	19.9:1

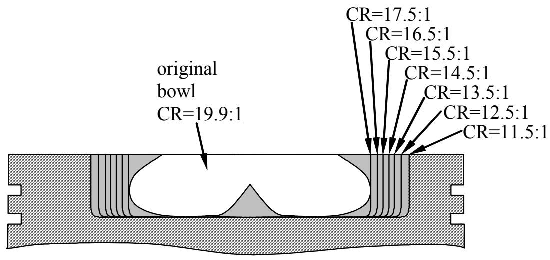

Figure 9: Original and modified combustion chambers.

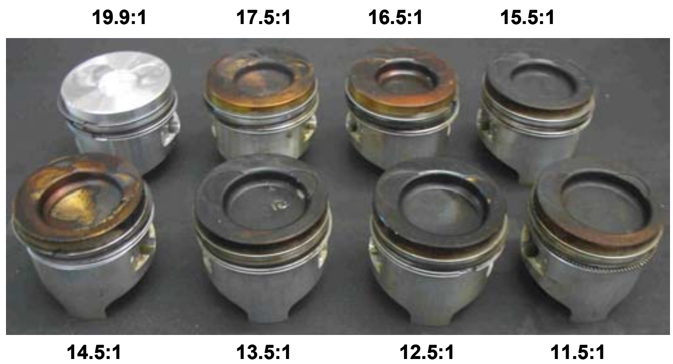

Figure 10: Pistons used in tests.

TEST RESULTS

Initial tests were made [20] with the engine in its commercial (standard) Diesel version, so that the results could be used as a baseline for comparison and evaluation of improvements. The measured Diesel engine specific consumption map is shown in figure 11.

After the engine conversion and preliminary tests, the Otto engine version was completely mapped from 10% of throttle up to WOT. The resulting specific fuel consumption map is presented in figure 12. As expected, the engine modification from Diesel to SI lead to a maximum torque and maximum power improvement

41 % and 49 %, respectively. Please note that the maximum speed of the engine was similar for Diesel and SI versions.

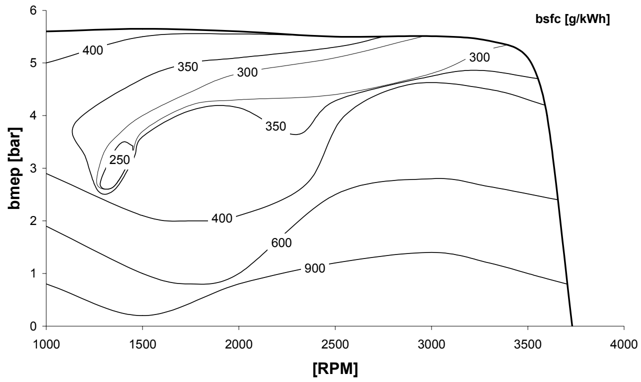

Figure 11: Diesel engine specific consumption map.

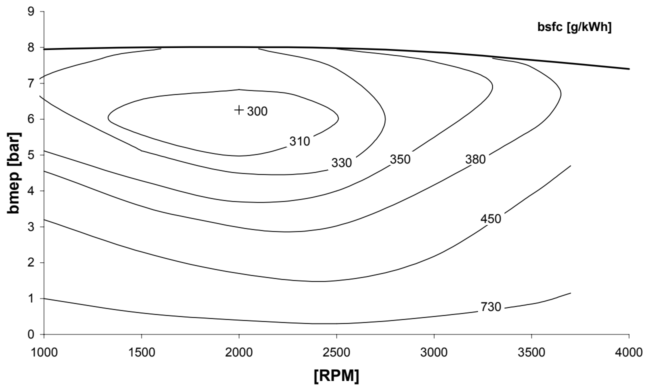

Figure 12: Otto engine specific fuel consumption map.

Then the engine was tested using the 11.5:1 CR piston with all the different camshafts at WOT conditions (Miller engine). A specific fuel consumption map (figure 13) based on the data from both the Late IVC and Early IVC camshafts was build. This engine map represents the Miller engine with constant compression ratio.

Comparing the Miller engine with the Otto engine an improvement in terms of torque and bmep can be seen from 2000 up to 3500 rpm. This improvement is caused by the performance of the first Late IVC camshaft, which has better volumetric efficiency at these speeds.

Superposing the maps of the Otto engine and the Miller engine (figure 14), it can be seen that from full down to 30% of load there is always improvement when using the Miller cycle. Bellow 30% of load, the improvement is only achievable at certain engine speeds. This load region, in the case of the Miller engine, was achieved with the Early IVC camshaft with less overture and at these conditions the engine run particularly “rough” and lacked stability of running.

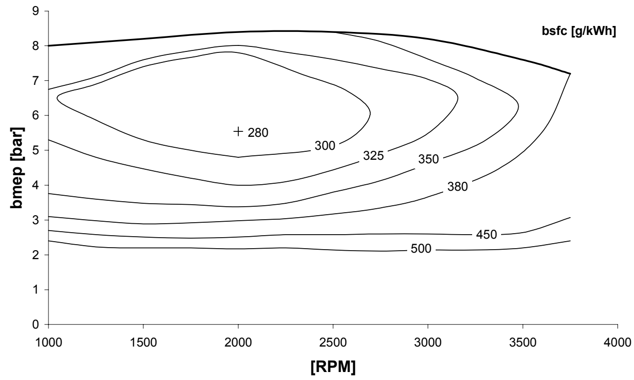

Figure 13: Miller engine specific fuel consumption map.

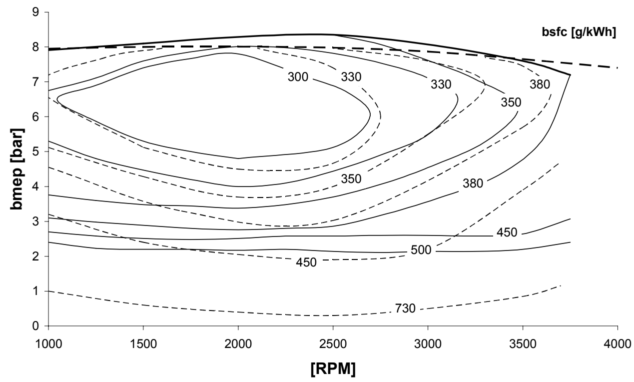

Figure 14: Comparison of the Otto and Miller engine. Continuous line for Miller engine; Dashed line for Otto engine.

The last set of tests was made using the various camshafts and the different pistons. For each camshaft the compression ratio was sequentially increased until audible knock was detected. Table 4 shows the performed tests for each camshaft and compression ratio (grey squares). The best conditions for sfc are represented in dark squares. For the camshafts producing the lower loads (LIVC3 and EIVC2) the compression ratio was increased, without achieving audible knock with rare exceptions. The tests were halted at a compression ratio of 17.5:1, because in both camshafts the torque decreased and the specific fuel consumption of the engine increased.

Figure 15 shows the torque and specific fuel consumption for the EIVC2 camshaft with different compression ratios for 2500 rpm. The decrease of the torque value after this point may be explained by an increase of the engine internal friction as a result of the higher compression ratio.

Table 4: Performed tests for the Miller VCR engine.

Cam	Compression ratio						
	11.5	12.5	13.5	14.5	15.5	16.5	17.5
LIVC1							
LIVC2							
LIVC3							
EIVC1							
EIVC2							

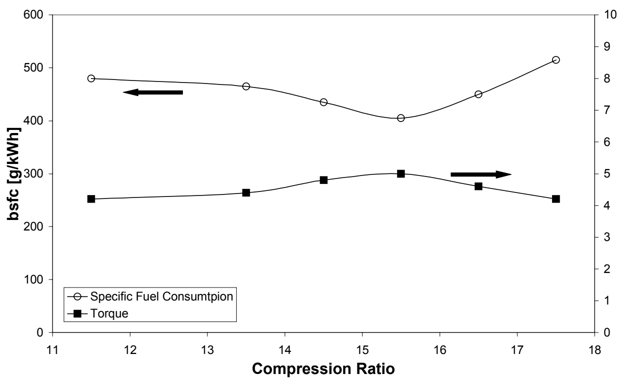

Figure 15: Torque a sfc as function of compression ratio.

Figure 16 presents the fuel consumption map for the Miller VCR engine. For this map the values of the Miller VCR engine using the best efficient camshaft/piston were selected. These optimum points are shown in table 4 with dark squares.

The improvement of the Miller VCR engine can be evaluated by making the comparison between that engine and the Otto engine. Figure 17 presents the difference of sfc in relative terms. It can be seen that the best improvement is reached for lower loads and speeds. For engine speeds lower than 2500 rpm and loads lower than 6 bar bmep, the improvement can go up to 24%. At the high speed and high load the Miller VCR engine loses efficiency when compared with the Otto engine, probably due to lower combustion stability. At this speed range, the dynamic effects of air intake can lead to different working conditions that may require other intake valve timing strategy and compression ration variation.

As noticed for the Miller engine (Figure 14), the Miller VCR shows an increase of torque from 1000 up 3500 rpm engine speed range. This can be explained by the torque increase due to the increase of the compression ratio and at the same time an improved volumetric efficiency achieved with the modified intake cam, LIVC1. This cam is also the cause for the decrease of efficiency at the high speed and high load region.

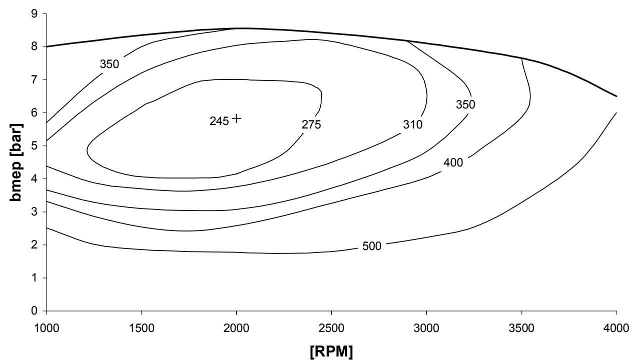

Figure 16: Miller VCR engine specific fuel consumption map.

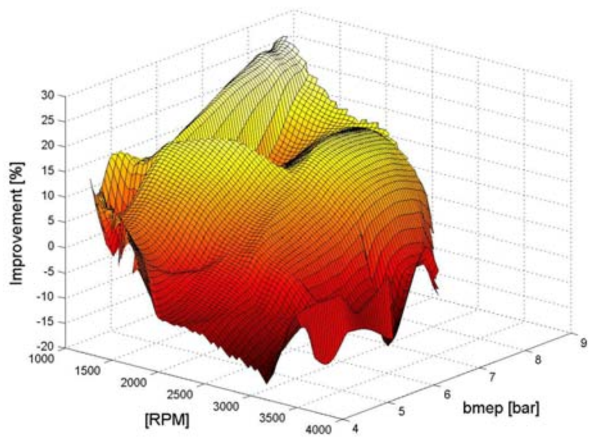

Figure 17: Sfc improvement of the Miller VCR over the Otto engine.

Figure 18 presents the bsfc curves for the three different engines. It can be seen that the Miller VCR engine is always more efficient than the Otto engine and Miller engine for loads higher than 50% (4 bar bmep). At 2500 rpm the improvement from Otto to Miller VCR engines reaches 18 %.

Comparing curves from figure 18 with those from figure 1 it can be seen that Miller VCR bsfc increases more at low loads that simulation results predicted. Again here the explanation seems to be the significant increase in the engine internal friction and combustion deterioration.

Figure 19 presents the improvement from the Diesel engine to the Miller VCR engine in terms of specific fuel consumption. The maximum improvement happens for the maximum loads of the Diesel engine, 4 to 6 bar of bmep and for engine speed from 1500 rpm to 2500 rpm. This is the region where the Miller VCR engine performs better (Figure 16) and complementarily, the Diesel engine runs very unstable. For higher speeds the Miller VCR engine loses efficiency when compared to the Diesel.

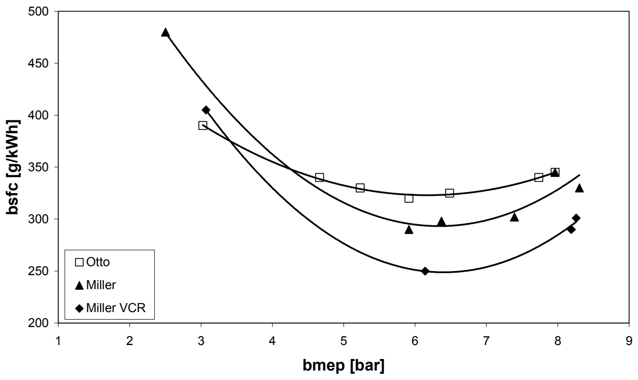

Figure 18: Brake specific fuel consumption for the tested engines at 2500 rpm.

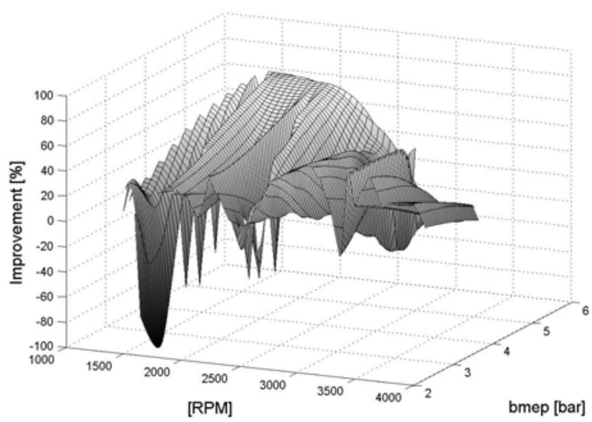

Figure 19: Improvement from the Diesel engine to the Miller VCR engine.

To reduce the number of optimizing variables, all SI engines tests used stoichiometric conditions. Therefore it would be possible to further improve these engines (Miller) in terms of thermal efficiency by the use of lean and extra-lean mixtures. This way the improvement comparing to the Diesel engine would be further extended.

The modification of the intake duct geometry lead to a significant reduction of the swirl within the cylinder. This was important for the Otto cycle but for the Miller and Miller VCR cycle it may have influence on the combustion conditions especially at the lowest loads. This may explain the poor performance of the Miller VCR engine at low loads.

CONCLUSIONS

From the work described above it can be concluded:

A computer model was used to simulate spark ignition engines with different load control strategies, namely the Otto cycle (using a throttle valve), a conventional Miller cycle (with VVT) and an improved Miller cycle (with VVT and VCR). The latter engine cycle proved to be the most efficient.

The same single cylinder engine was successfully run under the Diesel, Otto and Miller cycles, enabling a direct comparison between the performances of these cycles. This engine was originally a DI Diesel engine that was later modified to run as SI engine. Under the Miller cycle, this engine run with different camshafts and different piston geometries which, in reality, transformed it into a VVT and VCR engine.

The Miller engine with VCR proved to be much more efficient than the Otto engine for most of the working range. Differences of specific fuel consumption between these engine configurations were actually higher than 20% for low speeds and low loads.

Comparing the Miller engine with VCR with the Diesel engine, improvements in specific fuel consumption could be noticed for most of the speed/load range. The peak improvements occurred for the conditions where the Miller engine excels, which coincidentally is a working region where the Diesel engine runs poorly.

Miller VCR cycle engine proved to be the best option between the spark ignition and compression ignition cycle engine for part load applications.

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ACRONYMS

ABDC: After Bottom Death Center
ATDC: After Top Death Center
BBDC: Before Bottom Death Center
BMEP: Break Mean Effective Pressure
BSFC: Brake Specific Fuel Consumption
BTDC: Before Top Death Center
CA: Crank Angle
CR: Compression Ratio
EIVC: Early Intake Valve Closure
EVC: Exhaust Valve Closure
EVO: Exhaust Valve Open
IVC: Intake Valve Closure
IVO: Intake Valve Open
LIVC: Late Intake Valve Closure
MBT: Maximum Break Torque
MEP: Mean Effective Pressure
SFC: Specific Fuel Consumption
SI: Spark Ignition
VCR: Variable Compression Ratio
VVT: Variable Valve Timing
WOT: Wide Open Throttle