

# Feasibility study of Operating 2-stroke Miller Cycles on a 4-stroke Platform through Variable Valve Train

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## Abstract

A 2-stroke combustion cycle has higher power output densities compared to a 4-stroke cycle counterpart. The modern down-sized 4-stroke engine design can greatly benefit from this attribute of the 2-stroke cycle. By using appropriate variable valvetrain, boosting, and direct fuel injection systems, both cycles can be feasibly implemented on the same engine platform. In this research study, two valve strategies for achieving a two-stroke cycle in a four-stroke engine have been studied. The first strategy is based on balanced compression and expansion strokes, while the gas exchange is done through two different strokes. The second approach is a novel 2-stroke combustion strategy - here referred to as 2-stroke Miller - which maintains the expansion as achieved in a 4-stroke cycle but suppresses the gas exchange into the compression stroke. The first 2-stroke cycle generated a torque increment of 63% at 1000 rpm on a supercharged 4-stroke engine without increasing the maximum cylinder pressures. The second 2-stroke strategy - which uses the Miller cycle concept - generated a torque increase of 51.5% at 1000 rpm and higher thermal efficiencies than the first valve strategy. The 2-stroke Miller cycle has been proposed as a means of transitioning from 4 to 2-stroke with progressive torque delivery and without requiring a throttled intake. The 2-stroke cycle appears to be a potential solution to overcome some of the difficulties faced by downsized gasoline engines during high load operations. In particular this cycle will be beneficial in a

condition where the engine is already operating at peak cylinder pressures and additional torque output is not easily achievable.

## 1. Introduction

Gasoline engine downsizing is a process whereby the engine speed/load operating points are shifted to a more efficient region through the reduction of engine capacity whilst maintaining the full load performance via pressure charging [1]. As specific output increases so too do the technical challenges, such as: a robust combustion system that allows a high compression ratio to maintain part load efficiency; good low speed torque and transient performance; real world fuel consumption benefits through a reduction in full load fuel enrichment; base engine robustness and durability [2]. High torque at low engine speed delivers good vehicle drivability. It is the desired target of modern downsized/downspeed engine design [3, 4]. The difficulty in maintaining high torque output at low engine speed is a result of low-speed pre-ignition (LSPI) [5, 6], which often leads to the undesirable detonation knock.

The introduction of variable valve timing (VVT), direct fuel injection and boosting system technologies in modern downsized engines has created an opportunity for conventional 4-stroke engine platforms to alter the operating cycle. One of the possible configurations is achieving the 2-stroke cycle. By running a 2-stroke cycle engine at the same crankshaft rotational speed and work production per unit of

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displaced volume as a 4-stroke cycle engine, the 2-stroke would in theory develop twice as much power as 4-stroke. Therefore, the 2-stroke concept offers the potential of producing the desired high torque at low engine speed without facing the danger of LSPI.

Historically, 2-stroke engines had a very different mechanical configuration from that of the 4-stroke counterparts. Conventional 2-stroke engines do not have poppet valves on the cylinder head, but use intake and exhaust ports on the side of the engine cylinder. The piston is always doing two separate actions. On its way to TDC, it is compressing the mixture for combustion while drawing fuel/air into the crankcase. On its way to BDC, it allows the combustion products to escape and lets in the mixture into the combustion chamber.

Ricardo has developed a prototype engine called the 2/4-SIGHT engine which can switch between 2 and 4-stroke operation [7]. It was demonstrated that the vehicle acceleration and performance figures could be maintained with a 2.1 litre V6 engine replacing a 3.5 litre baseline. An electro-hydraulic valve (EHV) actuation system was used for the prototype development rig. The boosting system was comprised of a 2-stage supercharger/turbocharger combination with intercooling. A 27% fuel economy benefit could be achieved over the NEDC drive cycle and a reduction in CO2 emissions from 260 g/km to 190 g/km.

The Miller cycle works by providing an expansion stroke which is longer than the compression stroke [8]. A practical method of achieving this with a variable valve train system is to provide an early or late intake valve closure. This effectively increases the ratio of the expansion stroke to that of the compression stroke by reducing the effective compression ratio. The effective compression ratio is reduced by limiting the amount of air let in during the intake process. The increased expansion ratio improves the engine thermal efficiency.

The aim of this work is to investigate the opportunity of adopting a 2-stroke cycle on a downsized 4-stroke poppet valve engine

in order to increase the power output; this will be achieved without increasing the max cylinder pressure limit. To further extend the 2-stroke cycle energy efficiency, a 2-stroke Miller cycle with extended expansion stroke is proposed in this work and studied using the modelling analysis developed. The 2 stroke Miller cycle could also potentially be used to transition between 4-2 stroke without having to use a throttle: Higher power is achieved by slowly increasing the effective compression and reducing the expansion until achieving balanced compression and expansion strokes.

## 2. Poppet-Valve 2-Stroke Engine Cycle

In order to improve the energy efficiency of 2-stroke concepts, a new operating cycle has been developed in this work, which also enables a poppet-valve engine to achieve 2-stroke operation. This new cycle has been invented to improve the thermal efficiency of 2-stroke by adopting the principles of the Miller cycle. The objective of this new cycle is to maintain the same expansion of the 4-stroke cycle, as a reduction of this will penalize the thermal efficiency; the scavenging and intake processes are thus shifted further into the compression stage by adopting high boost pressures.

Figures 1 and 2 illustrate the concepts by showing the compression and expansion of the gases for conventional 4-stroke, 2-stroke cycle and the extended 2-stroke Miller cycle. The theoretical 4-stroke cycle has a full 180 degree expansion and compression. The 2-stroke cycle shows that the geometric expansion and

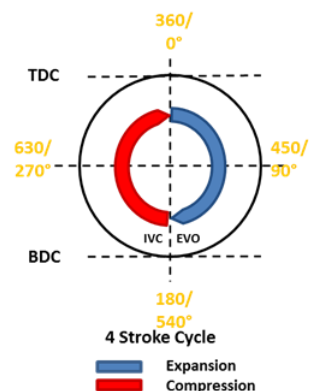


Figure 1: Theoretical 4-stroke cycle concept

compression strokes are both reduced by 50% in order to allow sufficient scavenging. The theoretical 2-stroke Miller cycle extends the expansion to a full 180 degrees, while significantly reducing the scavenging process. All these cycles can be materialised in a poppet valve engine platform with a flexible valvetrain system.

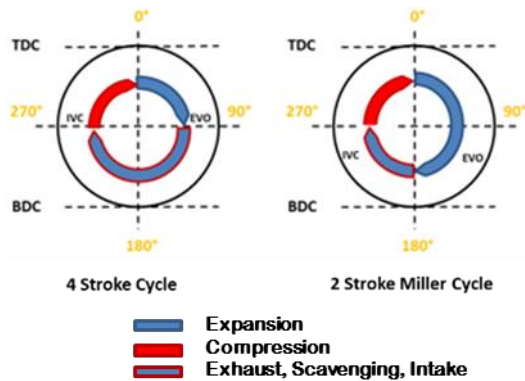


Figure 2: Theoretical 2-stroke and 2-stroke Miller cycle concepts

The valve timings can be modified in three dimensions: opening angle, duration and lift. In order to reduce the number of variables, it was decided not to modify the lift of the valves. To simplify the calibration of the 2-stroke Miller cycle, the EVO timing was kept the same as the four-stroke model at 150 degrees CA, as shown in Figure 3. The two cycles are therefore identical from the start of injection at -40 CA to EVO at 150 CA. The valve timings of the two stroke models have been identified through an engine

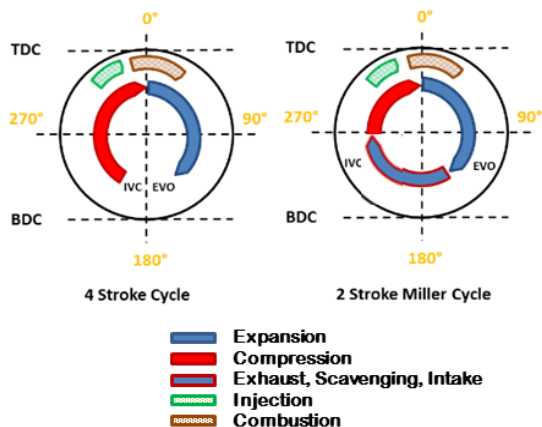


Figure 3: 4-stroke and 2-stroke Miller cycles

calibration and optimisation exercise which will be covered in section 4. Figure 4 beneath illustrates the 4-stroke and 2-stroke cycles in more detail. One of the most fundamental aspects for optimal operation of a 2-stroke poppet-valve engine is the scavenging process of the exhaust gas residuals. Scavenging efficiency is directly related to the thermal efficiency as it dictates the amounts of fresh charge that can be used at the start of each cycle. Good scavenging is also needed to reduce knocking risk as the hot residuals typically lead to auto-ignition. The 2-stroke Miller cycle is sensitive to auto-ignition due to the reduced scavenging time, which increases the amount of residual gases. High boost pressures will thus be required during scavenging to increase the scavenging efficiency; this will also ensure sufficient volumetric efficiencies as the intake occurs during compression.

The extended 2-stroke Miller cycle can achieve higher power by slowly increasing the effective compression and reducing the scavenging; when auto-ignition starts to occur, the expansion is reduced to allow for higher amounts of scavenging, until achieving balanced compression and expansion strokes. The extended 2-stroke Miller cycle can thus be used to transition between 2-4 stroke, it also allows torque control in two stroke without the need of a throttle.

### 3. Model Development

#### 3.1 Basic NA 4-stroke engine model

A naturally aspirated (NA) 4-stroke engine model is built based on the Lotus single cylinder optical research engine at Loughborough University. The engine is equipped with an active variable poppet-valve train system. The model was built using Ricardo Wave engine simulation software. The basic engine parameters are listed in the Table 1 below.

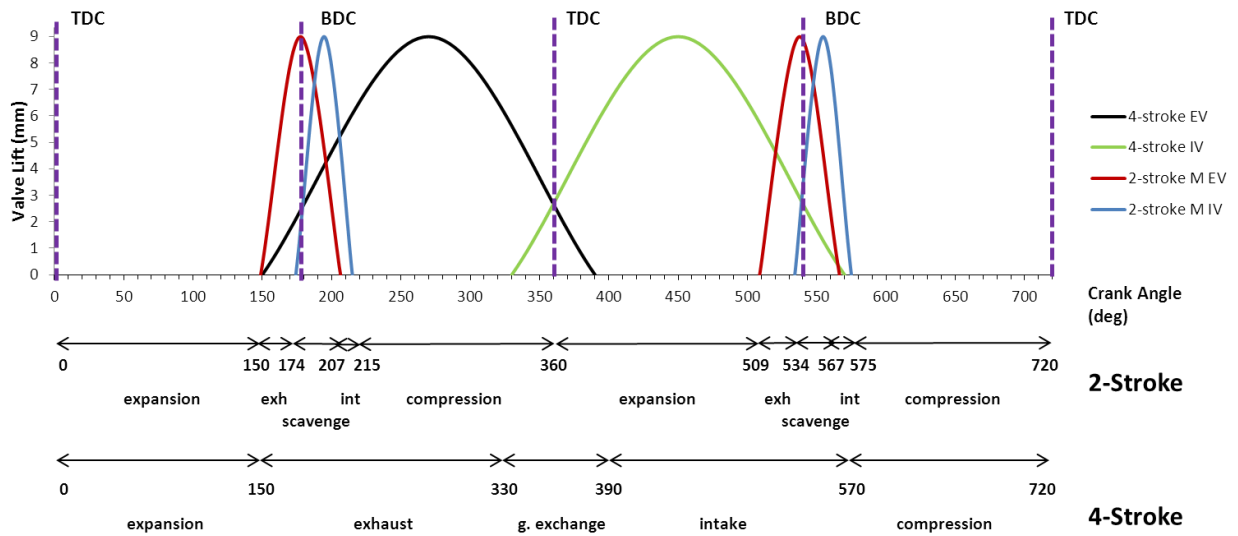


Figure 4 - Four and Two stroke Cycle Schematics

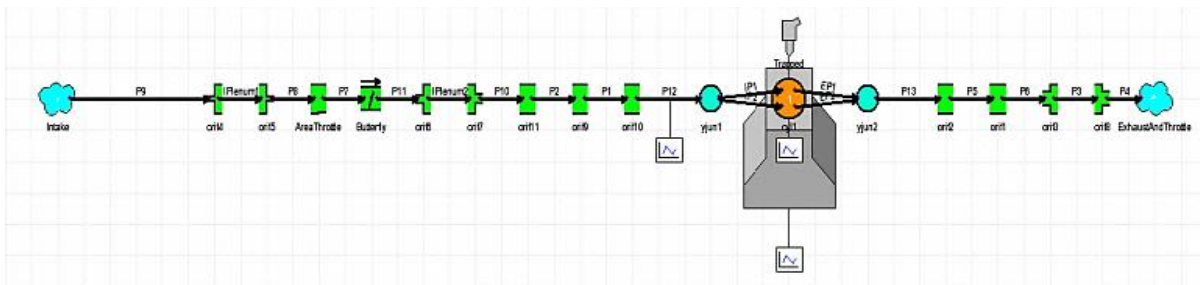


Figure 5: Single-Cylinder Engine Model in WAVE Software

This engine model is used as the building block for the other engine models, including boosted 4-stroke, 2-stroke and 2-stroke Miller cycles. Figure 5 illustrates the basic configuration of the engine; on either side of the system lies the ambient which was set to reference conditions of 300 K and 1 bar. The intake plenum, throttle valve, piping and every other duct leading into and out of the combustion chamber have been replicated from the Lotus research engine. For the direct fuel injection, the injector was modelled as

trapped A/F injector. This automatically adjusts the fuel mass flow to maintain a pre-specified air-fuel ratio. The start of injection (SOI), injection pressure, spray angle and nozzle diameter were matched to those adopted on the experimental engine.

The engine model does not have a spark plug; instead the combustion profile has to be pre-defined. Two main correlations used are the location of 50% burn and the combustion duration. These parameters

were adjusted to match the value and crank angle location of the maximum pressure point measured on the Lotus research engine.

Table 1 - Basic Engine Parameters

Displaced Volume	500 cc
Stroke	82.1 mm
Bore	88.0 mm
Connecting Rod	142.0 mm
Compression Ratio	10.1:1
Number of Valves	4

### 3.2 Model validation

Numerous iterations were performed until a satisfactory correlation was achieved. Table 2 shows the comparison between experimental and simulated values at 2000 rpm and 2.7 bar IMEP. The throttle opening was adjusted in the simulation in order to match the experimental IMEP value.

The percentage error values are within  $\pm 5.7\%$ . The ISFC value has a higher percentage error; this might be due to the estimation of the experimental fuel consumption from the fuel injection duration. The peak pressure also shows a very good correlation between simulation results and experimental measurements.

### 3.3 Boosted 4-stroke engine model

Having completed the 4-stroke NA model validation, a supercharger was added to the intake and connected to the crankcase through a compressor shaft. Poppet valve 2-stroke cycles require a heavily boosted intake to operate; the supercharged 4-stroke model is thus used as a baseline to evaluate the changes in performance when switching to 2-stroke or 2-stroke Miller cycles.

Table 2- Comparison of experimental vs. simulation results

Variable	Unit	Experimental	Simulation	Percentage Error (%)
Speed	Rpm	2000	2000	-
IMEP	Bar	2.74	2.79	1.82
GMEP	Bar	3.45	3.31	3.95
FMEP	Bar	1.18	1.14	3.53
P <sub>MAX</sub>	Bar	17.47	17.57	0.56
AP <sub>MAX</sub>	CA	16.7	17.29	3.56
Air mass	kg/hr	12.11	11.69	3.47
Fuel mass	kg/hr	0.83	0.76	9.4
Indicated power	Hp	3.1	3.12	2.24
ISFC	kg/kWh	0.37	0.32	11.39

A simple type of compressor was modelled. The supercharger does not use an adiabatic efficiency map, nor does it require a gearing ratio with the engine. Instead it uses a simple algorithm to calculate the work required to compress a given amount of air to a pre-defined pressure through an isentropic efficiency. This meant that a specific intake boost pressure could be achieved without extensive iterations and compressor matching exercises. The isentropic efficiency was set to a typical value of 70%. Although simple, the compressor model was deemed suitable for

researching the feasibility of operating in two-stroke at steady-state conditions. Boost pressures have been kept within the operating range of current production downsized engines, with a maximum achievable boost pressure of 1.5 bar. Additional boosting would have required coupling the compressor with a turbocharger. Previous experimental testing by Ricardo [9] has proven that a turbocharger/supercharger combination more than confidently generates the boost pressures required for successful 2-stroke operation in a 4-stroke poppet valve engine platform.

To simplify the comparison between all models, a maximum cylinder pressure limit has been set to  $61 \pm 1$  bar. The injection timings, durations and combustion profiles have been left constant for all models, as originally correlated with the Lotus engine. The AFR is also kept constant at 14.7:1. All the models will also be operating at WOT, as the main point of focus is maximum power output development.

Figure 5 (a-c) shows the performance increase achieved by introducing a supercharger to the naturally aspirated 4-stroke model. The boost pressure was increased until the  $61 \pm 1$  bar maximum cylinder pressure limit was reached. At 6000 rpm, there is 4 kW brake power increment achieved by the supercharged cycle. The volumetric efficiency has also increased from an average of 90% to 107%.

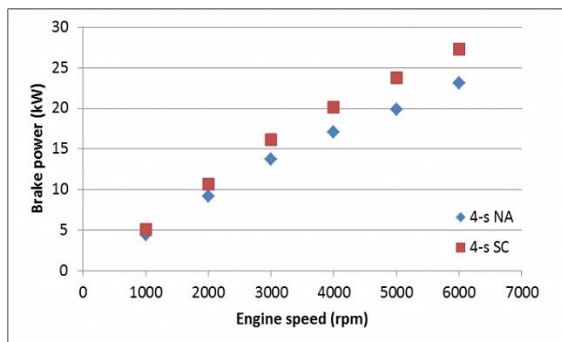
Ramping up the boost pressures would further increase the power output but also lead to excessive cylinder pressure. To further increase boost pressures and thus power output the ignition timing must be delayed. A delayed ignition shifts the location of the maximum heat release to a point further into the expansion of the piston. This strategy lowers the maximum cylinder pressure which in turn enables higher boost pressures to be adopted. However, the delayed combustion penalizes the thermal efficiency of the engine. The scope of the following investigation consists in demonstrating that additional power can be developed



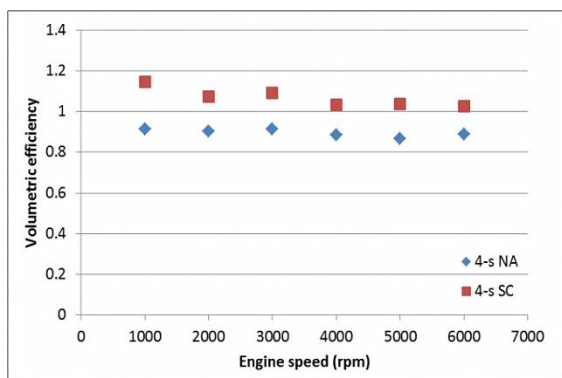
without exceeding the pressure limits and without delaying the spark timing simply by running the engine in two-stroke mode.

#### 4. Developing the 2-Stroke Cycles

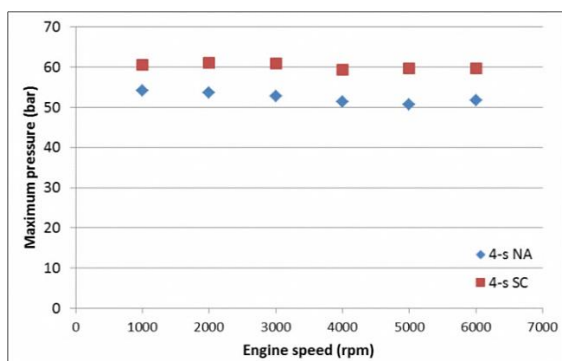
The 2-stroke models were constructed by instructing the software to initiate new injection and combustion events every 360 degrees CA instead of 720. An initial valve profile commonly referenced in the literature for a poppet valve 2-stroke cycle was used as a starting point and also repeated every 360 degrees CA.



(a)



(b)



(c)

Figure 6: Naturally-aspirated vs. supercharged engine performance

Extensive DoE, statistical modelling and optimization exercises were carried out to determine the most appropriate valve timing, opening duration and boost pressure characteristics at each engine speed for maximum power output. Once again the same cylinder pressure limits were applied. The injections and combustion profiles have been left unaltered. The EVO timing for the 2-stroke Miller cycle was kept fixed at 150 degrees CA, as with the 4-stroke cycles.

The 2-stroke Miller cycle has been proposed as a means of operating at part load in 2-stroke without requiring a throttle; this cycle aims to maximise the thermal efficiency by balancing effective compression, expansion and scavenging. To simplify the comparison with the other models it was decided to optimize the 2-stroke Miller cycle for maximum torque output.

Having defined the valve timings and profiles for both 2-stroke models, an experimental test was performed on the Lotus Optical Engine to ensure that they could be replicated at high engine speeds. To protect the engine the injectors were disabled and it was turned by a motor; the valve profiles were replicated successfully throughout the entire engine speed range without interfering with the piston motion.

#### 5. Two-Stroke Simulation Results

Figure 7 shows the PV-diagrams produced by the 2-stroke and Miller 2-stroke cycles. It can be seen that the expansion line for the 2-stroke Miller carries on for longer due to the delayed exhaust valve opening. This is the main reason behind the increased thermal efficiency. However it can also be seen that the compression line is higher; this is due to the high boost pressures required to scavenge the air. The work done by the cycle is therefore reduced by the high pressure scavenging process. This reduction of the PV-diagram volume is effectively an added pumping loss to the engine.

Figure 8 shows the calculated engine power at varying engine speed of both 4-

stroke NA and boosted cycles and 2-stroke standard and Miller cycles. It can be seen that both 2-stroke cycles investigated deliver higher power than their 4-stroke counterparts. It can also be observed that above 3000 rpm the power output of the Miller cycle falls rapidly. This effect is a result of low scavenging efficiency, as the supercharger struggles to generate sufficient intake pressures. This effect suggests that the 2-stroke Miller cycle is limited to low-mid engine speeds unless additional charging methods are adopted, such as a turbocharger/supercharger combination.

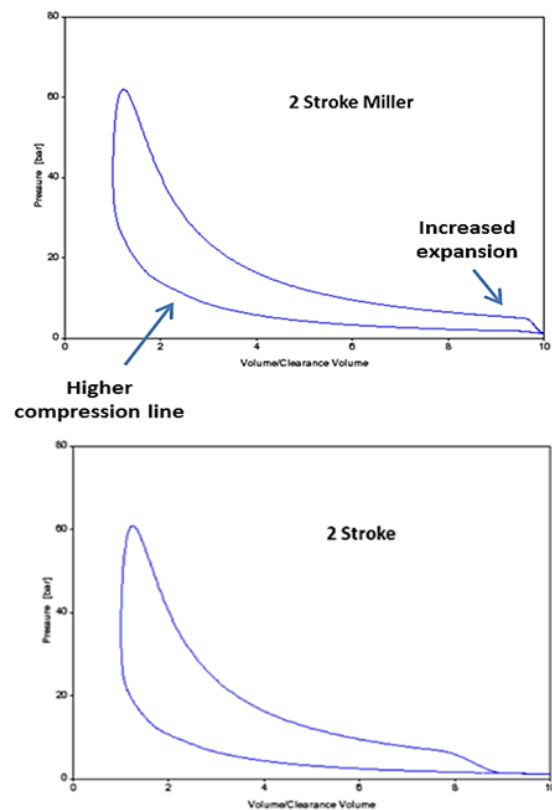


Figure 7: PV-diagrams of 2-stroke cycles

The higher boost pressure requirement of the 2-stroke Miller cycle due to a reduced scavenging time was previously anticipated during the development process. Valve and intake geometry optimisations could be of great benefit towards improving the air delivery and reducing the boost pressure requirements.

Previous studies [10] highlighted that the scavenging flow in a poppet valve engine

has a tendency to go straight from the inlet valve to the exhaust valve without clearing the cylinder; this effect is referred to as short-circuiting airflow. The addition of a valve shroud and of a deflector in the intake proved successful in improving the scavenging efficiency by generating a reverse tumble motion. Figure 9 illustrates the reverse tumble effect in a poppet valve engine.

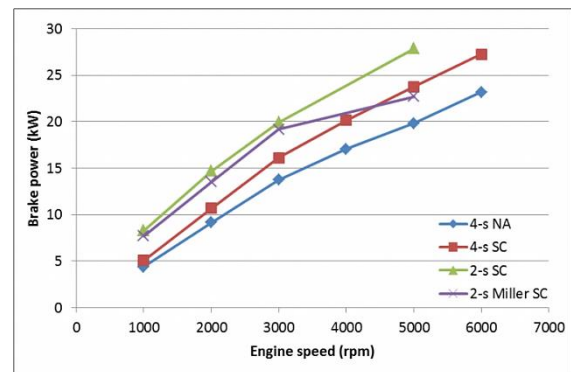


Figure 8: Brake power of 4-stroke and 2-stroke cycles

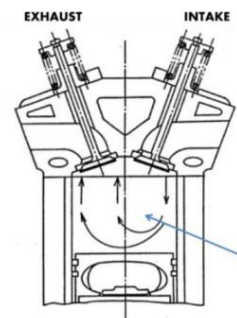


Figure 9: Reverse Tumble Motion [11]

Figure 10 shows the calculated brake torque at varying engine speed of 4-stroke and 2-stroke cycles. It can be seen that the 2-stroke cycles generate a substantial amount of additional torque at low speed, with an increase as high as 63%. At high engine speeds the torque generated falls rapidly; this effect is primarily due to the reduced volumetric efficiency and poor scavenging as a result of the reduced time available each cycle. This result reinforces the conclusion that a poppet valve 2-stroke cycle is limited to low engine speeds.

Figure 10 illustrates the brake thermal efficiencies of each cycle. As anticipated the 2-stroke Miller cycle has a higher thermal efficiency than the standard two-stroke model due to the increased expansion. However, both 2-stroke cycles investigated deliver lower thermal efficiency than their 4-stroke counterparts.

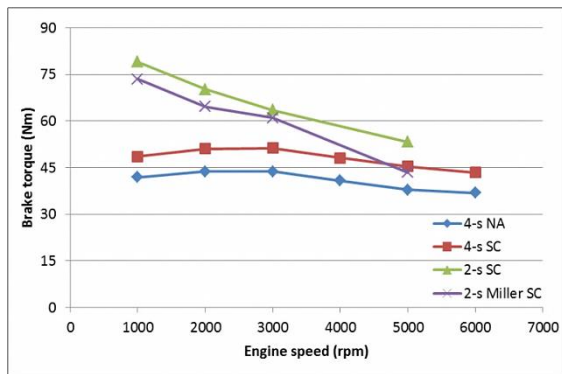


Figure 10: Brake torque of 4-stroke and 2-stroke cycles

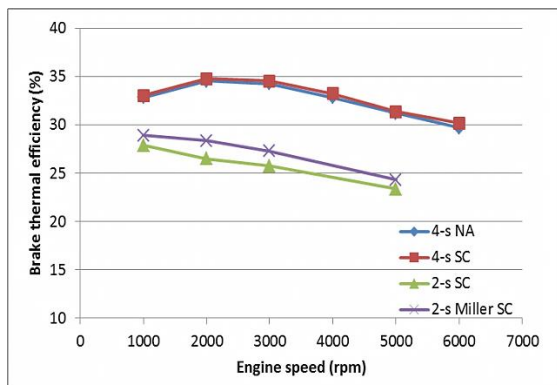


Figure 11: Brake thermal efficiency of 4-stroke and 2-stroke cycles

Although the 2-stroke Miller cycle adopts the same expansion of the 4-stroke models, it has additional losses associated to the high boost pressures required by the supercharger. The intake process occurs during compression and thus works against the piston motion. The results shown for the 2-stroke Miller cycle were achieved after optimizing for maximum torque production; higher thermal efficiencies can be achieved however, by lowering the boost pressures, reducing the effective compression ratio

and increasing scavenging, at the expense of power output.

The standard 2-stroke cycle also suffers from a reduced expansion as well as higher pumping losses associated to boosting.

The thermal efficiency of the supercharged 4-stroke model would be significantly lower if sufficient spark delay had been adopted to increase torque production without exceeding the cylinder pressure limits.

A relatively small shift of the thermal efficiency from 4-stroke to 4-stroke supercharged cycles can be observed. This is a result of the very small boost pressures adopted by the cycle to comply with the cylinder pressure limits. Figure 12 shows the boost pressures of all supercharged models, relative to 1 bar ambient pressure.

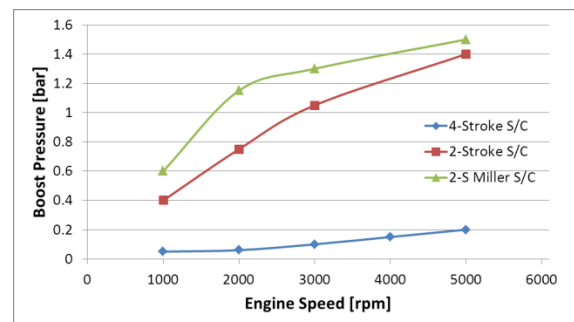


Figure 12: Boost Pressures of Supercharged Cycles

Figure 13 shows the calculated residual gas fractions. The residual gas fraction represents the combustion residuals which have not been removed by the scavenging process. It can be seen that the 2-stroke Miller cycle has the highest amount of residual gases, due to the decreased efficiency of the scavenging process. The standard 2-stroke cycle has obtained significantly better scavenging at low engine speeds but struggles to rid the residuals at high speeds. This high amount of residuals limits the amount of fresh charge and is ultimately responsible for the loss of torque with speed of the 2-stroke models. In addition, the high amount of residual gas increases the



tendency to achieve LSPI [12]. Variable geometry intake, port shielding and air tumble will likely be required to ensure no auto-ignition would occur for the Miller cycle. A high gas residual fraction also reduces the fuelling quantities due to diminished oxygen presence.

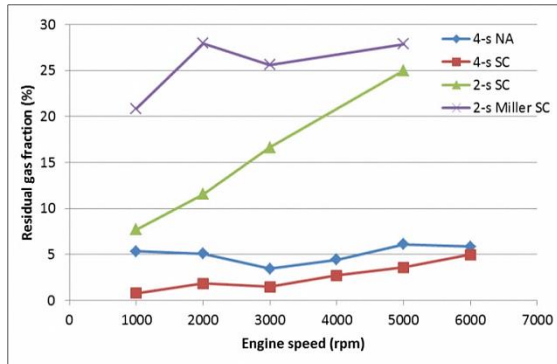


Figure 13: Residual gas fraction of 4-stroke and 2-stroke cycles

The exhaust gas temperatures of all cycles are plotted in Figure 14. It can be noticed that the 2-stroke cycles have significantly lower exhaust gas temperatures; this is due to the excess air which flows through the exhaust in an attempt to scavenge the cylinder. This effect suggests that even when high power output is demanded from the vehicle, 2-stroke operation would generate lower exhaust temperatures than the 4-stroke cycle at full load; therefore there is a potential for after-treatment systems in the exhaust to retain higher efficiencies since most catalysts are affected by lower conversion efficiencies and precious metal oxidation at very high temperatures.

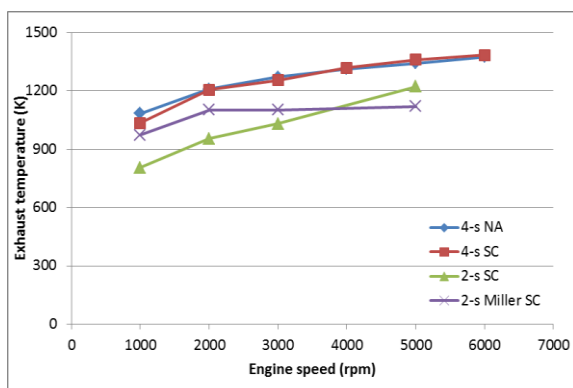


Figure 14: Exhaust gas temperature of 4-stroke and 2-stroke cycles

## 6. Conclusions

(a) A new engine cycle hereby called 2-stroke Miller has been proposed and simulated in an effort to improve the fuel economy of the 2-stroke cycle. This new cycle has successfully generated more power than the 4-stroke supercharged baseline whilst remaining within the maximum cylinder pressure limits and without delaying the ignition. The new 2-stroke Miller cycle managed to achieve higher thermal efficiencies and thus lower fuel consumption figures than the standard 2-stroke cycle.

(b) The 2-stroke Miller cycle has also been developed to enable a 4-stroke engine to gradually switch to 2-stroke mode and generate a progressive torque increase. The EVO timing has been kept to 150 degrees CA, as with the four stroke cycles. Torque increase is achieved by increasing the scavenging time and advancing EVO until reaching balanced compression and expansion strokes of the two stroke cycle. Torque reduction is achieved by reducing the effective compression ratio and maintaining a full expansion for thermal efficiency. This strategy enables torque modulation without the additional pumping losses of a throttled intake.

(c) The modelling study showed that a poppet-valve 4-stroke engine with variable valve timing, direct injection and boosted intake can achieve 2-stroke operations for additional power development. A further brake torque output of 63% has been achieved by the two stroke cycle at 1000 rpm whilst maintaining the cylinder pressure limits.

(d) Both 2-stroke models have lower exhaust temperatures compared to the 4-stroke cycles; this reduction of the exhaust temperatures has been associated to the higher mass flows from the scavenging process. Lower exhaust temperatures offer an opportunity to reduce catalyst ageing and improve efficiencies during high load operations.

## 7. Discussion and further work

(a) The effectiveness of two stroke operation in a four stroke engine is mainly limited by the scavenging efficiency, which is greatly affected by the time available. It is likely that these operating strategies would be more successful in a compression ignition engine because of the reduced sensitivity to auto-ignition; the longer piston stroke would also allow more time and space for the process. Further development of this valve strategy should consider Diesengine applications.

(b) A Computational Fluid Dynamics study was not carried out to optimize reverse tumble motion in the cylinder and scavenging efficiencies. Active intake deflectors and valve shrouds should be introduced to characterize the improvements of scavenging efficiency and fully understand the feasibility of the 2-stroke Miller cycle.

(c) The 2-stroke Miller cycle was only optimized for maximum power output and not fully characterized throughout the speed/load range. Improved fuel consumption figures and lower exhaust residuals can be achieved by reducing the effective compression ratio. A more complete analysis is required to fully characterize the potential of this cycle.

(d) A transient analysis of the 2-stroke Miller cycle is required to explore the feasibility of adopting it to switch between 4 and 2-stroke and modulating the torque without the use of a throttle.

(e) An additional DoE and optimization exercise which considers a complete set of engine variables and valve profile configurations is required.

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