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Original Research Article

A Comparison of Variable Valve Strategies at Part Load for Throttled and Un-Throttled SI Engine Configurations

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Abstract

The presented work concerns the study of the fuel consumption and emissions benefits achieved at part load by employing a fully variable valve train in a 1.6L SI gasoline engine. The benefits achieved when using variable valve timing alone, and combined with an early intake closing strategy for un-throttled operation were explored in order to highlight the merits of throttle versus un-throttled engine operation in conjunction with variable valve timing and lift. In addition, particular interest was given to the presence of internal Exhaust Gas Recirculation (EGR) and its ability to reduce pumping loss at part load. An engine model employing multiple sub models to handle variable valve operation was constructed using a commercial gas dynamics engine code, allowing detailed analysis of three valve strategies. Using the engine model, a theoretical study validated by experimentally available data was carried out to study key valve timing cases. A detailed breakdown of the mechanisms present in each case allowed a comprehensive understanding of the influence of valve timing on gas exchange efficiency and fuel consumption.

Keywords: Variable Valve strategy, part load, un-throttled SI engine, EGR

Nomenclature

ATDC	After Top Dead Centre
BDC	Bottom Dead Centre
BTDC	Before Top Dead Centre
CA	Crankshaft Angle
CFD	Computational Fluid Dynamics
CI	Compression Ignition
EGR	Exhaust Gas Recirculation
EIVC	Early Intake Valve Closing
EVC	Exhaust Valve Closure
EVO	Exhaust Valve Opening
GMEP	Gross Mean Effective Pressure
IMEP	Indicated Mean Effective Pressure
ISFC	Indicated Specific Fuel Consumption
IVC	Intake Valve Closing
IVO	Intake Valve Opening
MOP	Maximum Opening Point
P_{max}	Maximum in-cylinder pressure
PMEP	Pumping Mean Effective Pressure
PV	Pressure-Volume (Diagram)
SI	Spark Ignition
TDC	Top Dead Centre
UTM	Universiti Teknologi Malaysia
VVA	Variable Valve Actuation
VVT	Variable Valve Timing
WOT	Wide-Open Throttle

1. Introduction

The work in this study relates to the efficiency improvement of gasoline engines for the automotive industry. Increasing concern for the impact fossil fuel combustion emissions are having on our environment is the greatest driver for the application of new fuel-saving technologies. With compression ignition engines approaching their conventional (non-variable, non-hybridised) efficiency limit, further focus is required to increase the efficiency of gasoline engines as has been the case in recent years.

Internal combustion engines that burn fossil fuels are still the favoured power plant for road transport, where the vast majority of Europe’s new cars remain powered by gasoline or diesel motors. Gasoline cars account for 44% of all new registrations, diesel cars for 55%, with all other technologies including hybrid and electric making up the remaining 1% (Mock, 2012).

Unfortunately, the combustion of fossil fuels produces many harmful emissions. The largest contributor to climate change is CO_2 emissions, which is proportional to fuel consumption. Legislations and protocols have been the greatest driver for manufacturers to look into technologies that improve fuel economy.

Compression ignition engines are reaching their efficiency limit mainly due to their throttle-less operation. However gasoline spark ignition engines that typically require a throttle valve in order to remain close to stoichiometric combustion at varying engine loads, require additional technologies to counteract their relative, inherent inefficiency due to pumping losses. Throttling at part load is the largest contributor to the differences in efficiency between CI and SI engines. At the full load condition the pressure difference over the throttle valve is very small, reducing the pumping losses to a minimum. At part load the pumping losses are far greater, reducing the total efficiency of the cycle.

Conventional internal combustion engines control intake and exhaust valves by means of a fixed geometry cam driven system. The cam shafts rotate relative to crank angular velocity, allowing constant valve event timing for all engine conditions. However the optimum valve event timing differs considerably at varied engine conditions. Low valve lift and durations benefit driveability at low speeds, but at high speed it acts as a flow restriction sacrificing maximum performance. Conversely, high lift and duration benefits high speed operation, but reduce volumetric efficiency at low speed.

Passenger car engines operate most frequently at part load during low speeds. Unfortunately this is the range at which SI engines are least efficient. The primary cause for this inefficiency is the requirement for a throttle plate to control engine load inherent in spark ignition engines. In general, the efficiency increases proportional to engine load. Various strategies have been attempted to increase engine load at lower engine speeds to increase the efficiency.

Other technologies that target pumping losses are currently being explored by the industry.

Variable valve actuation provides improvements in engine performance, efficiency and emissions by optimising the event timing of the valves as a function of engine speed and load (Sellnau and Rask, 2003). The key mechanisms that can be varied in valve operation are:

- Intake valve opening (IVO) and closing (IVC) timing
- Exhaust valve opening (EVO) and closing (EVC) timing
- As a function of the above, valve duration
- Valve lift

Fully variable valve actuation can effectively replace the need for a throttle. Valve parameters such as lift and duration control the volume of inlet charge the cylinder can trap and combust. The valves take on the role of controlling the air fuel ratio at part load, therefore the throttle valve can be left wide open or deleted completely (Kitabatake et al, 2011). The technology is expensive; however it can be used in synergy with other technologies successfully due to its flexibility. There are many systems currently in production by most manufactures, and they vary in cost, complexity and performance.

2. Methodology

The base engine was modelled on a 1.6L in-line cylinder 16 valve gasoline engine based on an experimental engine from the Universiti Teknologi Malaysia and (Kuruppu et al, 2014). Combustion data was gathered from a similar displacement test engine (Cairns et al, 2009). Modification of the engine model was carried out during the course of this investigation in order to simulate the Variable Valve Actuation (VVA) capability. Throughout the study, conclusions drawn from published papers have been used to validate model results.

Combustion duration and timing is very sensitive to changes to valve strategy and in reality would be physically measured on an engine test bed. However, due to the complexity of combustion modelling, the engine model requires these values as inputs. Single cylinder experimental engine data attained during a study of un-throttled operation was available at Brunel University London and has been used as a benchmark throughout this study.

The cylinder bore of the single cylinder test engine is marginally larger, projecting to a 1.9L total displacement in a 4 cylinder configuration. Engine specifications are included in Table 1. The engine speed and load investigated in this study has been largely governed by the availability of existing comparable data.

The flow chart depicted in Figure 1 represents an overview of the order of tasks carried out during the study.

Table 1. Engine configuration comparison

	Single Cylinder test engine	WAVE Engine model
No of cylinders	1	4
configuration	single (NA)	in-line (NA)
Bore (mm)	82.5	76
Stroke (mm)	88.9	88
Cylinder displacement (cc)	475.2	399.2
Engine displacement(litres)	0.475L (projected 1.9L 4cyl)	1.6L
Geometric compression ratio	9.8:1	9.8:1
Variable Valve Timing	Fully Variable (Inlet & Exhaust)	Fully variable (inlet & exhaust)
Fuel injection	port injection	port injection
Fuel	95 RON unleaded gasoline	Indolene

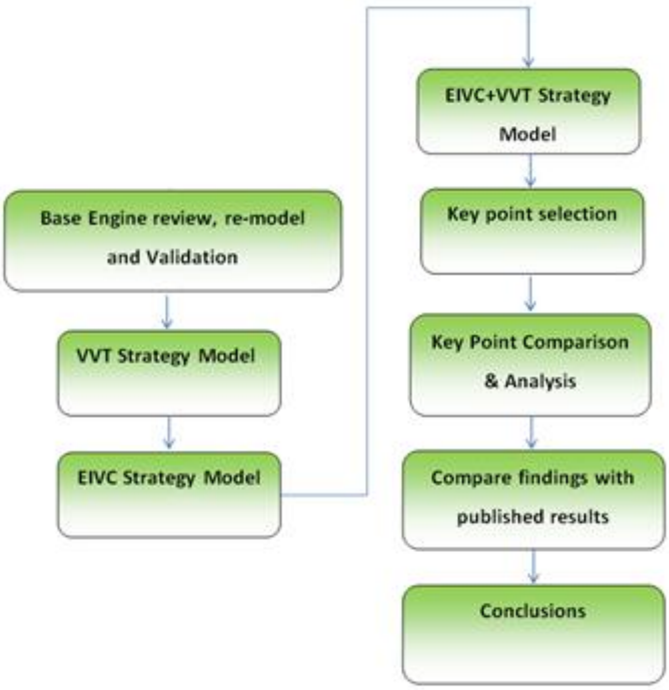

Figure 1. Methodology flowchart

3. Results for Throttled Engine

This section contains a review of the mechanisms that affect the cycle performance. Pressure volume (PV) plots are a useful metric to analyse and compare the cycle performance of the key points across each strategy. P-V graphs are plotted on logarithmically-scaled axes to provide a reasonable and readily understandable profile of the pumping loop. The key performance metrics studied were the gross mean effective pressure (GMEP) which is the work delivered to the piston over the compression and expansion strokes only while the pumping mean effective pressure (PMEP) represents the work transferred to the cylinder gasses during the exhaust and induction strokes only. The ratio of GMEP to PMEP, is also known as 'gas exchange efficiency' – a measure of engine efficiency, related to fuel consumption. Finally, the indicated specific fuel consumption (ISFC) was used, a parameter representing the rate at which fuel is consumed over the engine indicated power production cycle (before losses).

3.1 Baseline engine

The plot in Figure 2 shows the pressure-volume diagram for cylinder 1 at 2000rpm (6bar IMEP) during throttled operation in red. The second data set on the plot shows the full load performance at 2000rpm in blue as a comparison. Wide open throttle (WOT) minimises flow restriction, allowing cylinder pressures during the inlet stroke to remain at ambient pressure (1bar). Effort required to induct fresh charge is at a minimum. The pressure actually reaches above atmospheric at points during the stroke as a result of high gas momentum and well-timed pressure waves, achieving a volumetric efficiency greater than 1.

With identical valve timing, the cylinder pressure during the exhaust stroke remains similar between cycles, however slight pressure irregularities during the stroke can be identified as the result of pressure waves induced by the exhaust valve opening. Cycle similarities end at the beginning of the induction stroke. At part load, the flow restriction at the throttle reduces the intake pressure to below atmospheric, causing an increase in negative work. A lower mass of air trapped in cylinder reduces cylinder pressure throughout the compression stroke. As a result, maximum cylinder pressure is lower, as is the work done on the piston during the expansion stroke. During full load, the in-cylinder pressure is considerably greater at the start of the compression stroke (BDC), yielding a pressure increase through to TDC. The increase translates to a higher maximum cylinder pressure during combustion, and consequently an increase in work done during the expansion stroke. The relatively large ratio between gross torque and pumping torque compared to the part load cycle enables higher gas exchange efficiency and IMEP to be achieved.

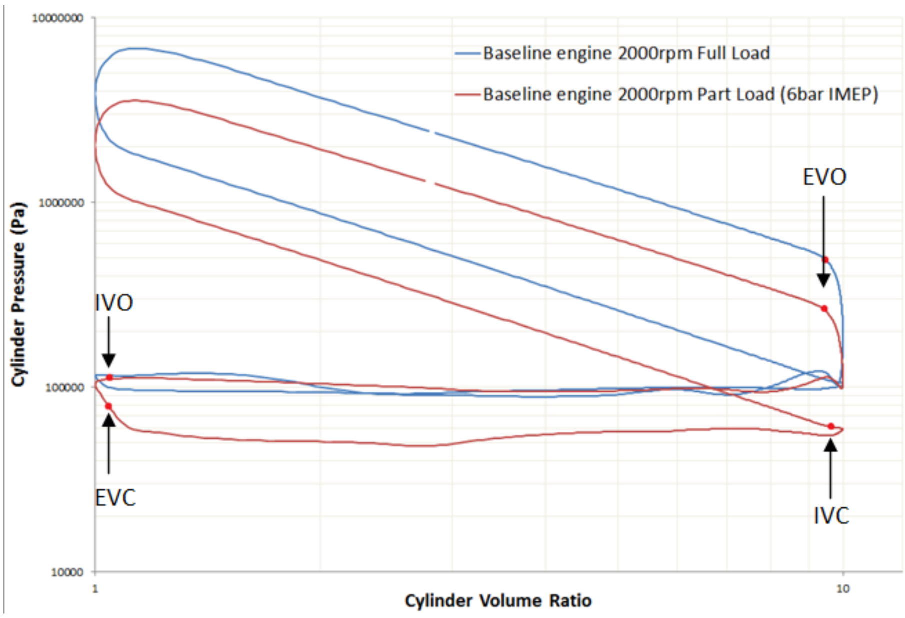

Figure 2. Part load vs. full load

The in-cylinder pressure plot Figure 3, shows both the combustion P_{max} (in blue) and the motored curve (red dashed) where no combustion takes place. As described in section **Hata! Başvuru kaynağı bulunamadı.** previously, the optimum P_{max} angle is between 10-15 degrees CA in order to maximise work and reduce heat transfer **Hata! Başvuru kaynağı bulunamadı.**. The engine simulation at 6bar IMEP yields a 12.5 degree angle of P_{max}.

3.2 Throttled Key Point – Case A

The pressure-volume plot depicted in Figure 4 shows a comparison of the base engine and key point 'A', where approximate valve event timings are annotated. It is clear that the pumping loop area has been reduced from the base engine, and the power loop area appears marginally less in the VVT engine.

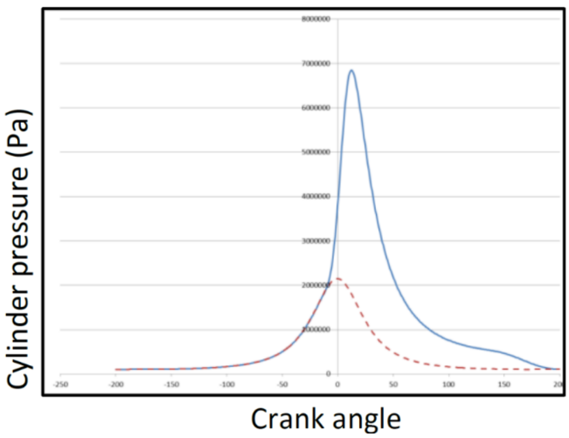

Figure 3. Cylinder pressure plot

Consequently the ratio between PMEP and GMEP, known as gas exchange efficiency, has increased by 7% (**Hata! Başvuru kaynağı bulunamadı.**).

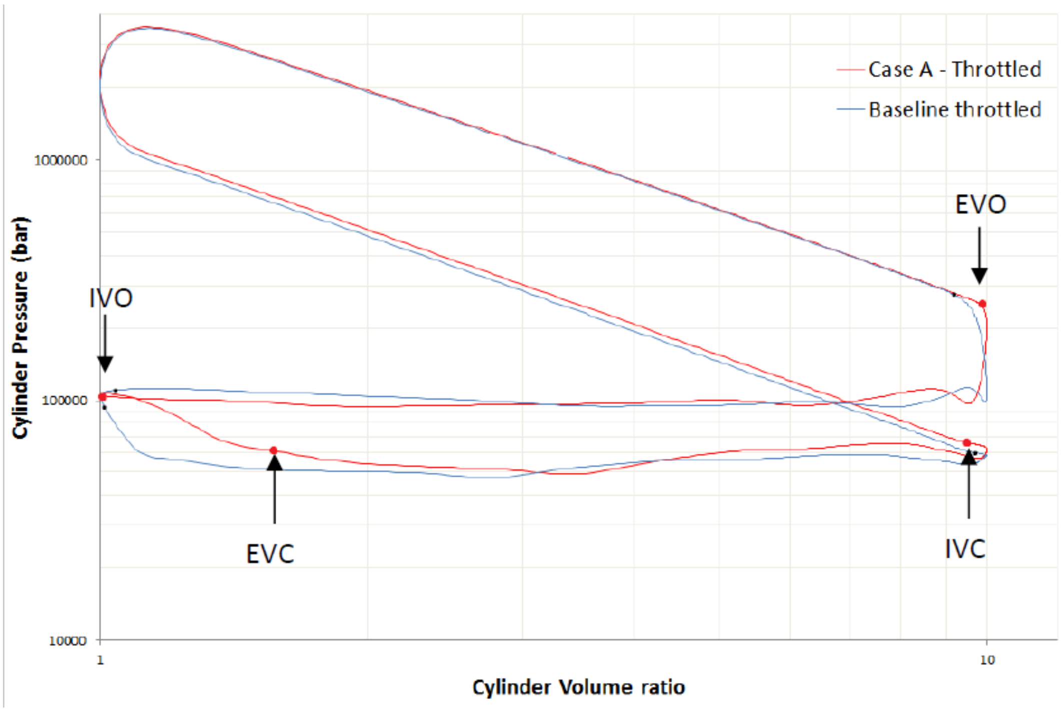

Figure 4. Throttled VVT vs. baseline engine P-V diagram

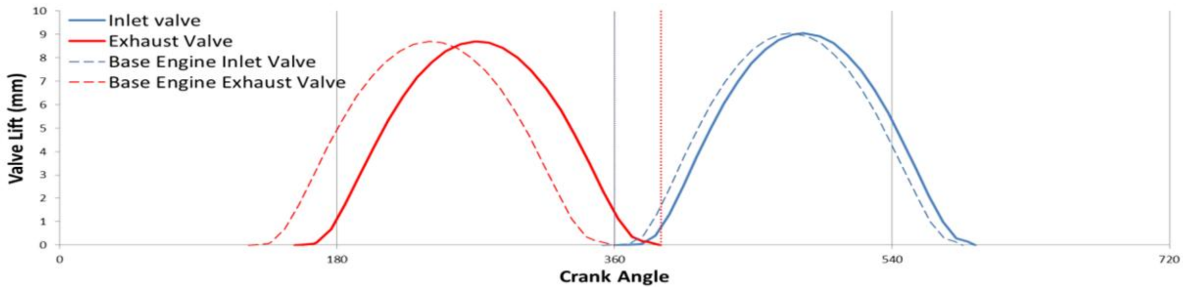

Figure 5. Case A throttled Valve lift

The exhaust valve phasing has retarded both the opening and closing by approximately 22 degrees compared to the standard engine. The retarded exhaust valve closing event has delayed the drop in cylinder pressure just before BDC relative to baseline.

As a result, the expansion ratio has increased providing an increase in work.

The pressure difference across the exhaust valve at the moment of valve opening creates a strong pressure wave. Upon reflection the positive pulse momentarily raises the cylinder pressure as seen just after BDC at the beginning of the expansion stroke. This phenomenon occurs in both cases, albeit slightly delayed in case-A due to the exhaust valve closing later. The delay in exhaust valve maximum opening point (MOP) in case-A enables a steady 1bar pressure throughout the exhaust stroke to be maintained. The reduction in effective valve area and flow co-efficient is apparent in the base engine as the flow begins to choke halfway through the stroke, raising the cylinder pressure and increasing the pumping work.

The first few degrees of the induction phase where the intake valve area is low, gases from atmospheric pressure in the exhaust fill the cylinder providing an increase in pressure over baseline, thus reducing PMEP. However, as the intake valve opens further, the negative pressure gradient across the intake valve pulls exhaust gas through the cylinder and into the intake. This phenomenon can be understood clearly in terms of mass flow rate as depicted on the following page. Positive exhaust mass flow depicted by Figure 6 describes gasses travelling into the cylinder, whereas negative intake mass flow describes mass flow travelling into the intake. Gasses flow in the reverse direction for the entire duration of valve overlap.

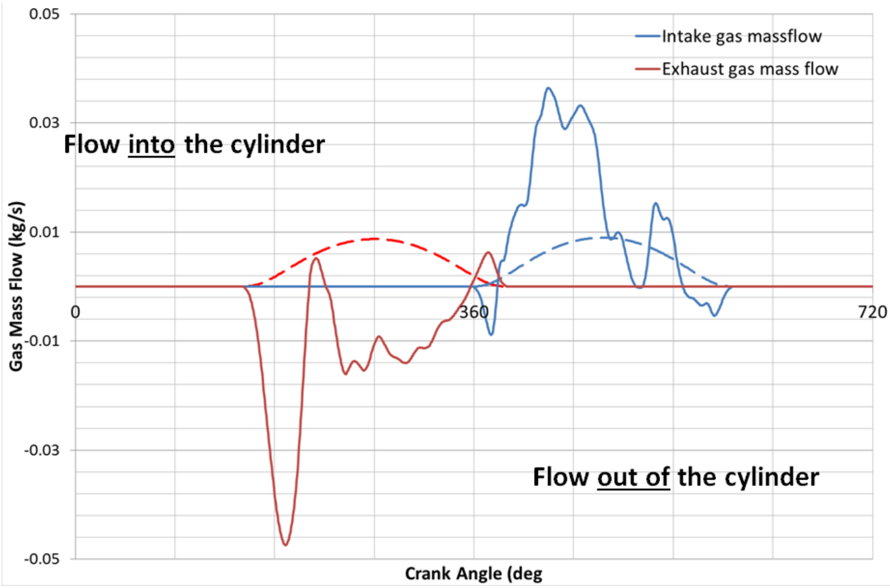

Figure 6. Cylinder mass flow

Table 2. Case A tabulated summary of results

Metric	Unit	Throttled Baseline	Throttled Case A	% Difference to baseline
ISFC	kg/kW/hr	0.223526	0.221674	-0.83
GMEP	bar	6.46079	6.38916	-1.11
PMEP	bar	-0.43949	-0.406902	-7.41
GMEP/PMEP	-	14.67	15.73	7.27
CO	ppm	4922.77	4680.13	-4.93
HC	ppm	62.6794	60.8027	-2.99
NOx	ppm	4715.59	4336.06	-8.05
Residual gas f	%	6.80336	9.45301	38.95
EVO_deg	deg	-130.631	-152	-
EVC_deg	deg	8.631	30	-
IVO_deg	deg	-7.059	0	-
IVC_deg	deg	226.941	234	-

The low pressure in the intake port due to throttling provides a negative pressure difference across the valve encouraging backflow. Compounded by high valve overlap, exhaust gasses travel from a region of ambient pressure to low pressure, filling the intake with residual gasses. The more the intake valve opens, the more the pressure in the cylinder starts to equalise that of the intake, reducing cylinder pressure further. The delay in IVC relative to the base engine means higher cylinder pressure at BDC, however a larger fraction of exhaust gas residuals inhibits the expansion work. Compression is negative work, and if no gain is achieved by the expansion stroke, the overall result is a reduced power loop.

3.3 Throttled Key Point – Case B

Case B has marginally less valve overlap than base engine (10deg from 16deg base engine) however both valves have been simultaneously retarded. Approximate valve timing event annotations have been plotted onto Figure 7. The strategy employs late valve overlap, promoting high internal EGR. The further away from TDC the valve overlap occurs, the greater the effect the piston motion has on the airflow (Mechadyne, 2006).

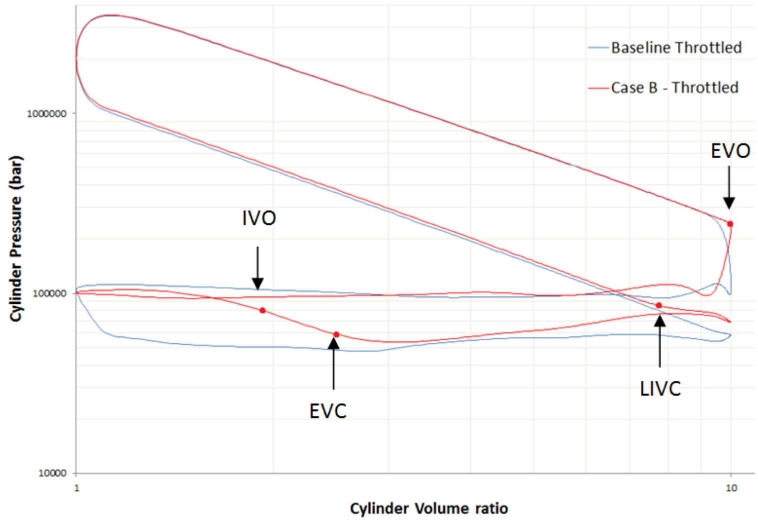

Figure 7. Case B PV diagram

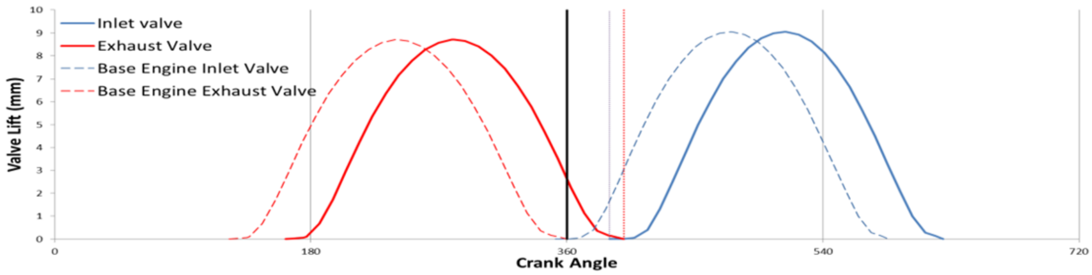

Figure 8. Case B valve lift

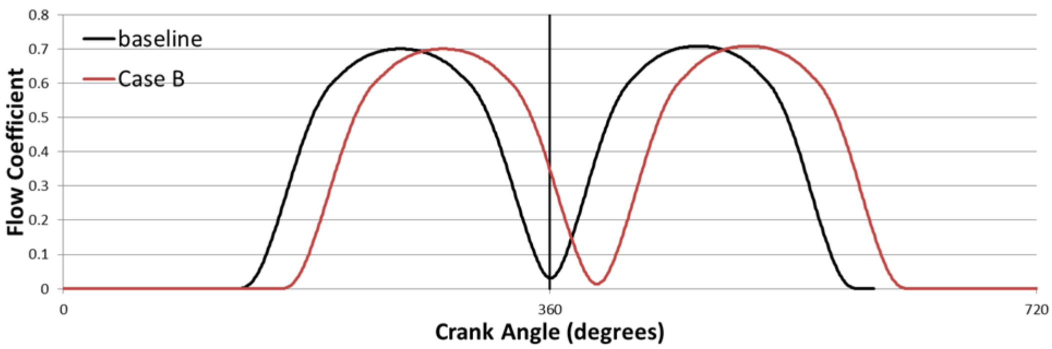

Figure 9. Combined flow coefficient

Further benefit has been gained in the expansion ratio over the previous case ‘A’ due to the exhaust valve opening 10 degrees later, a total of 31 degrees later than baseline. As a result of further exhaust gas expansion, the pressure difference over the exhaust valve is lower, decreasing the rate of pressure drop before the start of the exhaust stroke. The crank angle duration which the cylinder pressure takes to equalise to the exhaust back pressure has a detrimental effect on GMEP.

In addition, the pressure wave induced by the valve opening is lower in magnitude but longer in duration, therefore increasing the pumping work needed to displace the exhaust gases. Later in the stroke the cylinder pressure remains at a constant 1bar level - a good indication of a high flow co-efficient exhaust valve operation during the stroke.

The initial increase in pressure during the induction stroke is unusual, but appears to be the result of the reflection of another positive pressure wave in the exhaust port, initiated by the piston arriving at top dead centre. The magnitude and timing of this return pulse is influenced heavily by the exhaust length and exhaust gas temperature which changes the speed in which the wave travels. The exhaust length is tuned to increase cylinder scavenging at the beginning of the induction phase and is optimised for a particular speed and load range. To gain full benefit using this strategy, both inlet and exhaust manifold lengths would ideally have to be re-optimised.

The cause of the low cylinder pressure during the intake stroke is due to the low valve overlap and late intake valve opening. The combined flow coefficient of both inlet and exhaust valve is lower than both standard and case-A as shown in Figure 9**.** The pressure then raises proportional to the inlet valve opening towards BDC.

At BDC the intake valve is still open, causing backflow into the low pressure intake port thus reducing the effective compression ratio. The initial gain in cylinder pressure before BDC is completely offset by the late closing, reducing the compression ratio to similar magnitude as both the standard and case-A.

The cycle benefits from a 2.5% reduction in fuel consumption over the baseline engine, where the majority of the improvement was seen in the reduction of PMEP. A summary of the cycle results are included in Table 2.

Table 2. Case B tabulated summary of results

Metric	Unit	Throttled Baseline	Throttled Case B	% Difference to baseline
ISFC	kg/kW/hr	0.223526	0.217901	-2.52
GMEP	bar	6.46079	6.35934	-1.57
PMEP	bar	-0.43949	-0.333607	-24.09
GMEP/PMEP	-	14.67	19.13	30.45
CO	ppm	4922.77	4522.5	-8.13
HC	ppm	62.6794	60.1825	-3.98
NOx	ppm	4715.59	3996.24	-15.25
Residual gas f	%	6.80336	10.3815	52.59
EVO_deg	deg	-130.631	-162	-
EVC_deg	deg	8.631	40	-
IVO_deg	deg	-7.059	30	-
IVC_deg	deg	226.941	264	-

3.4 Throttled Key Point – Case C

This strategy has the highest valve overlap of all the key points, with late EVC and early IVO, 40 degrees of overlap is achieved, with the highest combined valve lift occurring at 10 degrees aTDC. The P-V diagram in Figure 10 highlights some interesting anomalies, adding up to a considerably different cycle to the standard base engine while Figure 11 provides the corresponding valve profile.

The inlet valve opens 20deg BTDC where a large pressure difference between the exhaust and inlet causes exhaust gasses to flow backwards into the inlet port. This initial mass flow of exhaust gasses supplements the mass in the cylinder, delaying the reduction in in-cylinder pressure. As soon as the exhaust valve closes, cylinder pressure drops below that of the intake manifold and flow changes direction once more. The impact of pressure wave oscillations on the gas mass flow can clearly be seen in Figure 12.

To reiterate, the convention for Figure 12 is as follows;

- Negative exhaust mass flow denotes the flow out of the cylinder
- Positive intake mass flow denotes flow into the cylinder
- The opposite in both cases defines backflow

A second pressure wave in the intake causes a momentary back flow, after which the flow changes direction again, reducing the cylinder pressure. A final smaller pressure wave increases the cylinder mass as the inlet closes just after BDC. The three points at which back flow occurs in the intake have been highlighted on both Figure 10 and Figure 12, where the back flow increases the pressure in the cylinder. Pressure oscillations have heavily influenced this particular cycle, initiated by the occurrence of the back flow at IVO. The frequency of the oscillations suggests the intake manifold effective length is too short. The sum result has reduced the pumping work to the lowest of all throttled cases.

The increase in cylinder pressure moments before BDC has increased the magnitude of negative work during the compression stroke. However, due to the relatively high gas residuals present, only a marginal increase in expansion work has been achieved.

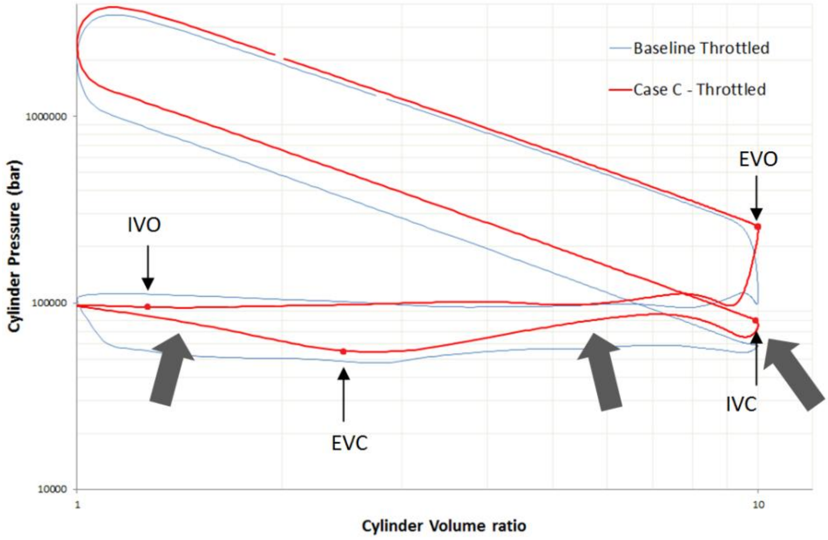

Figure 10. Case C pressure volume diagram

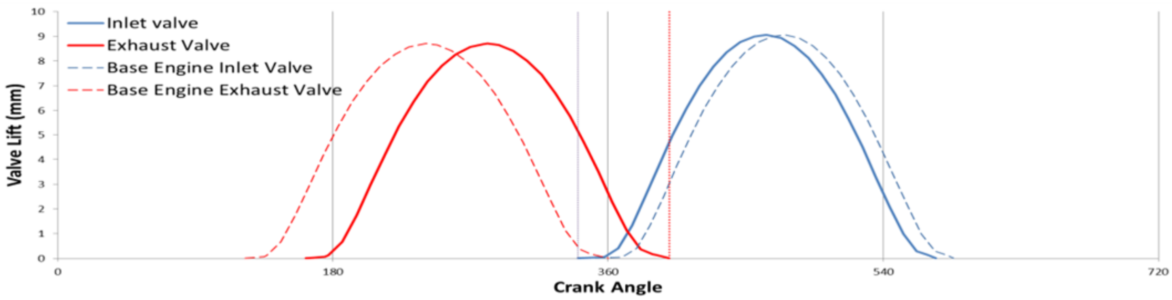

Figure 11. Case C valve lift

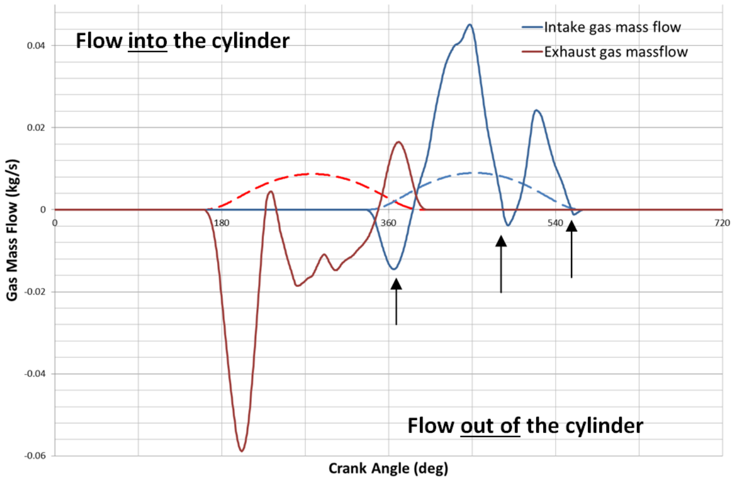

Figure 12. Case C mass flow

To summarize, pressure waves present in both the intake and exhaust are clear drivers for the increase in EGR. The tabulated results from the cycle are included in Table 3.

Table 3. Case C tabulated summary of results

Metric	Unit	Throttled Baseline	Throttled Case C	% Difference to baseline
ISFC	kg/kW/hr	0.223526	0.221894	-0.73
GMEP	bar	6.46079	6.20753	-3.92
PMEP	bar	-0.43949	-0.311634	-29.09
GMEP/PMEP	-	14.67	19.88	35.52
CO	ppm	4922.77	4061.4	-17.50
HC	ppm	62.6794	53.4252	-14.76
NOx	ppm	4715.59	2815.27	-40.30
Residual gas f	%	6.80336	20.1663	196.42
EVO_deg	deg	-130.631	-162	-
EVC_deg	deg	8.631	40	-
IVO_deg	deg	-7.059	-20	-
IVC_deg	deg	226.941	214	-

4.1 Base un-throttled engine

By observing the P-V diagram (Figure 13), it may be seen that the early intake closing strategy has clearly reduced the pumping loop area (PMEP) considerably over the baseline throttled strategy. However it also appears that the sum area of the power loop is reduced. What is important is the ratio between the two which denotes gas transfer efficiency. The mechanisms that affect the performance over the cycle are further investigated.

The intake valve opening time (Figure 14) is also shared between strategies, however beyond TDC the performance varies considerably. The throttled engine experiences a large pressure difference across the valve at part load, inherent with throttled engine operation. The effect is lowered overall intake pressure thus sapping engine torque in order to induct the air into the cylinder.

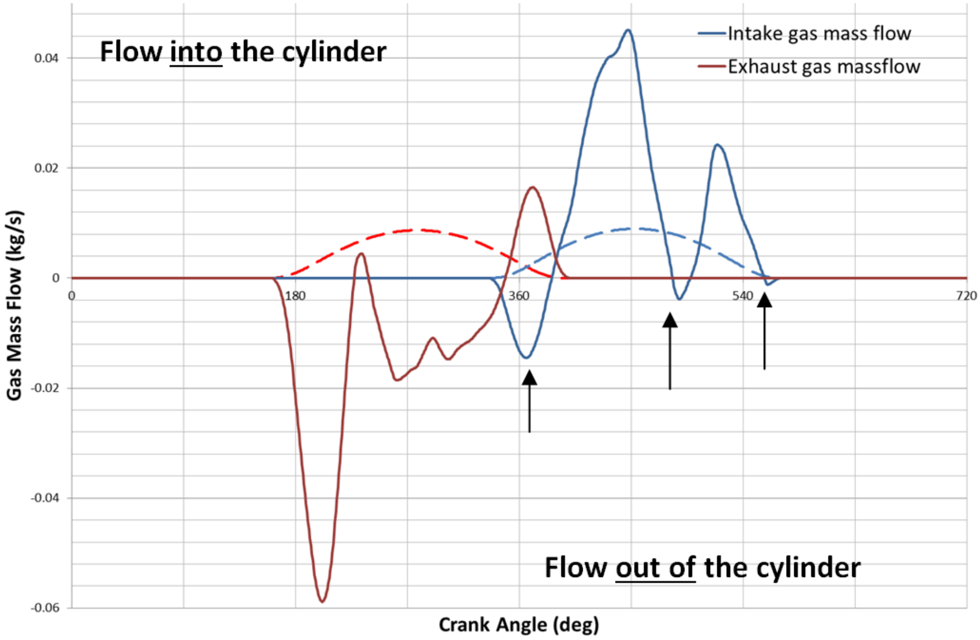

Figure 13. PV diagram

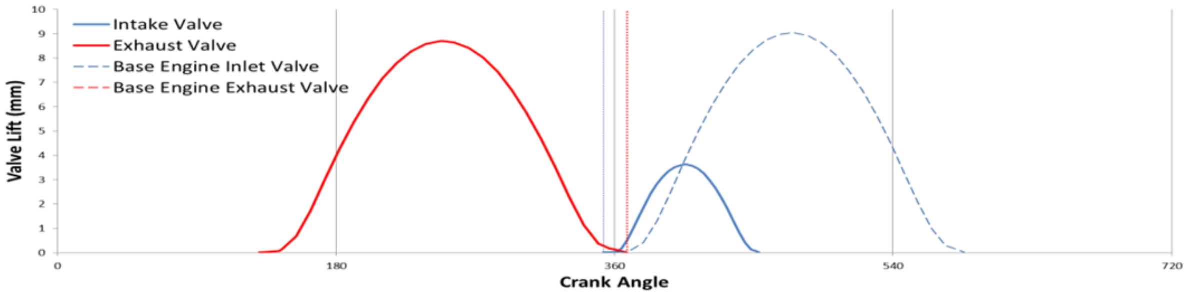

Figure 14. Baseline Un-throttled valve lift

The throttle is deleted in the EIVC strategy, removing the drop in pressure, therefore raising the static intake air pressure to a relatively constant atmospheric pressure level throughout the intake phase. Increasing the static intake pressure completely changes the dynamics of flow in the cylinder, in which case the pressure difference over the exhaust and intake is considerably lower.

As a consequence of early intake valve closing, a reduced mass of mixture is trapped and expanded, reducing the cylinder pressure to below that of the throttled cycle at BDC. However the work lost to expanding the mass is regained during compression as the trapped mass acts as a spring. Furthermore, the charge is not compressed until piston reaches the equivalent crank angle that the inlet valve closed at. The loss of effective compression ratio is apparent in the lower cylinder pressure seen throughout the compression stroke. A lower P_{max} is achieved, reducing also the expansion stroke.

PMEP reduced over 50% compared to the throttled strategy, combined with only a 3% decrease in GMEP, the total gas exchange efficiency increased 100% at this key point. A significant BSFC saving is achieved also, with less air and fuel mass being required to produce 6bar IMEP in order to overcome the lower negative pumping torque. Overall results are presented in Table 4 below.

Table 4 - Baseline tabulated summary of results

Metric	Unit	Throttled Baseline	Un-throttled Baseline	% Difference to baseline
ISFC	kg/kW/hr	0.223526	0.213212	-4.61
GMEP	bar	6.46079	6.25473	-3.19
PMEP	bar	-0.43949	-0.202671	-53.88
GMEP/PMEP	-	14.67	29.31	99.86
CO	ppm	4922.77	4692.92	-4.67
HC	ppm	62.6794	62.4894	-0.30
NOx	ppm	4715.59	3830.26	-18.77
Residual gas f	%	6.80336	6.78066	-0.33
EVO_deg	deg	-130.631	-130.6	-
EVC_deg	deg	8.631	8.6	-
IVO_deg	deg	-7.059	-7.078	-
IVC_deg	deg	226.941	97.8	-

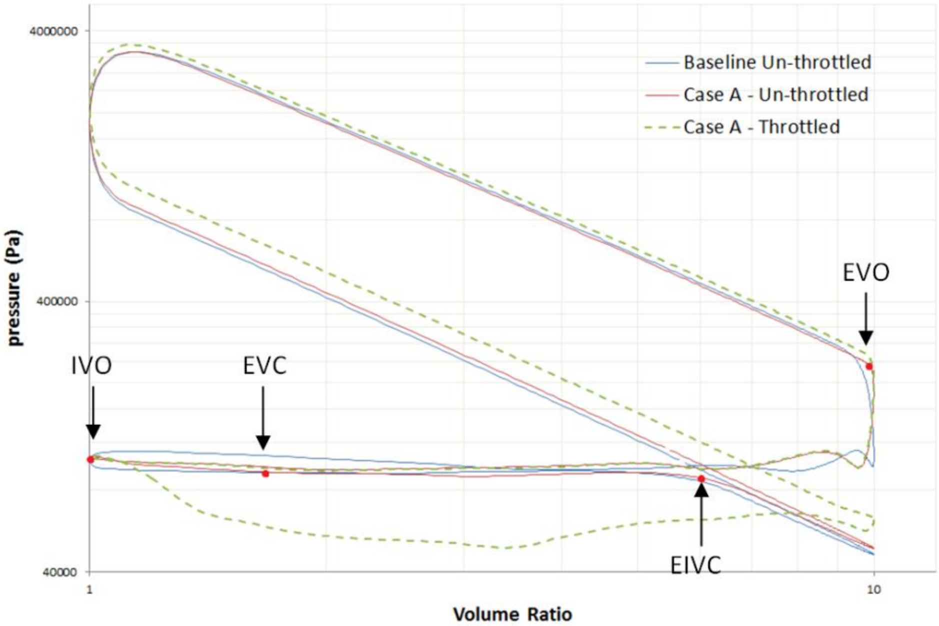

Figure 15. Case A un-throttled P-V diagram

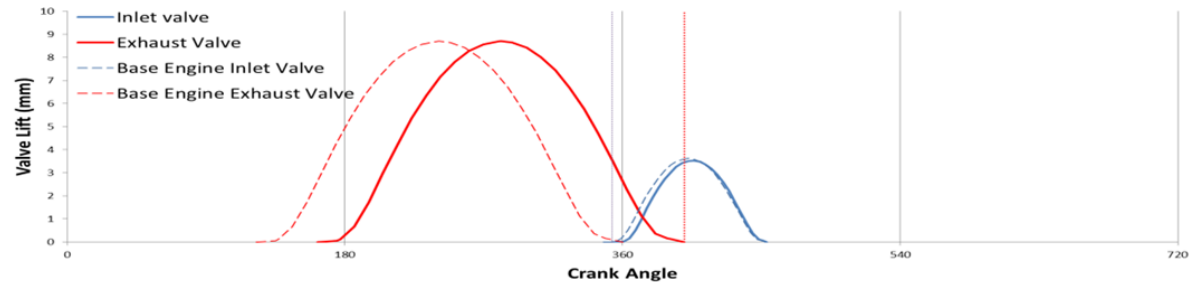

Figure 16. Case A un-throttled valve lift

4.2 Un-throttled Key Point – Case A

As with all the throttled cases, the inlet valve duration is 234 degrees. The un-throttled case A maintains the constant engine load by decreasing both the valve duration and lift to

100.6 degrees and 3.4 mm respectively (Figures 15 and 16). A result of the un-throttled strategy and the requirement to maintain constant IVO timing is that IVC occurs significantly earlier at 100.6 decrees ATDC as stated in Table 5.

Table 5. Case A tabulated summary of results

Metric	Unit	Un-throttled Baseline	Throttled Case A	Un-throttled Case A	% Difference to Unthrottled baseline
ISFC	kg/kW/hr	0.213212	0.221674	0.212271	-0.44
GMEP	bar	6.25473	6.38916	6.19051	-1.03
PMEP	bar	-0.202671	-0.406902	-0.19933	-1.65
GMEP/PMEP	-	29.31	15.73	29.40	0.30
CO	ppm	4692.92	4680.13	4530.84	-3.45
HC	ppm	62.4894	60.8027	61.2868	-1.92
NOx	ppm	3830.26	4336.06	3702.35	-3.34
Residual gas fraction	%	6.78066	9.45301	9.26886	36.70
EVO_deg	deg	-130.6	-152	-152	-
EVC_deg	deg	8.6	30	30	-
IVO_deg	deg	-7.078	0	0	-
IVC_deg	deg	97.8	234	100.603	-

The initial phase of the intake stroke from case-A benefits from a higher exhaust valve flow coefficient (Figure 17) compared to the base un-throttled engine due to the retarded EVC. The higher static pressure of the intake manifold compared to throttled operation minimises back flow to the intake after TDC. This phenomenon is seen in Figure 15 where the cylinder pressure remains close to ambient pressure even after IVO.

The combined flow coefficient of both the intake and exhaust valves from the base strategy exceeds that of case-A for a portion of the stroke, allowing a higher cylinder pressure to be achieved. However the valve area soon exceeds the baseline engine area once again due to the IVC occurring 3 degrees later. As a result, a greater mass of air is trapped allowing higher cylinder pressures to be attained.

The higher compression ratio is not converted into a higher expansion ratio due to the presence of residuals (Table 5). Trapped residuals of inert pre-combusted gas displace fresh charge and lower combustion temperatures. A considerable NO_x benefit becomes available as a result; conversely however, a lower P_{max} is achieved thus reducing the expansion work.

Marginally lower PMEP is achieved over the un-throttled baseline due to the retarded exhaust closing. The overall gas exchange efficiency improvement was negligible. No considerable change in BSFC was achieved.

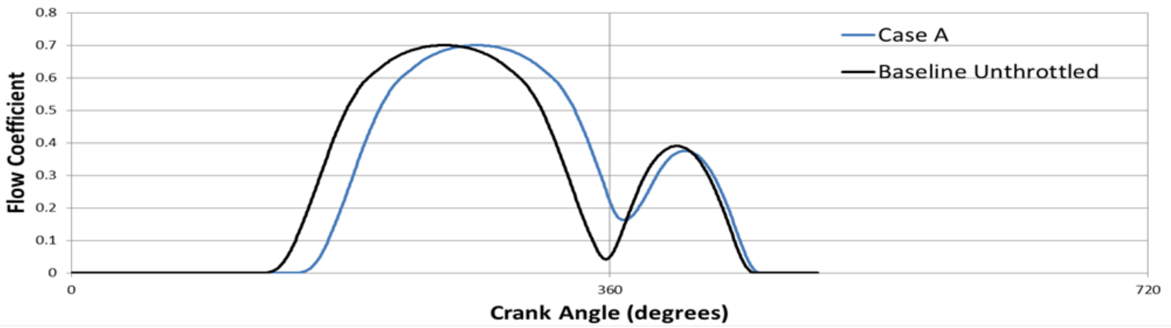

Figure 17. Case A un-throttled valve flow coefficient

4.3 Un-throttled Key Point – Case B

Case B employs a late, low overlap strategy that has a large exhaust valve area initially beyond TDC encouraging internal EGR. The pumping loop follows a similar trace to that of the throttled case until the point of IVO (Figure 18). The cylinder pressure in the throttled case continues to decline due to the relatively low pressures seen in the intake. However, the cause of the drop in cylinder pressure in the un-throttled case is due to the combined flow coefficient of both the intake and exhaust valves reducing to below that of the base un-throttled cycle (Figure 20).

Throttling of the flow occurs at the inlet valve where 2.6mm of lift is achieved (Figure 19). The choked flow reduces the pressure in-cylinder considerably over the baseline un-throttled cycle. As the inlet valve eventually opens fully, the mass flow increases into the cylinder raising the pressure. Further retarded IVC over baseline yields an increase in trapped mass and therefore higher pressures are achieved during the compression stroke. Increased inert exhaust residuals lower combustion temperature, thus reducing the magnitude of P_{max} and importantly the expansion work. The result is higher PMEP and a lower GMEP, leading to a lower overall gas exchange efficiency and greater fuel consumption - an overall relative decrease in performance.

Table 6 shows the relative performance compared to the throttled baseline engine and the equivalent throttled case. VVT has provided no benefit to the cycle compared to the base un-throttled engine.

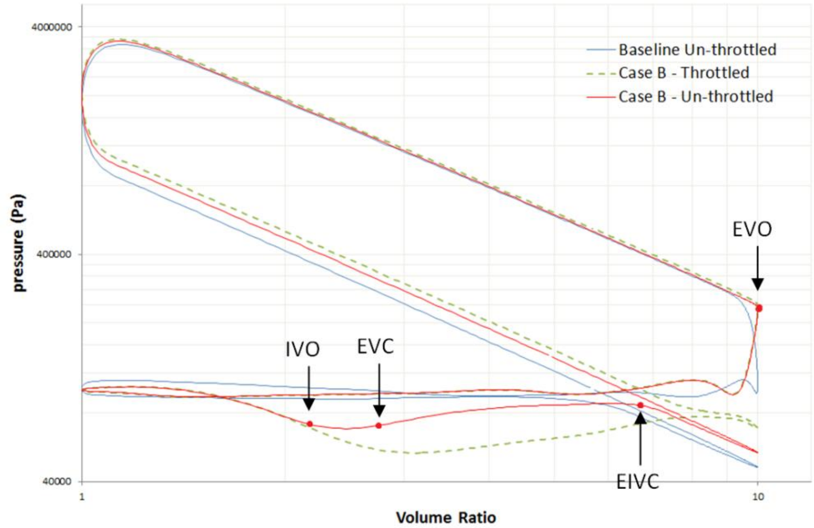

Figure 18. Case B un-throttled P-V diagram

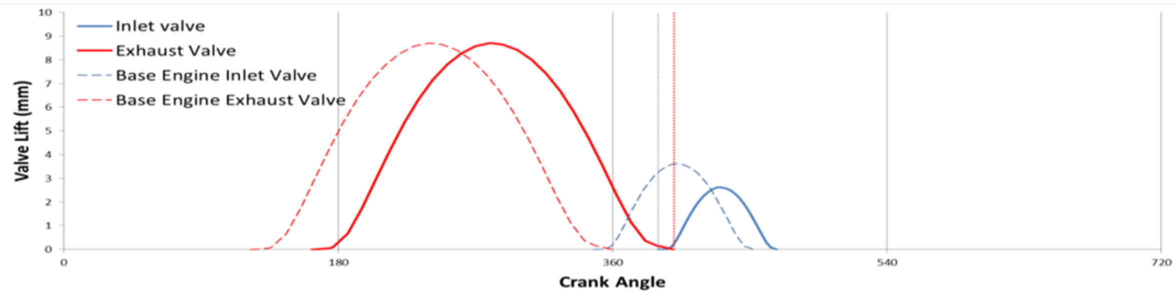

Figure 19. Case B un-throttled valve profile

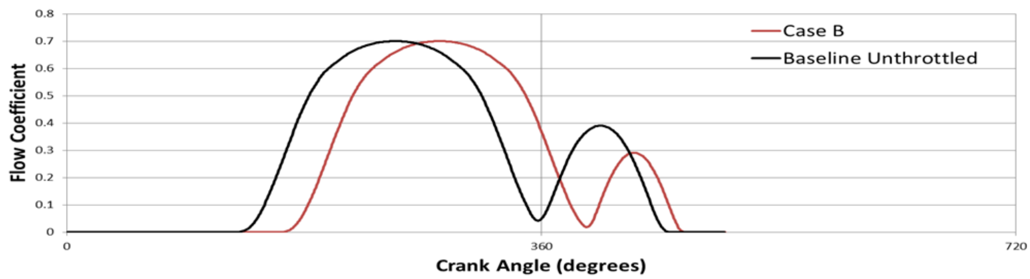

Figure 20. Case B un-throttled combined valve flow co-efficient

Table 6. Case B un-throttled tabulated summary of results

Metric	Unit	Un-throttled Baseline	Throttled Case B	Un-throttled Case B	% Difference to Unthrottled baseline
ISFC	kg/kW/hr	0.213212	0.217901	0.214994	0.84
GMEP	bar	6.25473	6.35934	6.16833	-1.38
PMEP	bar	-0.202671	-0.333607	-0.248016	22.37
GMEP/PMEP	-	29.31	19.13	23.25	-20.68
CO	ppm	4692.92	4522.5	4448.46	-5.21
HC	ppm	62.4894	60.1825	59.9749	-4.02
NOx	ppm	3830.26	3996.24	3622.56	-5.42
Residual gas fraction	%	6.78066	10.3815	10.4819	54.59
EVO_deg	deg	-130.6	-162	-162	-
EVC_deg	deg	8.6	40	40	-
IVO_deg	deg	-7.078	30	30	-
IVC_deg	deg	97.8	264	106.843	-

4.4 Unthrottled Key Point – Case C

The expansion stroke benefits from an improved expansion ratio due to the retarded exhaust valve opening event. As discussed in previous cases, the improvement is outweighed by the increase in cylinder pressure at the beginning of the exhaust stroke (Figures 21 and 22). The cylinder pressure during both the latter phase of exhaust and the initial phase of intake remains very similar to the base engine, with little variation present.

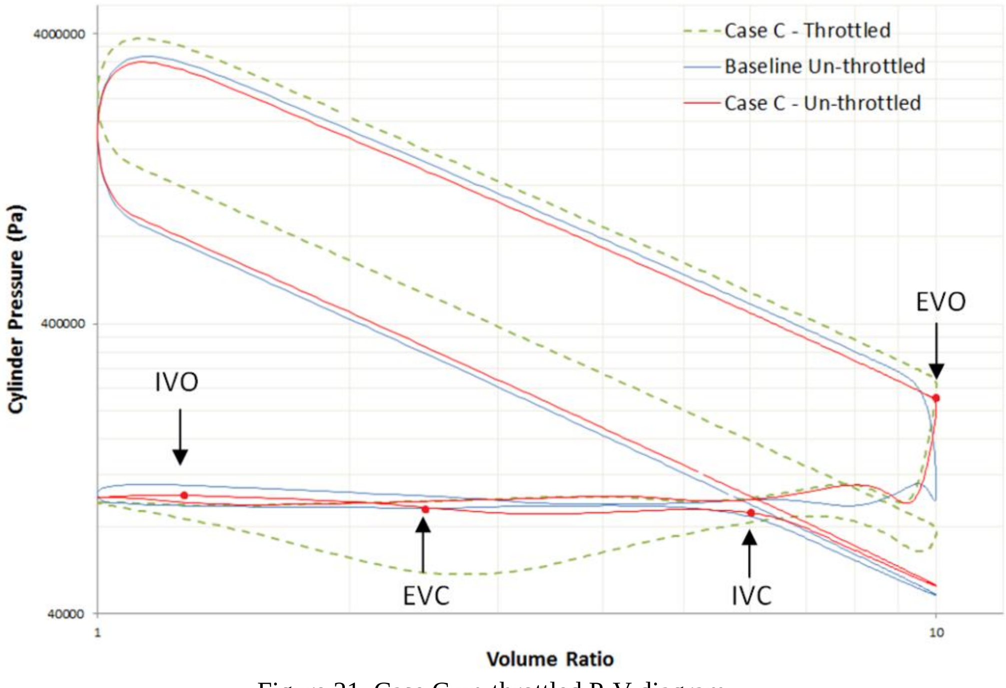

Figure 21. Case C un-throttled P-V diagram

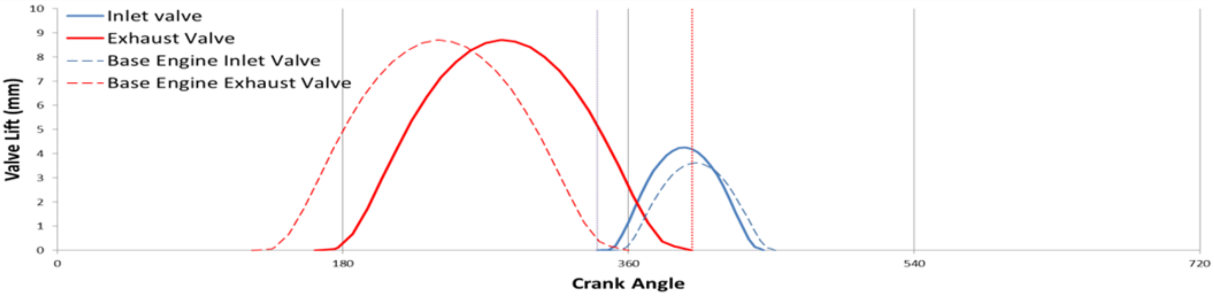

Figure 22. Case C un-throttled valve lift profile

The residual gas fraction shown in Table is considerably less than that of the throttled engine - an interesting phenomenon as the valve overlap is identical between cases. The main difference between them is the difference in intake pressure. The throttled case has a significantly lower intake pressure, creating a large pressure difference across the valve. When valve overlap occurs in the throttled case, the pressure gradient promotes high fractions of residual gas. However, with the absence of the throttle plate, the pressure difference between exhaust and inlet valves is negated, removing the driving force promoting backflow. This phenomenon can be clearly identified in Figure 23, where minimal backflow occurs with large valve overlap compared to the dashed line of the throttled case. The vertical dashed black line signifies the early intake valve closing angle. Other notable differences to baseline include an increase in EGR over baseline, leading to a reduction in NOx, an increase in PMEP, coupled with a decrease in GMEP which reduces the overall gas exchange efficiency, and increases fuel consumption.

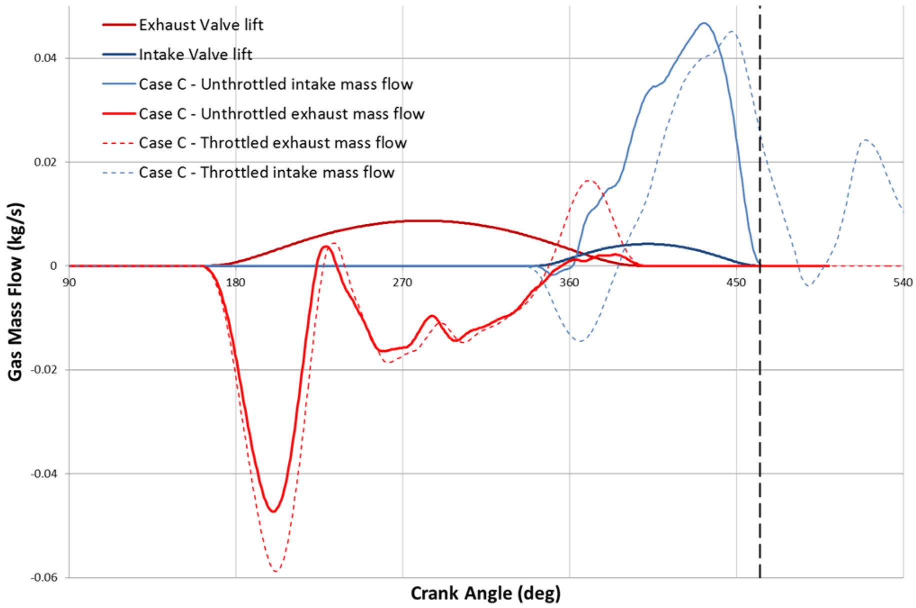

Figure 23. Case C comparison between throttled and un-throttled mass flow rate

Table 7 - Case C un-throttled tabulated summary of results

Metric	Unit	Un-throttled Baseline	Throttled Case C	Un-throttled Case C	% Difference to Unthrottled baseline
ISFC	kg/kW/hr	0.213212	0.221894	0.213357	0.07
GMEP	bar	6.25473	6.20753	6.05146	-3.25
PMEP	bar	-0.202671	-0.311634	-0.214545	5.86
GMEP/PMEP	-	29.31	19.88	27.42	-6.46
CO	ppm	4692.92	4061.4	4425.66	-5.69
HC	ppm	62.4894	53.4252	86.7074	38.76
NOx	ppm	3830.26	2815.27	3512.47	-8.30
Residual gas fraction	%	6.78066	20.1663	11.7741	73.64
EVO_deg	deg	-130.6	-162	-162	-
EVC_deg	deg	8.6	40	40	-
IVO_deg	deg	-7.078	-20	-20	-
IVC_deg	deg	97.8	214	103.312	-

Throttled VVT

This section provides a summary of observations, compiled to explain the effects of each valve timing event in the throttled engine configuration. ‘Late’ and ‘early’ refer to the relative change of the valve timing in respect to baseline.

Late exhaust valve opening marginally

increases expansion ratio benefitting cycle work. However, the overall power loop area improvement is outweighed by the increase in negative pumping work during the exhaust stroke. The cause of this phenomenon can be related to the further expansion of the exhaust gasses which reduce the pressure difference over the exhaust valve, causing the slower evacuation of exhaust gasses from the cylinder. As a consequence, a higher cylinder pressure during the initial phase of the exhaust stroke is seen. A trade-off is present between the work gained by further expanding the exhaust gasses, and the work lost due the blow down pumping work needed to evacuate the combusted gasses.

The exhaust duration remains constant in all cases; therefore late EVO translates to a Late EVC. In all cases where late EVC occurs, the cycle pumping torque is reduced. However this is at the expense of increased exhaust gas residuals. At a condition where exhaust gas residual fraction is constant, a relative initial increase in cylinder pressure translates into higher maximum cylinder pressure (P_{max}) and an overall increase in expansion work. However with an increase in exhaust residuals between similar cases, inert gasses that are pre-combusted reduce the combustion temperature, therefore inhibiting the expansion work. To counteract this effect, greater throttle angles can be achieved to maintain load, therefore increasing intake pressure and reducing the effect of pumping loss and gaining a fuel consumption benefit.

Cases with advanced IVO before TDC experience the greatest pressure difference across the inlet and exhaust valves. Valve opening and closing events induce large pressure waves that influence the cycle considerably. This could lead to inconsistent cycle performance over different speed and load key points. Retarding intake valve opening angle has proven beneficial to PMEP by delaying the pressure drop in the cylinder. However when employed with a fixed valve duration, the consequence is delayed intake valve closing.

Key point B demonstrates the loss of effective compression ratio where IVO occurs considerably beyond BDC. Low mean piston speeds equate to low gas momentum therefore cylinder mixture mass is lost due to back flow out of the intake valve at the beginning of the compression stroke. The limitation of fixed valve duration creates a trade-off between the gain achieved at IVO and IVC. What is interesting is that the strategy capable of the greatest fuel consumption benefit at the studied speed and load, case B, relies on both intake valve timing events to be retarded 37degrees from baseline.

In summary, an ISFC improvement of 2.5% is achieved over the base engine at key point ‘B’. The gas exchange efficiency has improved 30%, owing its benefit to the improvement in PMEP achieved with retarded EVC and IVO.

Un-throttled VVT

The base un-throttled engine maintains the original exhaust valve timing and duration, as well as the intake opening timing from the baseline engine. The Intake valve duration and lift however, reduces in order to maintain the normalised engine load. Deleting the throttle raised the static intake pressure to atmospheric level in all cases, significantly reducing the pumping work. A decrease of PMEP was observed between the throttled and un-throttled baseline engine, reduced by 52% from 0.44bar to 0.21bar. Combined with a relatively small decrease in GMEP of 3% between strategies, the gas exchange efficiency has dramatically increased by almost 100%. The sum of the above reductions equates to a 4.6% indicated specific fuel consumption benefit. ISFC improvement is slightly greater than seen in the test data, however the engine model results are considered an over estimate as combustion duration effects due to EGR are not taken into account.

Test engine data detailed a further 2% reduction in ISFC with the addition of VVT to the un-throttled strategy; however this benefit was not seen on the Wave engine model. According to the work carried out in relevant research papers (Cairns et al, 2009), increasing the valve overlap in engines optimised for un-throttled operation leads to an increase in EGR, which can be traded off for an increase in intake valve duration. An important observation was made from simulated results at case-C where a key point with higher valve overlap did not lead to an increase in EGR. Further investigation was conducted to understand the mechanism that differed from the test engine to produce different results.

Table 1. ISFC difference between test and simulation data

	Test engine data		Wave engine simulation	
	ISFC		ISFC	
Unit	g/kW.h	% diff	g/kW.h	% diff
Base	224.47	0.00	223.526	0.00
Base VVT	219.85	-2.06	217.901	-2.52
Un-throttled	215.69	-3.91	213.212	-4.61
Un-throttled VVT	211.79	-5.65	212.271	-5.04

The simulation results for all of the un-throttled cases show both exhaust and intake manifold pressures at atmospheric pressure. With negligible pressure difference across both of the valves, there is no driving force to promote internal EGR apart from the piston. With low values of valve overlap symmetrical about TDC, marginal increase of internal EGR is seen. The centre of valve overlap is defined as the angle mid-point between EVC and IVO. Regions of highest EGR relate to valve timing strategies where both EVC and IVO are simultaneously advanced or retarded from TDC.

The further the centre of valve overlap is moved from TDC, the more influence the piston motion has on the gas flow. To further support this statement, Figure 24 clearly shows the two regions on the valve timing map where the EGR is at its highest. The island marked ‘A’ denotes an area where the centre of valve overlap is considerably beyond TDC, therefore backflow in the exhaust port occurs due to the piston’s downward motion, thus increasing the residual mass fraction. Similarly, the island marked ‘B’ denotes an area in which the intake valve opens considerably earlier than TDC, so backflow occurs in the inlet port as the piston is tending towards TDC during the exhaust stroke.

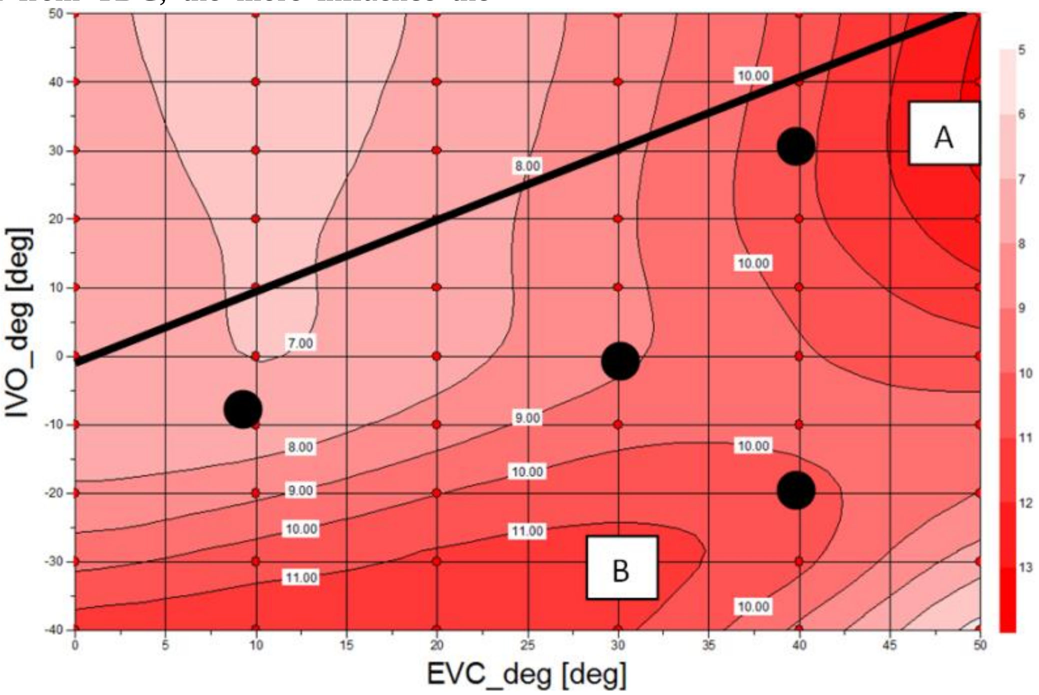

Figure 24. Un-throttled residual gas mass fraction %

The region from the top left to the bottom right of the figure, sees considerably less EGR, whereas the throttled case benefits from an EGR increase. In cases of high overlap that is symmetrical about top dead centre, there is minimal driving force to increase backflow. Throttled cases with high valve overlap responded well with a considerable increase in EGR due to the large pressure differential between the exhaust and intake manifolds.

Figure 25 shows the effect that the centre line of overlap has on the magnitude of internal EGR. Each line denotes a varying amount of overlap. The key observation to be concluded from the plot is that higher levels of EGR have been achieved with lower valve overlap where the centre of the overlap moves further past TDC, supporting the statement that the further the centre of overlap moves from TDC, the greater the influence of piston motion on EGR.

The cause of this lack in pressure differential is believed to be due to pressure waves in the exhaust system. Exhaust runner length and exhaust gas temperature have a large effect on the time the pressure waves take to reflect. The engine model utilises a relatively short intake an exhaust runner length, optimised for high engine speeds. Further investigation on the intake and exhaust manifold length is required to optimise the length parameters and take advantage of the EGR trade-off.

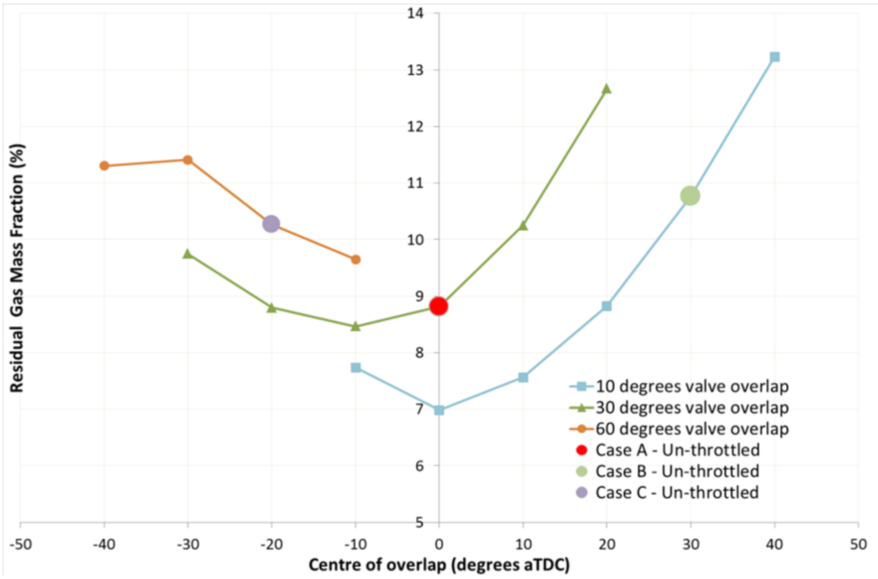

Figure 25. Overlap center in reference to TDC

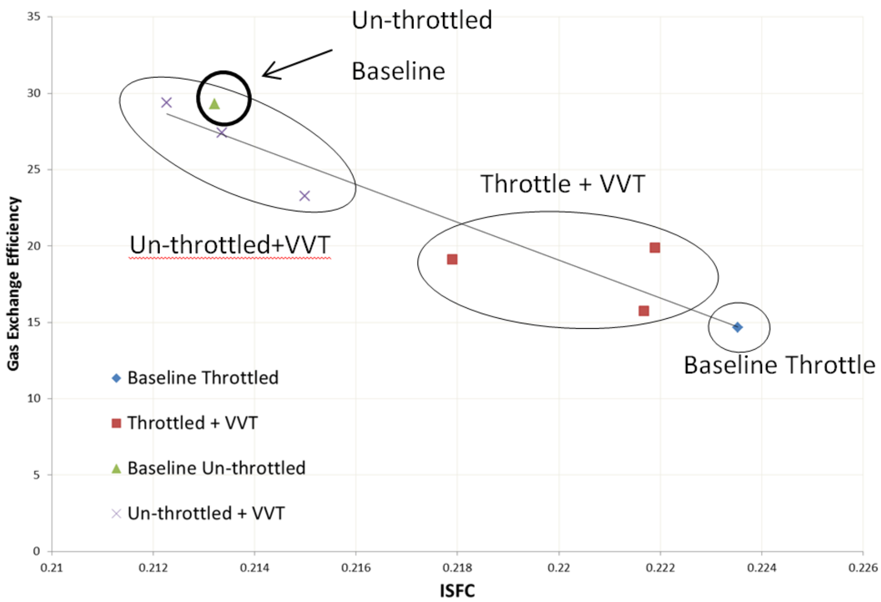

Figure 26. Overlap center in reference to TDC

4.5 Fuel consumption

Figure 26 shows the relationship between fuel consumption and gas exchange efficiency. An increase in gas exchange efficiency in the un-throttled case highlights the reduced need to burn fuel at a constant load, where the pumping work reduces dramatically. The trend in fuel consumption benefit due to the different valve strategies is very clear. The highest fuel consumption is seen in the base throttled engine where the largest pumping loop was observed. Conversely, a significant consumption improvement is achieved by implementing the un-throttled valve strategy. The mechanisms that are responsible for the fuel consumption benefit have been discussed previously.

5. Conclusions

This section is a breakdown of conclusions drawn throughout this study with justification provided for each point.

In the case of dual independent VVT a significant fuel consumption benefit of 2.5% can be achieved at part load with valve overlap. The mechanism most influential to its improvement is the increase in internal EGR. PMEP was, also, reduced. Cylinder pressure remains for the initial part of the intake stroke, reducing the pumping loop area. NO_x is reduced as a consequence of reduced combustion temperature. Increased throttle openings are achievable with the addition of residual gas, resulting in increased intake pressures and improved PMEP. Combusted gas displaces a fraction of fresh mixture, thus reducing engine torque. To compensate, more mixture is drawing into the cylinder requiring larger throttle openings. Exhaust valve opening timing is a compromise between increased expansion work and reduced blow down work. It is beneficial to open the valve close to BDC at part load. Late valve opening increases expansion at the cost of increased pumping work reducing GMEP. Early valve opening reduces expansion work, however a higher pressure delta across the valve aids blow down. With a fixed intake valve duration strategy there is a compromise between the benefit gained by delaying IVO, and the intake mass lost due to LIVC, resulting in a reduction of effective compression ratio.

In the case of un-throttled operation, deleting the throttle increases the intake pressure to ambient, considerably raising cylinder pressure during the induction stroke. As a result, PMEP is improved by 55% resulting in a 4.6% fuel consumption benefit over baseline. The ratio of GMEP to PMEP increased as a result of improved intake pressures and gas exchange efficiency doubled.

In the case of un-throttled operation with dual independent VVT the fuel consumption benefit simulated yields limited benefit from VVT, which is however still significantly lower in magnitude than that claimed by other research projects. Pressure wave reflection time is heavily influenced by the intake and exhaust manifold length. Short exhaust manifold length was optimised for high engine speed/load. Published papers support the benefit of re-optimising intake and exhaust manifold lengths in an un-throttled engine to improve part load performance. Increasing intake pressure to ambient considerably reduces the pressure difference across the exhaust and intake valve. The pressure difference reduces the driving force that encourages backflow and internal EGR at valve overlaps central at TDC. Relatively low values of EGR are seen at large overlaps where the midpoint is close to TDC. The largest influence on internal EGR is early/late overlap where the piston motion provides the biggest impact. Higher EGR was observed at regions where the centre of overlap was advanced or retarded from TDC. Higher, in fact, than a larger overlap strategy with a centre of overlap closer to TDC. Further downsizing the engine displacement will reduce the benefit achieved as the engine load is inherently increased. The benefit of EGR is reduced in proportion to load increase.

Overall, cam phasers are a relatively cheap technology that needs little or no engine architecture changes to implement. When applied to a throttled engine, good fuel consumption gains are seen with the added benefit of emissions reduction. The cost to benefit ratio is high. The addition of continuously variable valve lift to the strategy allows un-throttled operation to be achieved. Deleting the throttle further increases intake pressure, dramatically reducing pumping torque that is responsible for poor fuel consumption.

6. References

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