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Experimental study of intelligent valve actuation for high efficiency spark ignition engines

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ABSTRACT

Engine downsizing has been shown as an effective means to reduce the vehicle's fuel consumption but the full potential of engine downsizing is limited by the knocking combustion at boosted operations and the presence of pumping loss at part load conditions. In this work, an electro-mechanical valvetrain system named iVT (intelligent Valve Technology) by Camcon was used to investigate how the independently controlled variable valve timing and duration can be applied to minimise the knocking combustion by altering the effective compression ratio (ECR) at high load via Early Intake Valve Closure (EIVC) or Late Intake Valve Closure (LIVC), as well as reducing the pumping loss at part load. In particular, the effect of different valve lifts with fixed valve timings and constant duration was studied on the pumping loss, combustion process and emissions. The results show that fuel consumption was reduced up to 2.5% using iVT system compared to the baseline valve profile at 9bar net IMEP.

1 INTRODUCTION

IC engines have been the main power plants for various transport on land and sea. World-wide daily use of vehicles with IC engines in the 20th century has led to the very stringent legislation on their pollutant emissions over the last few decades. Moreover European Parliament and the Council set regulation for the maximum value of manufacturer's fleet average CO_2 emission level, targeting to 95g/km from 2020 (1) in order to combat the global warming caused by increasing CO_2 concentration. If the average value exceeds the limit, the manufacturer has to pay monetary penalty for each registered car (2). Another main issue associated with the use of IC engines is increasing fossil fuel consumption, which can lead to resource depletion as the amount of vehicles increases dramatically.

To fulfil the above requirements, automotive industry have been developing new technologies to improve the efficiency of modern IC engines. Downsizing is one of the successful methods of reducing fuel consumption and CO_2 emissions from a Spark Ignition (SI) engines. Engine downsizing can significantly reduce fuel consumption by operating the engine closer to its minimum fuel consumption region by reducing the engine displacement and with boosting. In this way pumping losses at part-load operations are reduced. However downsized engines are more prone to knocking combustion at high boost.

Variable Valve Actuation (VVA) can be used to reduce pumping losses at part load conditions and minimise knocking combustion at high loads by means of ECR reduction with Miller cycle. VVA has been studied for more than 20 years but not yet fully

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implemented in mass production vehicles. There are a lot of prototypes and all of them can be divided into two main types: cam-based and camless systems. Cam-based systems represent engines with modified camshaft valve train where camshaft is driven by crankshaft, whereas engines with camless systems have actuators which control valve events independently of the crankshaft. The main aim of a VVA system is to modify valve events accordingly to the changing engine load and speed. The parameters that can be controlled by VVA system are: valve opening and closing timing, duration of the valve event and valve lift. Some cam-based systems such as BMW Valvetronic, Toyota Valvematic, Honda VTEC and Fiat Multi-air have been implemented on some production vehicles (3-6). However none of those systems were able to provide continuous and fully flexible variation of lift and valve timings for an individual valve.

Camcon developed an electro-mechanical valvetrain system (iVT) capable of full control of the valve events at engine speeds up to 6000rpm. This system was installed on a single cylinder research engine for both intake and exhaust valves to study valve profile effects on fuel economy and emissions. In this paper, the effect of Miller cycle with different valve profiles will be presented and compared to the baseline at various engine loads at a constant engine speed of 1500rpm.

2 EXPERIMENTAL SETUP

A single cylinder SI direct injection gasoline engine with 4 valves was used for this research and the specifications are given in Table 1. The engine is equipped with iVT (intelligent Valve Technology) valvetrain system for each valve as shown in Figure 1. Each valve is actuated via an independent camshaft driven by an electrical motor. Full rotation of the cam produces a full lift event whereas partial rotation allows for lift control. Varying of the motor speed controls the start and duration of the valve opening. Combination of cam rotation and motor speed allows for independent control of timings and lifts of each valve through the valve control software. The fuel was pressurised to 50bar and supplied to the DI gasoline injector, the flow rate is measured by an instantaneous fuel flow meter (Endress+Hauser Promass 83A Coriolis) before the injector. The air supplied to the engine was either at room temperature and pressure or at pre-set boost pressure from an external supercharger with closed loop control. The air mass flow rate was measured by a laminar flow meter (Hasting HFM-200) installed before the throttle. The instantaneous intake and exhaust pressures were measured by a piezo-resistive pressure transducer located just before the intake valves and another one in the exhaust port respectively. Heat release and combustion characteristics were calculated by a combustion analysis software based on the instantaneous cylinder pressure from a piezo-electric pressure transducer and crank angle from a crankshaft encoder. The emissions were measured by a Horiba 7170DEGR. The engine was coupled to an AC dynamometer and installed on the test bed with closed loop control of oil and coolant circuits. The dynamometer allowed for motored and fired operation of the engine at set speeds. The spark timing, throttle angle and AFR were controlled via an engine control software.

Table 1. Engine specifications.

Engine Type	4-stroke, single cylinder, 2 intake and 2 exhaust valves
Bore x Stroke	81mm x 89mm
Connecting Rod length	155.5mm
Compression Ratio	10.8:1
Displacement Volume	458.6cc
Intake Valves Diameter (2)	29mm
Exhaust Valves Diameter (2)	26mm
Fuel Injection	Direct Injection

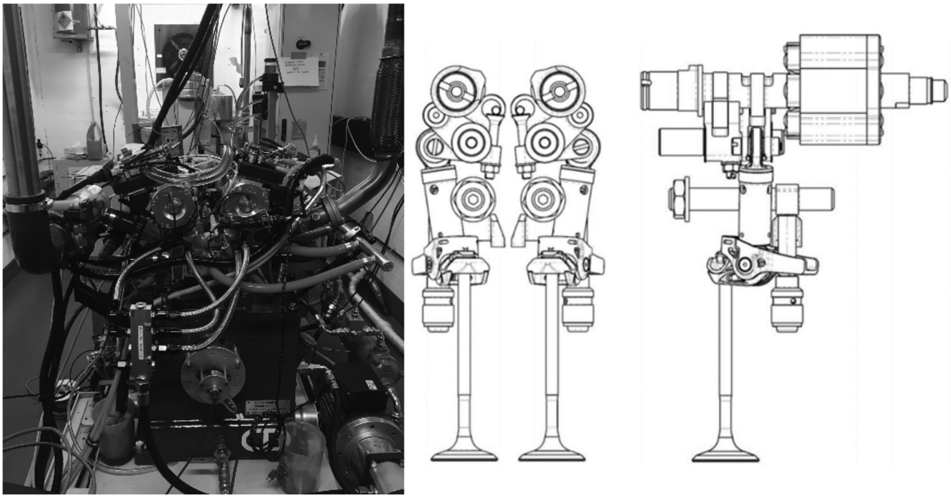

Figure 1. Single cylinder DI gasoline engine and iVT system (7).

3 TEST CONDITIONS AND MODES OF VALVE OPERATIONS

In this research, several valve profiles were applied to both intake valves (two valve mode) or one of the intake valves (single valve mode) and their effect on engine performance, combustion and emissions were investigated at 4, 6, 9 and 12.6bar net IMEP at a constant engine speed of 1500rpm. The fuel used was EU VI 95 RON Gasoline (E10) with 10% Ethanol content by volume. Fuel specifications can be found in Table 2. All tests were conducted with a relative AFR (Lambda) of 1 and the fuel injection timing was fixed at 268deg CA BTDC at an injection pressure of 50bar. The spark timing was set at MBT unless it was knock limited at higher load conditions.

Table 2. Fuel properties.

Fuel	95 Ron Gasoline E10
Density at 15 ºC (kg/m^3)	746.1
Higher calorific value (kJ/kg)	44220
Lower calorific value (kJ/kg)	41420
Stoichiometric AFR	13.92:1

The engine could be operated with both intake valves or one of them using the iVT system. In the case of the single valve mode operation, one of the intake valves was permanently closed during testing in order to induce swirl motion inside the cylinder.

Figure 2 shows the five valve profiles which were used for two valve and single valve modes for all the load cases: Baseline (BSL), Late Intake Valve Closing (LIVC) and Early Intake Valve Closing (EIVC) with three maximum valve lift variations from 100% to 64% where the duration and valve timings were kept constant. The exhaust valve profile was unchanged for all the tests. Valve parameters are shown in Table 3. In the paper, the results will be presented when both intake valves were actuated and the single valve results will be published in a separate paper.

As in previous researches, EIVC and LIVC profiles were used to reduce effective compression ratio of the engine in order to study the effects of Miller cycle on the efficiency, fuel economy and emissions (8 - 10). Additionally, EIVC was set with three different valve lifts in order to investigate their effects on pumping losses and combustion process, which could not be done in the previous studies by other researchers when the valve lift and IVC could not be independently controlled.

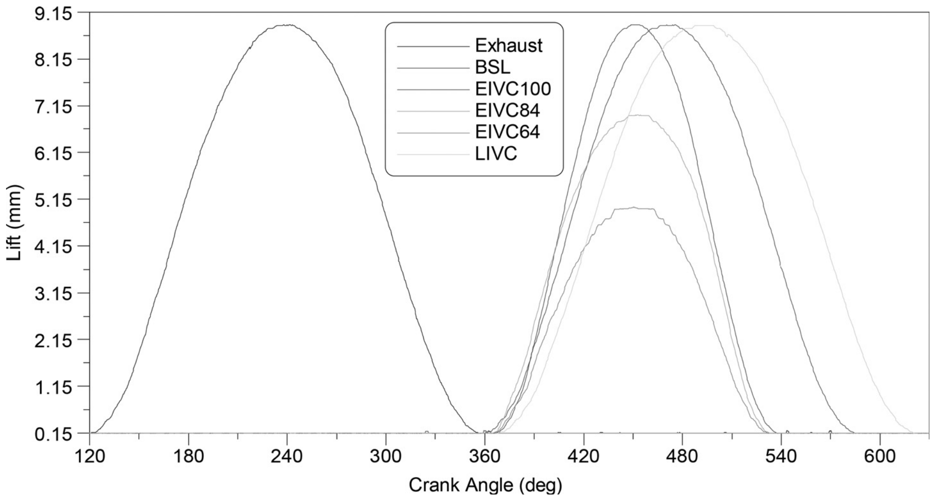

Figure 2. Valve profiles for single and two valve modes.

Table 3. Valve timings and durations.

Valve Profile	Lift (mm)	Duration (CA deg)	IVO/EVO (CA deg)	IVC/EVC (CA deg)
EIVC100	8.9	152	376	528
EIVC84	7	152	372	524
EIVC64	5	146	375	521
LIVC	8.9	226	382	608
Standard	8.9	200	373	573
Exhaust	8.9	211	133	344

4 RESULTS

4.1 Effect of valve profiles on the effective compression ratio

To better understand experimental results the effective compression ratio (ECR) must be evaluated for each test as IVC timing and valve lift directly affecting it. ECR was calculated based on the in-cylinder pressure for each valve profile and for each load point. The start of compression is defined at the crank angle when the polytropic compression line and intake manifold pressure are crossing on the logP-logV diagram (Figure 3). The cylinder volume at this crank angle is then divided by TDC volume to get the ECR.

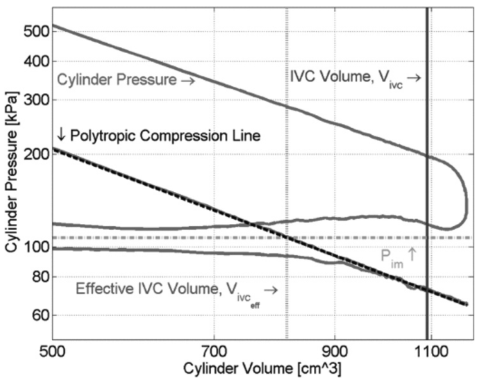

Figure 3. Pressure based method of ECR estimation (11).

As shown in Figure 4, the baseline profile has around 10.4 – 9.9 ECR from low to high load whereas EIVC100 has the highest ECR throughout the whole load range, between 10.8 and 10.5. The difference in ECRs between the EIVC100 and the baseline case is due to the IVC timing, which is around 10 degrees before BDC for EIVC100 and is 30 degrees after BDC for baseline profile. As a result, a small amount of fresh charge is pushed back into the intake during the compression stroke before the compression starts, resulting in a lower effective compression ratio for BSL. With increasing load more charge is being expelled into the intake manifold thus ECR is reduced further.

EIVC profiles have almost identical IVC timings, however the effective compression ratio is different for each of them. The reason for that is the variation of the valve lift. The smaller the lift the less fresh charge is sucked into the cylinder therefore lower effective compression ratio. This is clearly demonstrated by EIVC profiles across all the load points. Also with increasing load the ECR decreases respectively as more charge is needed while a certain valve lift creates a constant flow restriction throughout the whole load range.

LIVC produced the lowest effective compression ratio (8 to 8.3) as a lot of fresh charge was expelled into the intake manifold due to very late IVC. Opposite to other profiles ECR is increasing with load for LIVC. This is due to higher pressure in the intake manifold as throttle opens more. When intake pressure is close to atmospheric the pressure difference between in-cylinder and manifold pressure becomes smaller and less percentage of the fresh charge is expelled into the intake.

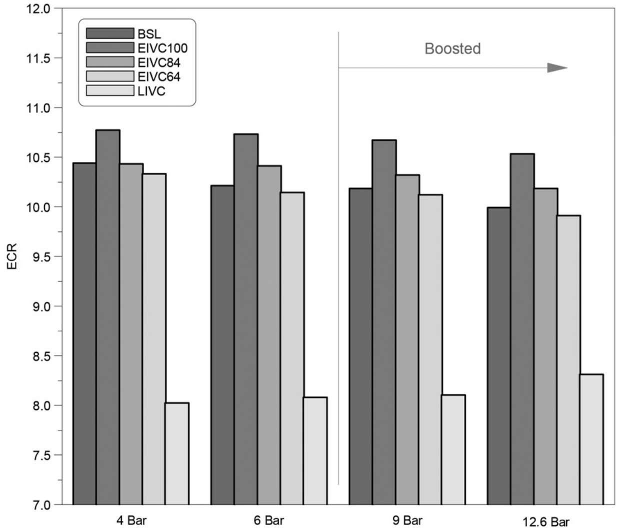

Figure 4. ECR comparison.

At each load in most cases the throttle was opened more for those profiles with lower ECR as higher intake pressure was needed in order to achieve the same net IMEP as shown in Table 4. However there were some cases where throttle angle was the same but ECR was different, this was due to other factors which increased the intake pressure regardless of throttle angle. This will be explained in the next section.

Table 4. Throttle angle and intake pressure comparison.

Pro-file	BSL		EIVC100		EIVC84		EIVC64		LIVC	
Net IMEP- (bar)	**Throttle (%)**					**Intake Pressure (bar)**				
4	1.83	0.47	1.80	0.42	1.85	0.46	1.85	0.50	1.96	0.61
6	2.75	0.64	2.68	0.59	2.82	0.65	2.85	0.69	3.80	0.83
9	2.18	0.83	2.15	0.78	2.24	0.85	2.24	0.90	2.44	1.08
12.6	3.75	1.22	3.40	1.12	3.65	1.20	4.65	1.34	14.00	1.48

4.2 Effect of valve profiles on the pumping loss

In a conventional throttle controlled SI engine the pumping loss is created due to the drop of manifold pressure below atmospheric pressure and it is one of the major causes of low engine efficiency at the low load. Comparison of Pumping Mean Effective Pressure (PMEP) for various valve profiles is shown in Figure 5.

At 4bar net IMEP, the LIVC valve profile produced the lowest negative PMEP due to larger throttle opening (0.13% more than STD) required to maintain the same IMEP with the lowest ECR. Also the charge was pushed out from the cylinder into the intake manifold during compression stroke, which increased the intake pressure for the subsequent intake strokes lowering pumping losses (Table 4).

Amongst all EIVC profiles, EIVC100 produced the highest negative PMEP, whereas EIVC64 produced the least amount of pumping loss. For EIVC100 to achieve 4bar net IMEP, the throttle was closed by 0.05% more than for the lower lifts because of higher ECR, this resulted in increased pumping loss. For EIVC84 and EIVC64 valve profiles, the throttle position was identical however intake manifold pressure was higher for the lower lift profile. A lower valve lift leads to lower flow area and hence a higher intake pressure is required to allow the same amount of air into the cylinder as the higher lift. This is usually achieved by larger throttle opening. However, in the case with EIVC64, it was noted that a bigger drop in the in-cylinder pressure at IVO caused an increase in the intake port pressure which was enough to trap the required amount of air into the cylinder without altering throttle angle. Higher intake pressure led to lower pumping losses for EIVC64. Therefore, when operating an engine at low load with a small throttle opening, the lower intake valve lift increased the intake pressure and reduced pumping loss.

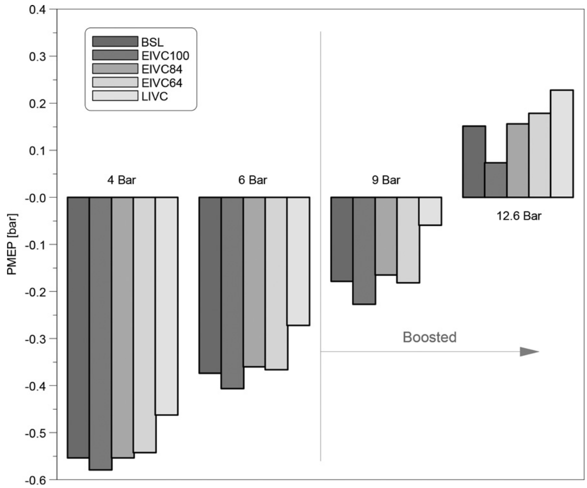

Figure 5. PMEP comparison.

However, an additional test with very short lift (EIVC44) produced the highest intake pressure and pumping losses larger by 0.01bar than EIVC64 (Table 5). According to J. B. Heywood this phenomenon could be explained by the relationship between valve lift and discharge coefficient (12). Depending on the valve shape, at high lifts the flow separates and discharge coefficient decreases generating higher pumping losses. At lower lifts the flow speed across the intake valves increases and the flow remains attached to the valve providing high discharge coefficient, therefore reducing pumping losses. However if the lift would be reduced further or flow speed would be increased, the flow might separate and cause an opposite effect, which was the case with EIVC44. The discharge coefficient at EIVC64 might have been higher than EIVC44 and this would reduce pumping losses, but this is purely theoretical and to quantify the results a measurement of discharge coefficient via CFD simulations is required. In the case of this experiment 64% lift was the optimum for EIVC profiles in reduction of pumping losses, therefore EIVC44 was not considered in further analysis.

Table 5. EIVC64 and 44 comparison.

	EIVC64	**EIVC44**
Intake Pressure (bar)	0.495	0.601
PMEP (bar)	-0.54	-0.55

The intake pressure for baseline profile was higher than EIVC84 due to the charge being expelled from the cylinder during compression strokes even though throttle was opened slightly less. However, negative PMEP was the same as EIVC84 due to lower effective flow area during the opening of the valve for BSL profile.

To summarise, EIVC100 got the highest pumping losses due to the lowest intake pressure. LIVC achieved major reduction in pumping losses due to increased intake pressure and the largest throttle opening.

At 6bar net IMEP a similar trend is present. LIVC had the lowest ECR and required a larger throttle angle, which increased the intake pressure, providing the lowest pumping losses among other profiles. EIVC100 had the highest pumping losses due to the lowest intake pressure followed by the baseline profile. EIVC64 had a higher intake pressure than EIVC84 but also higher negative PMEP. This indicates that EIVC84 might have higher discharge coefficient. A larger engine load requires larger volume of charge in the cylinder therefore flow velocity through the intake valves has to increase in order to accommodate for this. As discussed previously the discharge coefficient is affected by the change in flow speed, due to this the discharge coefficient could be reduced at EIVC64 causing larger pumping loss even though intake pressure was higher than at 84% lift.

For 9bar net IMEP tests, an external supercharger was used to provide the pressurised air. It was necessary to provide additional 0.6bar of boost pressure for the LIVC profile as it was not capable of achieving this load point with naturally aspirated mode, whereas other profiles didn’t require boosting. Overall, the same trend was present as at 6bar. LIVC had the lowest negative PMEP due to the highest intake pressure. EIVC100 has the lowest intake pressure thus the largest pumping losses. EIVC64 and 84 had the same throttle angle but intake pressure was increased for EIVC64 due to lower lift. EIVC84 achieved lower pumping loss potentially due to higher discharge coefficient as in the case with 6bar net IMEP. However further increased flow speed could have reduced discharge coefficient even more for EIVC64 resulting in a higher negative PMEP than the baseline valve profile, even though it had much lower intake pressure.

At 12.6bar net IMEP a positive pumping work was present due to additional boost. LIVC had the highest positive PMEP due to significantly higher intake pressure compared to the rest of valve profiles, followed by EIVC64. EIVC100 had the lowest positive PMEP due to lowest intake pressure which corresponds to the lowest throttle angle. EIVC84 had slightly higher positive pumping work compared to the baseline even though the intake pressure was slightly lower, this was caused by larger effective flow area during valve opening of EIVC84.

From the above results it is evident that intake port pressure has strong influence on the pumping losses. The intake port pressure can be increased via the throttle or valve lift. It is also possible that discharge coefficient can affect pumping losses. The discharge coefficient is influenced by the flow speed which can be altered either by intake pressure or valve lift or both. Where the difference between intake pressures is not significant the discharge coefficient is more dominant in reduction of pumping losses. Across all load cases LIVC reduced pumping loses significantly compared to the baseline profile due to significant increase in intake pressure. EIVC profiles with shorter lifts also were successful in reducing pumping losses due to higher intake pressure and possibly increased discharge coefficient.

4.3 Effect of valve profiles on the combustion process

Changes of the intake valve profile can also influence combustion duration, which is another factor affecting fuel economy. Comparison of combustion durations and spark timings can be seen in Figure 6.

At 4bar net IMEP the baseline valve profile produced the shortest combustion duration (10-90% burn), whereas EIVC84 and 64 led to the longest duration indicating that flame speed was lower due to reduced tumble flow and hence turbulence intensity. EIVC100 and LIVC profiles had similar combustion durations which were slightly longer (less than 2 degCA) than that of the baseline valve profile. The MBT spark timing was advanced accordingly to compensate for the longer combustion duration, however the flame development angle (crank angle degrees between spark timing and 10% burn) for short lift profiles was more than 30% longer than the baseline (Table 6). LIVC also had slightly longer flame development angle indicating weaker in-cylinder turbulence than the baseline.

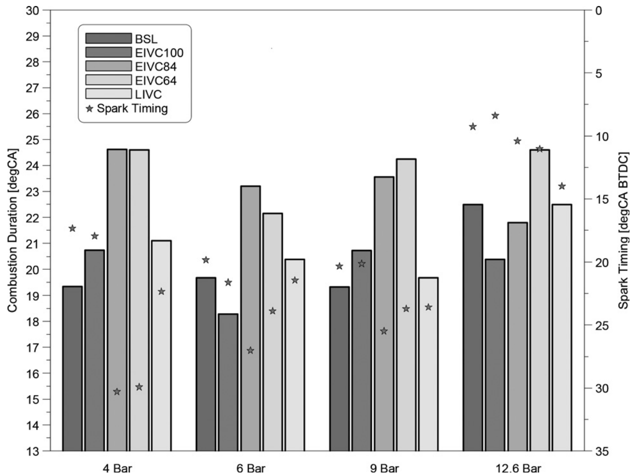

Figure 6. Combustion duration and spark timing comparison.

At 6bar net IMEP combustion duration was reduced for all profiles except for the BSL profile with shortest being EIVC100 and longest EIVC84. The MBT spark timing was advanced respectively for profiles with longer flame development angle, longest being EIVC with shorter lifts followed by LIVC. EIVC100 had slightly longer flame development angle than the baseline. Therefore the spark timing was also slightly advanced.

Table 6. Flame development angle and in-cylinder lambda comparison.

Profile	BSL		EIVC100		EIVC84		EIVC64		LIVC	
Net IMEP (bar)	**Flame Development Angle (degCA)**					**Lambda Cylinder**				
4	19.5	0.98	19.7	0.98	26.8	0.97	26.1	0.98	22.4	0.96
6	19.5	0.98	20.3	0.98	24.3	0.98	23.9	0.99	21.1	0.97
9	18.3	0.98	20.2	0.99	22.4	0.99	23.1	0.99	19.8	0.98
12.6	16.7	0.98	16.5	0.98	18.2	0.98	19.5	0.96	19.0	0.98

At 9bar net IMEP, the combustion duration was similar to 4bar but the advance in spark timing was limited by knocking combustion. LIVC spark timing was advanced by a few degrees compared to baseline profile due to lower in-cylinder temperature at the end of compression stroke thanks to lower effective compression ratio. This resulted in reduced knocking tendency and allowed to advance spark timing. EIVC84 and 64 spark timings were also more advanced than EIVC100 due to lower ECR. The longest flame development angle was found with EIVC64 and 84, followed by EIVC100 and LIVC.

At 12.6bar net IMEP knocking had more noticeable effect on spark timing. EIVC100 has the shortest combustion duration and the most retarded spark timing due to highest ECR. Whereas LIVC has the same combustion duration as the baseline but more advance spark timing due to lower knocking tendency. This was achieved because of reduced in-cylinder temperature as effective compression ratio was lowered. The same effect can be observed with EIVC84 and 64 spark timing compared to EIVC100. The longest flame development angle was produced by EIVC64 followed by LIVC and EIVC84. EIVC100 achieved shorter flame development angle and duration than BSL.

From the discussion above it can be concluded that lower ECR causes weaker in-cylinder flow motion leading to longer combustion duration and flame development angle. This effect is more prominent at lower load and very high load with knock limited combustion. On the other hand, a lower effective compression ratio allows to advance knock limited spark timing further. Shorter valve lifts lead to slower combustion except at the highest load case. This is evidenced by longer flame development angle and combustion duration of EIVC84 and 64. In almost all the load cases shorter lift had more noticeable effect on in-cylinder turbulence than the effect of lower ECR.

4.4 Effect of valve profiles on the indicated fuel consumption (ISFC)

Figure 7 demonstrates how valve profiles influence ISFC at various load points. At low load (4bar IMEP) LIVC valve profile reduced fuel consumption by 3.4g/kWh compared to the baseline due to reduced pumping losses. EIVC64 was also successful in reduction of ISFC however due to very poor in-cylinder charge mixing and slow combustion it was less effective than LIVC. EIVC100 and 84 lead to increased fuel consumption due to larger pumping losses and longer combustion duration.

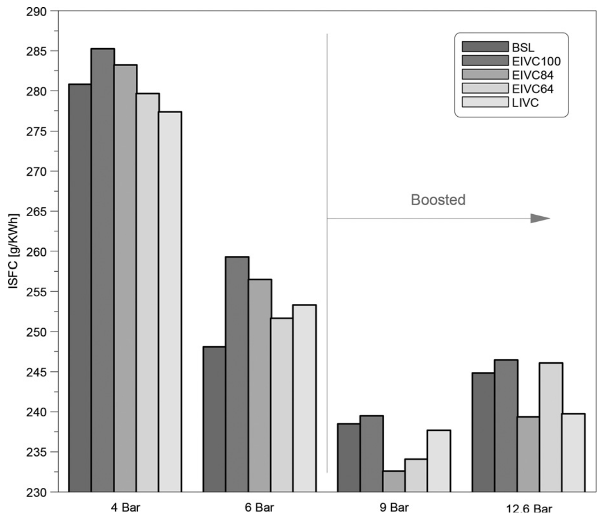

Figure 7. ISFC comparison.

At 6bar net IMEP the lowest fuel consumption was achieved by the standard profile due to shorter combustion duration and medium pumping losses. EIVC64 was the second best followed by LIVC profile. The reason why LIVC achieved higher ISFC value than EIVC64 even though it has shorter combustion duration and lower pumping losses, is richer local in-cylinder charge (Table 5). This could be due to some of the fuel being pushed out into the intake manifold and in the next cycle this fuel could be concentrated in one spot instead of homogeneously mixing. Whereas EIVC64 had in-cylinder AFR closest to stoichiometric. The worst fuel economy was achieved by EIVC100 due to the largest pumping loss.

A similar result for LIVC was achieved at 9bar net IMEP. Despite a shorter combustion duration and the lowest pumping loss LIVC achieved higher fuel consumption than EIVC84 and 64 due to richer in-cylinder mixture. Baseline profile also had higher fuel consumption due to lower in-cylinder lambda. EIVC100 had the largest ISFC due to larger pumping losses than the rest of the profiles and EIVC84 had the lowest fuel consumption due to stoichiometric in-cylinder AFR and lower pumping losses and shorter combustion duration than EIVC64.

At 12.6bar net IMEP in-cylinder lambda was slightly richer for EIVC84 than for LIVC, but EIVC84 had shorter combustion duration which led to lower ISFC. EIVC100 had the highest fuel consumption due to the smallest positive pumping work among other profiles. EIVC64 was the second worst due to longest combustion duration and richest in-cylinder mixture.

From the above results it is evident that valve profiles influence pumping losses and combustion duration which in turn affects fuel consumption. Moreover, in-cylinder AFR is affected by valve profiles influencing ISFC. Overall Miller cylce was succefull in reducing fuel consumption at low loads thanks to lower pumping loss and at high loads where spark timing was knock limited.

At 9 and 12.6bar net IMEP loads the net fuel consumption should also take into account of the compression work required where additional boost pressure was used. An external mechanical supercharger was used to provide the boost and was powered by electrical supply, in automotive application the supercharger would be driven by the engine crankshaft which would cause a power loss affecting ISFC. Due to this the compressor work was calculated and ISFC was then corrected taking into account supercharger work using the equations and variables below.

$$W_c = \frac{\dot{m}_{air} \times C_p \times T1}{\eta_C \times \eta_m} \times \left[(P2 \div P1)^{\frac{\gamma-1}{\gamma}} - 1 \right]$$

$$ISFC\ corrected = \frac{\dot{m}_{fuel}}{Pi - Wc}$$

Where,

$$C_p = 1.012\ J/gK,\ \gamma = 1.4,\ \eta_c = 60\%, \eta_m = 90\%$$

Figure 8 shows the values of ISFC when work required by the supercharger is taken into account. As mentioned before at 9bar net IMEP only LIVC profile required additional boost therefore the ISFC was corrected only for this profile. After correction LIVC achieved highest fuel consumption at 9bar due to the work required for the supercharger. At 12.6bar net IMEP all valve profiles had ISFCs corrected taking into account additional boost required for each profile. ISFC increased for all profiles proportional to the amount of required boost, for example EIVC100 had lowest increase in ISFC as it required the least boost whereas LIVC had the largest increase due to greatest amount of boost required. After correction EIVC84 still had the lowest ISFC among the rest of the profiles.

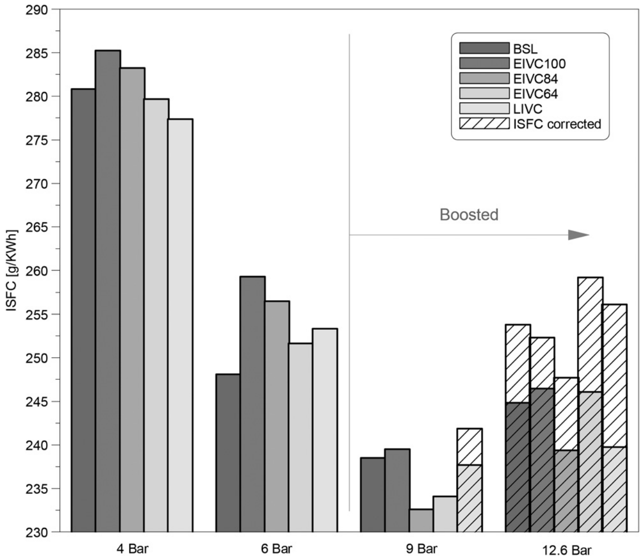

Figure 8. Comparison of ISFC vs corrected ISFC.

4.5 Effect of valve profiles on the exhaust emissions

Emissions results are shown in Figure 9a, b and c. Throughout all load cases higher concentration of ISCO was present with lower in-cylinder lambda i.e. combustion of fuel rich mixture. The lowest ISCO values were achieved when in-cylinder lambda was closer to 1 (stoichiometric combustion) with lowest value of 16 g/KWh at 9bar net IMEP with EIVC64 profile.

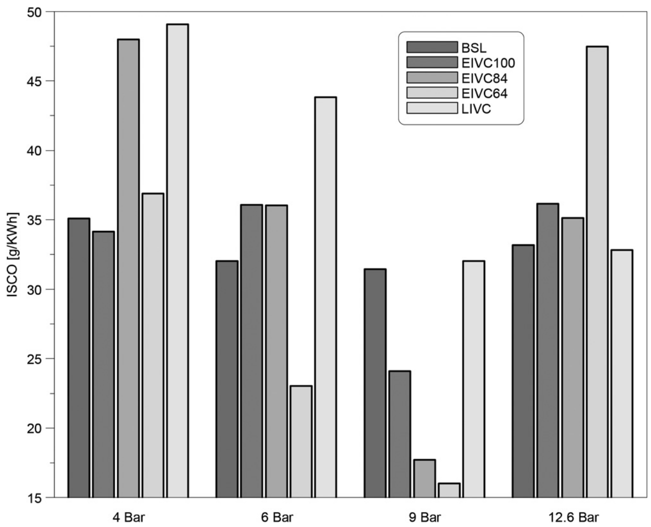

Figure 9a. ISCO comparison.

At 4bar net IMEP the lowest level of HC emissions was achieved by LIVC profile due to the lowest in-cylinder pressure during compression stroke which prevented some of the mixture being forced into crevices and reasonably fast combustion which prevented bulk quenching of the flame next to the cylinder walls, leading to more complete combustion compared to the rest of the profiles. At 6bar net IMEP the lowest ISHC was achieved by EIVC84 due to lowered in-cylinder pressure and improved combustion speed which allowed to reduce bulk quenching at the cylinder wall. At 9bar net IMEP HC emissions were dramatically reduced for all profiles indicating more favourable conditions for complete combustion. EIVC100 achieved lowest ISHC of 5.38 g/KWh due to lower in-cylinder pressure during combustion stroke and relatively fast combustion, reducing bulk quenching at the cylinder wall. At 12.6bar net IMEP HC emissions increased overall indicating less complete combustion due to stronger knocking. Lowest ISHC value was achieved by EIVC84 due to low in-cylinder pressure during combustion and fast combustion due to advance spark timing, which allowed to reduce quenching at the cylinder wall.

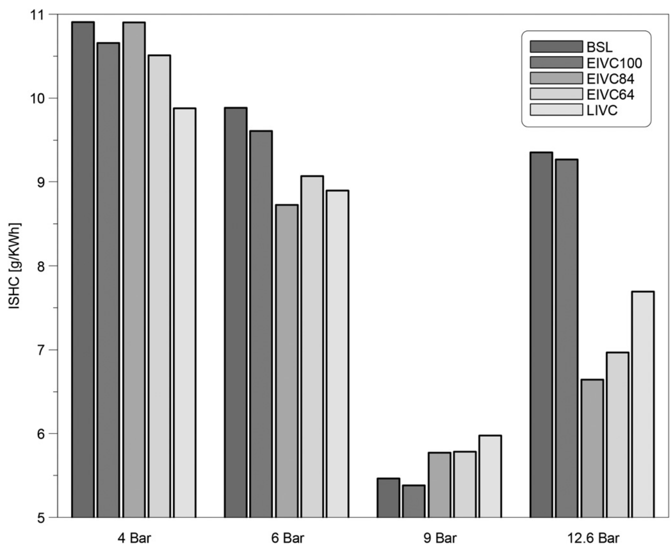

Figure 9b. ISHC comparison.

Lowest NOx level across all the load points was achieved with LIVC profile because of the lower effective compression ratio as indicated by the lower in-cylinder pressure during the compression stroke and lower peak temperature during combustion. As the load increased the ISNOx generally also increased due to increasing in-cylinder pressure during the compression stroke and peak temperature during combustion. EIVC profiles with shorter lifts generally produced higher NOx emissions because of higher pressure at the end of the compression stroke and more advanced spark timing which provides higher rate of temperature increase, so that in-cylinder peak temperature would be greater than from other profiles.

Overall EIVC and LIVC profiles were successful in reduction of emissions. The major reduction was achieved by LIVC in NOx emission across all the load cases.

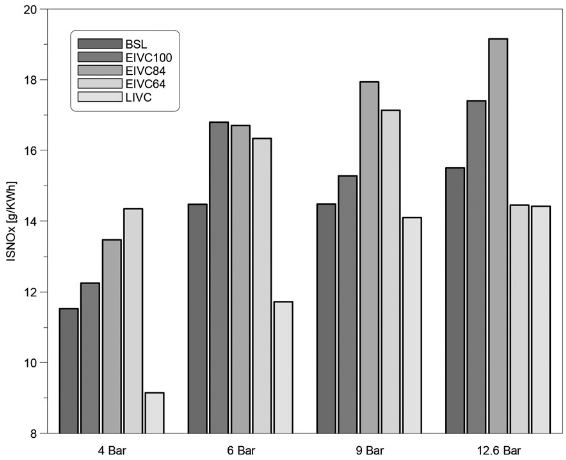

Figure 9c. ISNOx comparison.

5 CONCLUSIONS

In this study, the iVT system was employed to investigate the effect of valve lift profiles on the performance, combustion and emissions of a single cylinder direct injection spark ignition engine at different engine loads at a constant engine speed. Five intake valve profiles were tested at 1500rpm. The results are presented and analysed in this paper with main findings stated below:

- Lower valve lift increases intake manifold pressure when throttle is used
- The discharge coefficient may influence PMEP when the difference in intake pressure is not significant, not more than 0.05bar
- Lower ECR allows to advance spark timing when combustion is knock limited
- Lower valve lift reduces flame speed more than low ECR, thus combustion duration is longer (by 15% on average across the load cases between EIVC84, EIVC64 and LIVC)
- Valve profiles affect in-cylinder lambda even though exhaust lambda is kept at 1 (mixture in the cylinder is not always uniformly mixed)
- EIVC with short lifts and LIVC profiles were successful in reduction of fuel consumption up to 2.5% compared to the baseline profile at 9bar net IMEP
- LIVC reduced NOx emissions at all load cases but increased CO emissions
- Combination of EIVC profiles lowered CO and HC emissions but increased NOx almost at all load cases

In addition to the above results, studies were also carried out on the impact of operating one intake valve with different valve lift profiles and results will be published in separate papers.

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