VARIABLE VALVE TIMING EFFECTS ON COMBUSTION, PERFORMANCE AND EMISSIONS IN A SPARK IGNITION ENGINE FUELED WITH ETHANOL IN PART LOAD

Diego Golke
Universidade Federal do Rio Grande do Sul

Marcelo Rohrig
Geovane Alberto Frizzo Prante
Thompson Diórdinis Metzka Lanzanova
Mario Eduardo Santos Martins
Universidade Federal de Santa Maria

ABSTRACT

The stricter legislation on pollutant emissions and the demand for less CO2 per km has been driven the industry on a pursuit trade off in internal combustion engines (ICE). Urban area frequently stands for engine part load operation. For this reason, high engine performance on this map region is essential to mitigate greenhouse gas emission and improve urban air quality. However, spark ignition engine efficiency is worsened by load control method in part load that increase the pumping loses. Therefore, the present work investigated the effects on combustion, performance and emissions by varying the intake and exhaust valve timings. The tests were carried out in a three-cylinder 1.0 l engine with direct injection system. The engine was fueled with hydrous ethanol and the tests were performed at 4 and 8 bar IMEP, 1500 rpm and stoichiometric lambda. The valve timing setting was from a locked position of 5 CAD of NVO to 85 CAD of PVO. The spark timing was adjusted to maintain the CA50 at 10 CAD aTDCf. The injection timing and pressure were kept at 30 CAD aTDCi and 100 bar. The COV_{IMEP} was limited at 3%. The results showed slightly increase in engine efficiency and decrease of NOX emission with higher positive valve overlap.

INTRODUCTION

The global energy demand is expected to increase about 20% until 2040, being 25% from the transportation sector [1]. Efforts to increase the efficiency of ICE are a constant struggle from the automotive industry. The reduction in the GHG emissions, such as carbon dioxide (CO2), is direct related to increase in energy efficiency. In this path, the utilization of battery electric vehicles (BEV) instead of ICE is an environmentally friendly alternative suggested by some authors. However, a life cycle analysis indicates that efficient ICE could offer a larger reduction of emissions than BEV in markets with a highly carbon based electricity supply [2] [3]. Another emerging alternative is the hybrid electric vehicle (HEV), which has an great potential to reduce GHG emissions [4]. So, abruptly stop ICE production without a properly structure for electrification could lead not only to environmental, but either economic and social problems [5]. Therefore, the continuous improvements of ICE's efficiency are necessary to reduce emissions and overcome the regulatory legislations.

Technological advances on the pursue of more efficient ICE have being developed. Downsizing is a widespread strategy employed in ICE to reduce energy losses. The reduced engine size and consequently reduced volume displacement reduces the friction losses [6], and decreases pumping work as the engine must operate with a higher load more frequently [7]. In addition, downsized engines are normally equipped with direct injection (DI) system, which makes possible the operation in homogeneous and stratified combustion as well, and furthermore, allows the advantage of the cooling effect.

Conventional spark ignition engines with fixed valve actuation have several losses due to pumping work [8]. Improvements in the gas exchange process with advanced valve strategies are largely study and several systems have been developed for valve actuation [9] [10] [11]. One of the first system created was VTEC by Honda [12]. Basically, the system consists in two different cam profile for specific speed and load operation, that enables partial de-throttling and then reducing pumping losses. Other systems, such as continuously variable valve lift (CVVL) [13], provides more valve operation possibilities, allowing valve timing and lift freely actuation. A more sophisticated mechanism is the camless system with fully variable valve actuation (FVVA), with independent valve lift and timing actuation [14]. A common system found in many vehicles is the variable cam timing (VCT) mechanism. An example is the twin independent variable camshaft timing (Ti-VCT) system applied by Ford in its EcoBoost engines. With electro-mechanic actuators, the intake and exhaust valves opening and closing timings are continuously adjusted by varying the camshafts positions [15].

Incentives to usage of biofuels, such as governmental program as Renovabio [16], gives ethanol great potential in the Brazilian market. Hydrous ethanol has reduced production cost due to lesser energy required in the distillation process. This results in economic advantages when compared to anhydrous ethanol. [17]. Compared to gasoline has higher octane number and greater latent heat of vaporization, which lead to knock resistance and charge cooling effect, respectively [18]. These characteristics are suitable with DI turbocharged engine, which enables increased compression ratio [19]. However, the cooling effect increases the specific gravity of air which increases the pumping work due reduced pressure in the intake process. Still, the compression work is decreased by lower temperature and pressure [20]. Diverse studies showed the potential of hydrous ethanol in performance and emissions[21,22]. Another characteristic that gives ethanol a great potential is the higher ratio of specific heats than gasoline [23], and also produce more triatomic molecules, which increases the heat capacity, reducing the temperature of combustion and decreasing heat losses [24]. In addition, ethanol laminar flame speed is higher than gasoline, this reduces the combustion duration, which decreases heat losses [25]. However, the lower heat value (LHV) of ethanol increasing fuel consumption and reduces the tank mileage of a vehicle [26].

Lanzanova *et al.* [27] investigated the effects of residual gas trapping with a negative valve overlap (NVO) strategy, in a single cylinder spark ignition DI engine. Anhydrous ethanol was compared with hydrous ethanol in different compositions, and RON 95 gasoline. The experiments were performed at 1500 rpm and 3.1 bar IMEP with stoichiometric air/fuel ratio. A computational model was used to estimate the levels of residual gas fraction (RGF) at each operation point. The results showed increased indicated efficiency with high levels of RGF. Stable combustion was achieved with 40% of residual gas fraction and ethanol with 20% v/v of water. Reduction in NOx was more perceptive with water dilution than RGF.

This work shows an investigation of the effects of the intake and exhaust valve timing events variation on combustion, performance and emissions. The tests were carried out in a three-cylinder 1.0 l engine with direct injection system. The engine was fueled with hydrous ethanol, and the tests were performed at 4 and 8 bar IMEP, 1500 rpm and stoichiometric air/fuel ratio. The valve timing setting was from a locked position of 5 CAD of NVO to 85 CAD of PVO. The spark timing was adjusted to maintain the CA50 at 10 CAD aTDCf. The injection timing and pressure were kept at 30 CAD aTDCi and 100 bar. The COV_{IMEP} was limited at 3%.

EXPERIMENTAL SETUP

The experiments were performed using an in line three cylinders DI turbocharged Ford EcoBoost 1.0 liters engine. The engine is equipped with four valves per cylinder and double overhead camshaft (DOHC), with a twin independent variable cam timing (Ti-VCT) system. Engine specifications are shown in Table 1.

Table 1: Engine specifications

Parameter	Description
Displaced Volume	999 cc
Number of Cylinders	3
Stroke	82 mm
Bore	71.9 mm
Connecting Rod	137 mm
Compression Ratio	10:1
Number of Valves per Cylinder	4
Exhaust Valve Opening (EVO)	44° BBDC @ 0.5 mm lift
Exhaust Valve Closure (EVC)	0° ATDC @ 0.5 mm lift
Inlet Valve Opening (IVO)	5° ATDC @ 0.4 mm lift
Inlet Valve Closure (IVC)	53° ABDC @ 0.5 mm lift
Injection	High pressure direct injection with 6 holes solenoid injectors
Combustion Chamber Type	Pent-roof

For the acquisition of intake and exhaust pressure were used a MPX4250 and an AVL APT100 pressure transducers, respectively. The intake temperature was estimated using an AVL PT100 sensor. Temperatures of the exhaust gas, before and after the turbine, oil reservoir and cooling system were monitored with a K type thermocouple. Atmospheric air pressure, temperature and humidity were measured utilizing a HMT330 unit from VAISALA. A programmable engine control unit (ECU) Bosch MS6.3 was used to adjust engine operating parameters, such as angle of start of injection, angle of ignition, valve phase timing and others. The monitoring of air-fuel mixture was carried out using a Bosch LSU 4.9 lambda probe.

For the acquisition and processing of the in-cylinder pressure signal were used an AVL GH14D piezoelectric pressure transducer and the indicated system AVL IndiMicro 602. The online analysis of combustion parameters was performed by AVL Indicom user interface software. The engine load was applied through an AVL DynoPerform 240 dynamometer. Fuel consumption was obtained using an AVL 7130 fuel balance with principle based on gravimetric measurement. The combustion products were measured with the AVL Sesam FTIR i60 emission analyser. The AVL PUMA Open system was used to control the test cell and simultaneous data acquisition. The experimental setup scheme is shown in Figure 1.

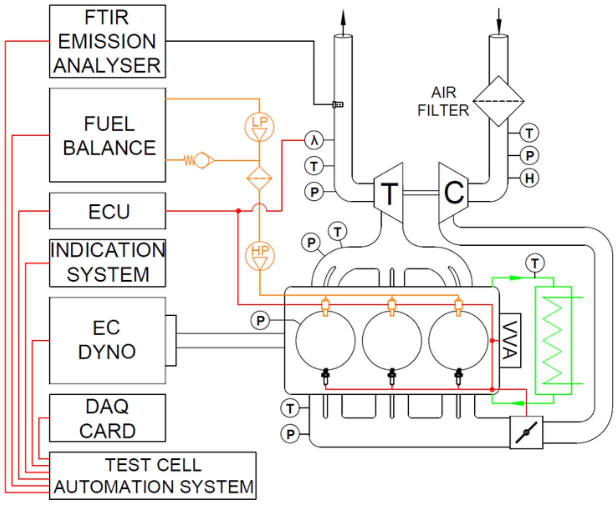

Figure 1: Experimental setup scheme

TEST PROCEDURE

The engine was warmed up until the cooling water and the oil temperature reached 364±2 K and maintained at these temperatures. The engine was fueled with hydrous ethanol E93W07 (7% v/v water content) and operated with stoichiometric air/fuel ratio. The speed of the engine was fixed at 1500 rpm and the spark advance was adjusted to maintain the CA50 in 10 CAD ATDC firing. The injection timing and pressure was kept in 330 CAD before TDC firing and 100 bar, respectively. Combustion variability was limited by COV of IMEP with 3% as a maximum.

At 1500 rpm the maximum brake mean effective pressure (BMEP) reached by the engine is 16 bar [28]. So, for this work the maximum partial load operation was considered as 8 bar IMEP. The tests were carried out at 4 and 8 bar IMEP. The valve timing setting was from a locked position of 5 CAD of NVO to 85 CAD of PVO

RESULTS

The combustion duration is depicted in Figure 2 for different valve timing at intake top dead center. Shorter combustion duration was observed for negative valve overlap and small positive valve overlap. As the PVO increase, the combustion duration is higher. The main reason is the greater fraction of residual gas as the amount of air and fuel remain constant to maintain same engine load. The residual gas impairs the combustion process, reducing the reaction rate and increasing the combustion duration. Basically, during the PVO, the combustion products goes back to intake port due the lower intake pressure in part load operation. Thus, the higher PVO duration means greater flow back from exhaust port to cylinder and intake port, increasing the residuals gas mass. After the exhaust valve closure, the residual gas flows through the cylinder due to the downward movement of the piston.

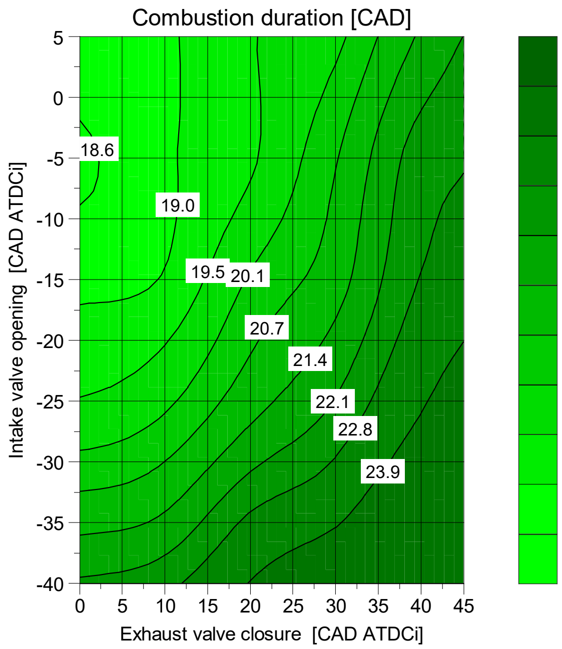

Figure 2: Combustion duration as a function of intake valve opening and exhaust valve closure for 8 bar IMEP

Lower engine load requires less air and fuel, resulting in lower heat energy released from the combustion. As a result, the in-cylinder pressure and temperature during combustion are lower than in case of higher engine load. In this way, the reaction rate is expected be lower at 4 bar IMEP than at 8 bar IMEP, increasing the combustion duration as shown in Figure 3. Higher combustion duration was found for longer PVO duration. However, higher combustion duration results in higher combustion instabilities, limiting longer PVO duration for 4 bar IMEP.

Ignition delay for 8 bar IMEP is shown in Figure 4 as a function of intake vale opening and exhaust valve closure. The increase in residual gas content further increase the ignition delay. The early stage of flame development has a significant effect on combustion stability. In addition, higher ignition delay means advanced spark timing, which can result in greater negative compression work.

The residual gas fraction has a greater effect on ignition delay than on combustion duration. The increase in ignition delay was around 3 times higher and the combustion duration was 2 times higher in 4 bar IMEP as seen in Figure 5 and Figure 3. The higher residual gas fraction in 4 bar IMEP caused considerably increase in ignition delay that deteriorate the combustion stability.

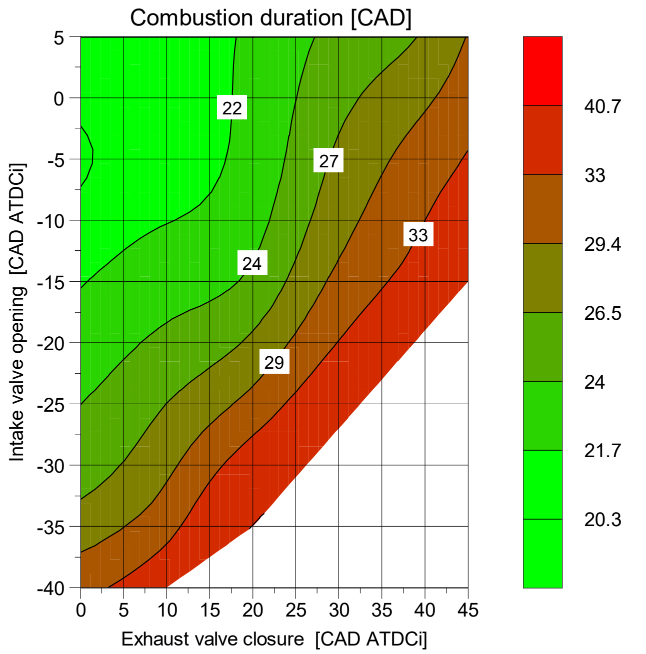

Figure 3: Combustion duration as a function of intake valve opening and exhaust valve closure for 4 bar IMEP

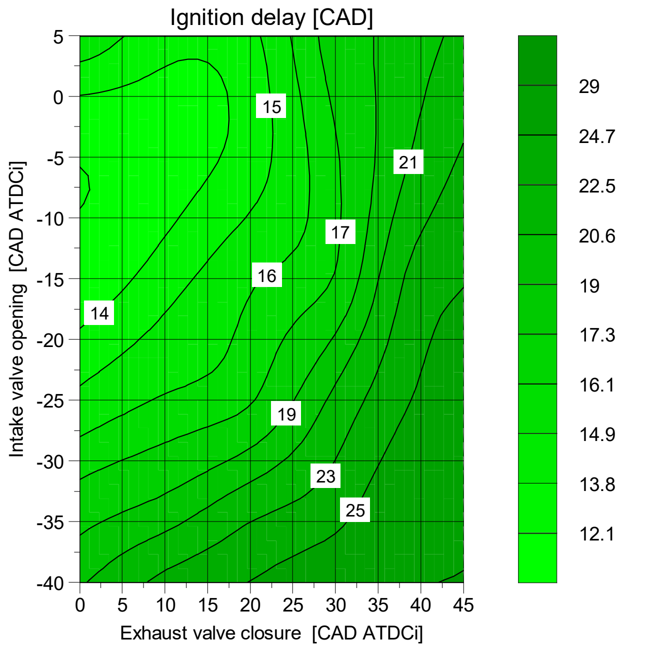

Figure 4: Ignition delay as a function of intake valve opening and exhaust valve closure for 8 bar IMEP

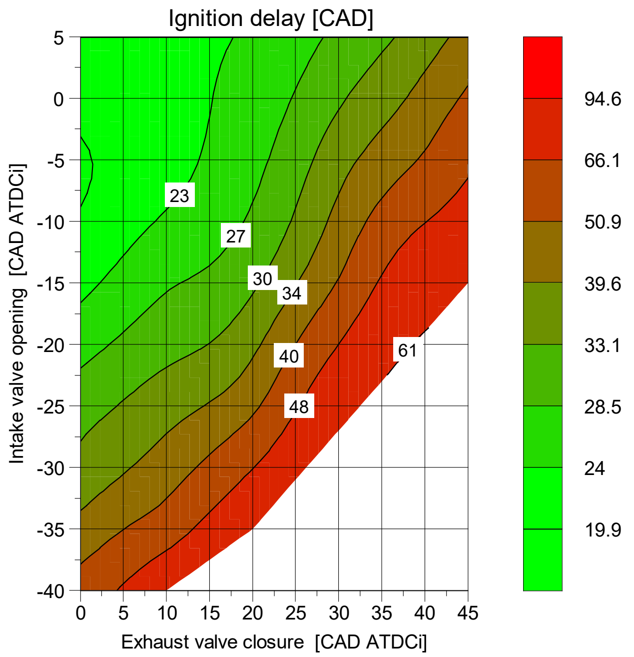

Figure 5: Ignition delay as a function of intake valve opening and exhaust valve closure for 4 bar IMEP

The Figure 6 shows the combustion instability measure through the COV of IMEP for 8 bar IMEP. The experimental tested condition resulted in low values, showing potential of more residual gas dilution to increase de-throttling operation. However, all VVT range of the engine was covered, requiring others way to further increase residual gas dilution.

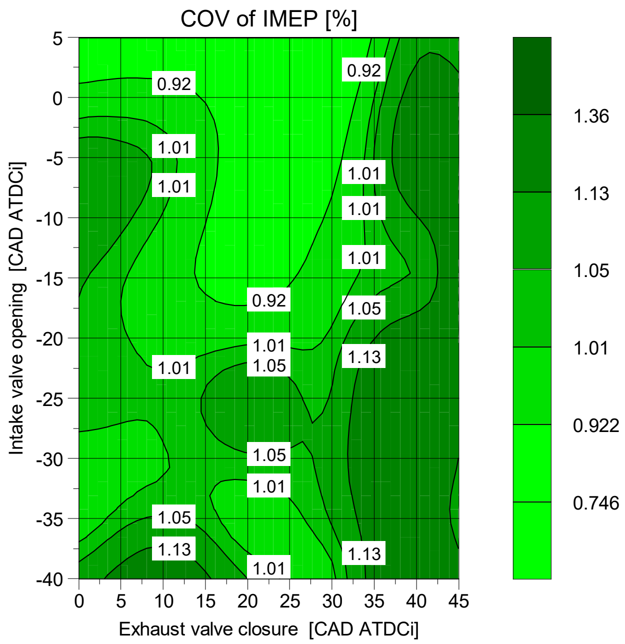

Figure 6: COV of IMEP as a function of intake valve opening and exhaust valve closure for 8 bar IMEP

Engine instability limit was achieved in 4 bar IMEP as depicted in Figure 7 for certain duration of PVO. Lower values were found with NVO or short PVO duration and high values to longer PVO durations. As the residual gas fraction increase, the combustion process is impaired and the combustion instability increase.

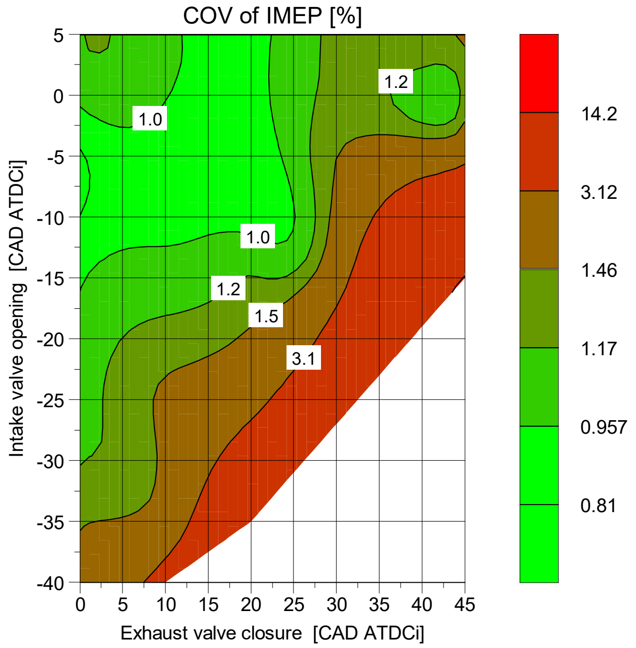

Figure 7: COV of IMEP as a function of intake valve opening and exhaust valve closure for 4 bar IMEP

The throttle opening increase for higher engine loads, increasing the intake pressure. For this reason, the residual gas that flow back to intake port decrease as the pressure difference between the intake and exhaust decrease. Then, the gas exchange efficiency does not increase considerably, although be possible stable engine operation in all VVT range tested as shown in Figure 8.

On the other hand, low engine load requires less air and then less throttle opening. Thus, the higher pressure difference between intake manifold and exhaust manifold increase counterflow of residual gas. For this reason, the increase in gas exchange efficiency is higher for 4 bar IMEP than in 8 bar IMEP even with PVO duration limited by combustion stability as seen in Figure 9.

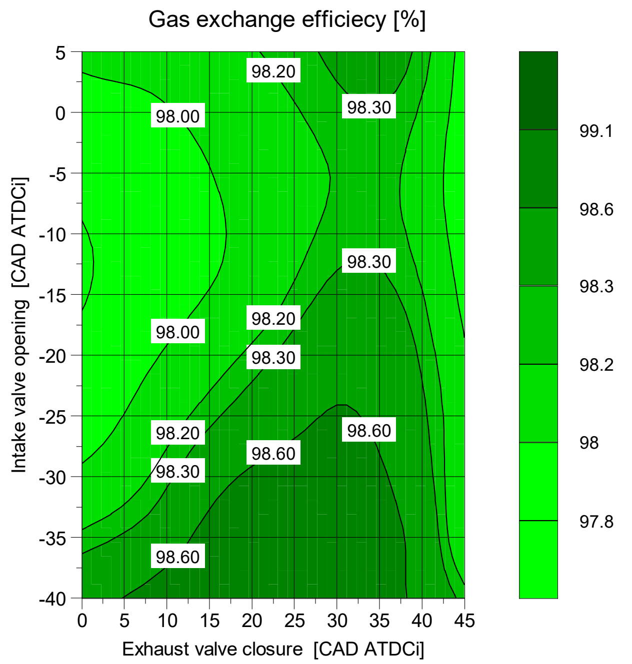

Figure 8: Gas exchange efficiency as a function of intake valve opening and exhaust valve closure for 8 bar IMEP

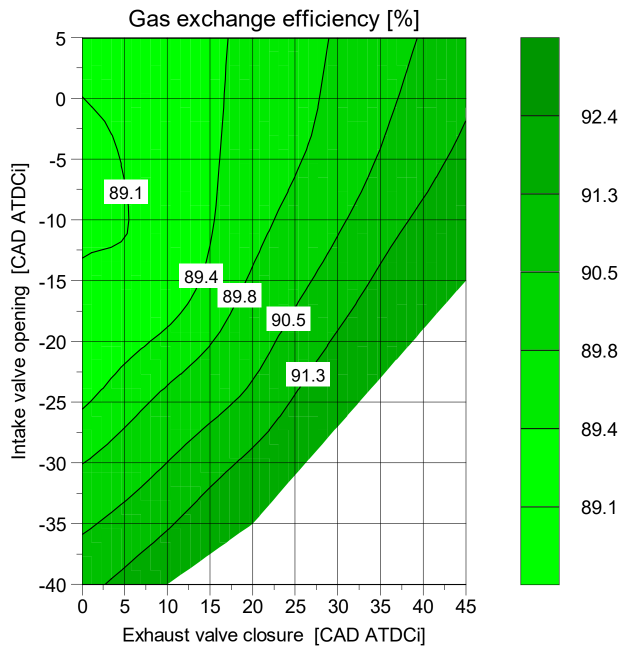

Figure 9: Gas exchange efficiency as a function of intake valve opening and exhaust valve closure for 4 bar IMEP

In Figure 10 is depicted the combustion efficiency as function of the IVO and EVC for 8 bar IMEP. The residual gas fraction not impaired the combustion process, resulting in high values of combustion efficiency in all operation points.

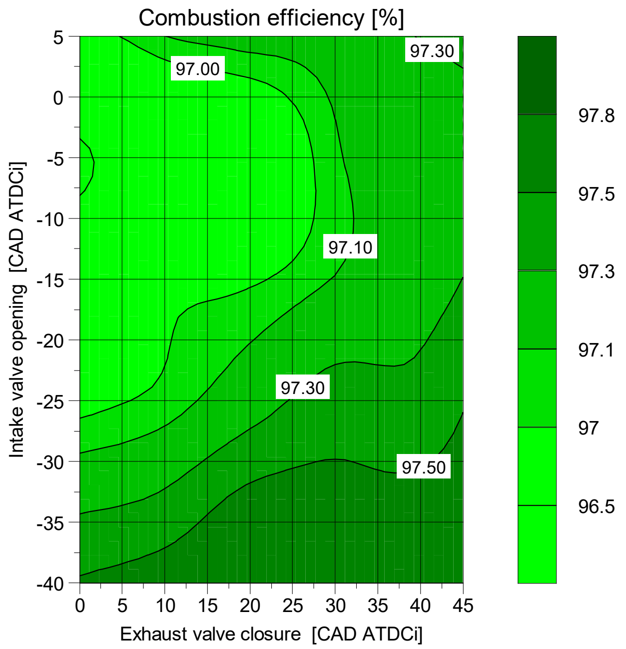

Figure 10: Combustion efficiency as a function of intake valve opening and exhaust valve closure for 8 bar IMEP

Higher residual gas fraction decreases the reaction rate and increase combustion instability as presented in Figure 7 to 4 bar IMEP. In this way, the combustion efficiency, shown in Figure 11, decreases as a result of higher residual gas dilution, which reduces combustion velocity and increase the combustion duration and ignition delay.

The indicated efficiency as a function of IVO and EVC is depicted in Figure 12 for 8 bar IMEP. Higher PVO durations slightly increased the indicated efficiency due small increase in combustion efficiency and gas exchange efficiency.

Although the increase in gas exchange efficiency in 4 bar IMEP, the indicated efficiency shown in Figure 13 does not increased as well. One reason is the decrease of the combustion efficiency with large PVO durations as presented in Figure 11. In addition, the indicated efficiency is reduced further due the increase in heat transfer resulted of the residual gas dilution as reported by [29].

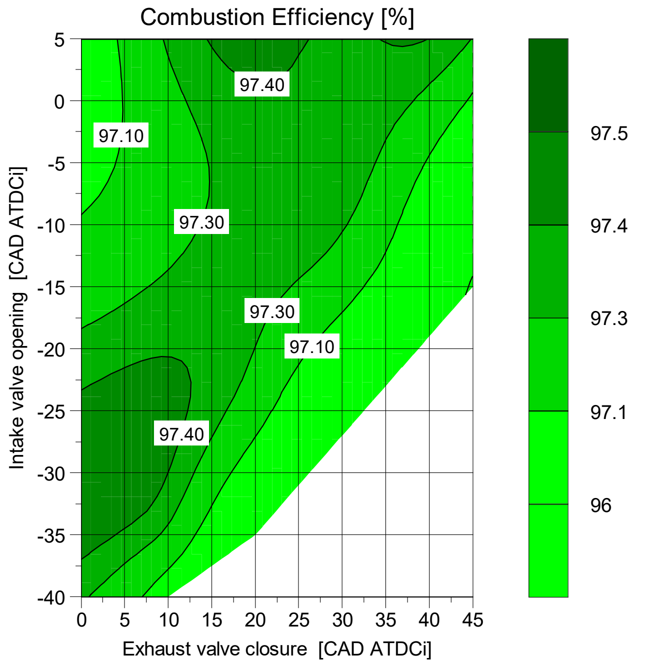

Figure 11: Combustion efficiency as a function of intake valve opening and exhaust valve closure for 4 bar IMEP

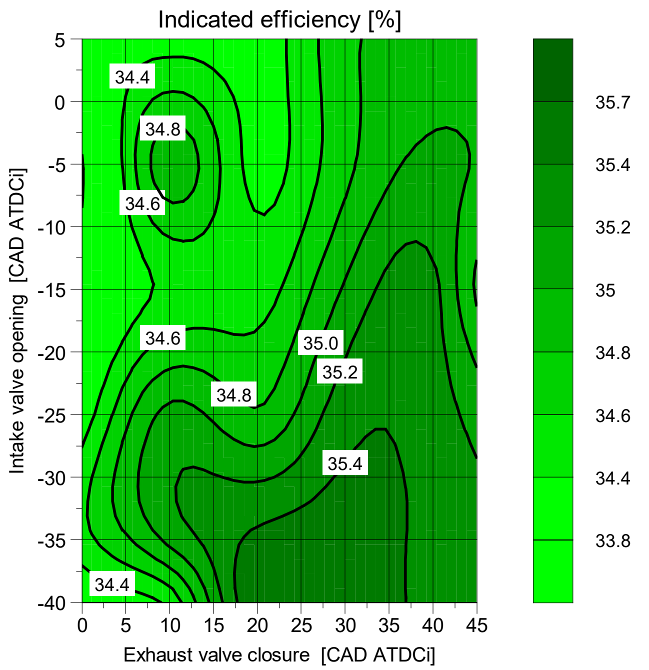

Figure 12: Indicated efficiency as a function of intake valve opening and exhaust valve closure for 8 bar IMEP

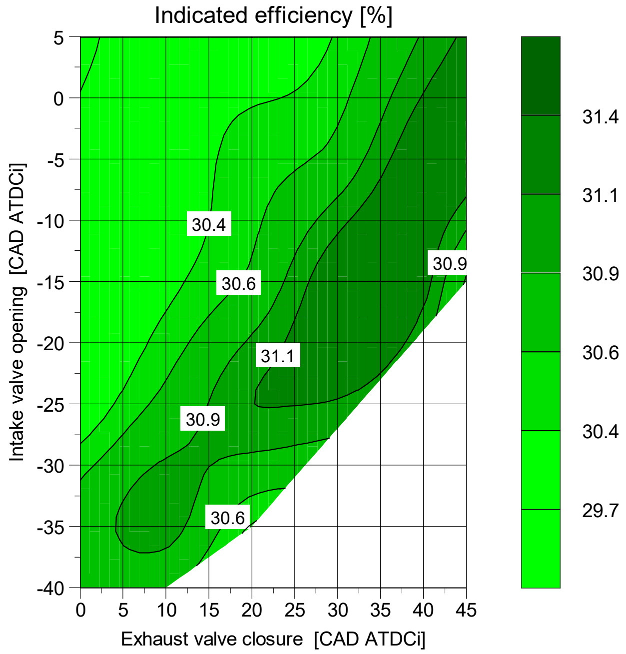

Figure 13: Indicated efficiency as a function of intake valve opening and exhaust valve closure for 4 bar IMEP

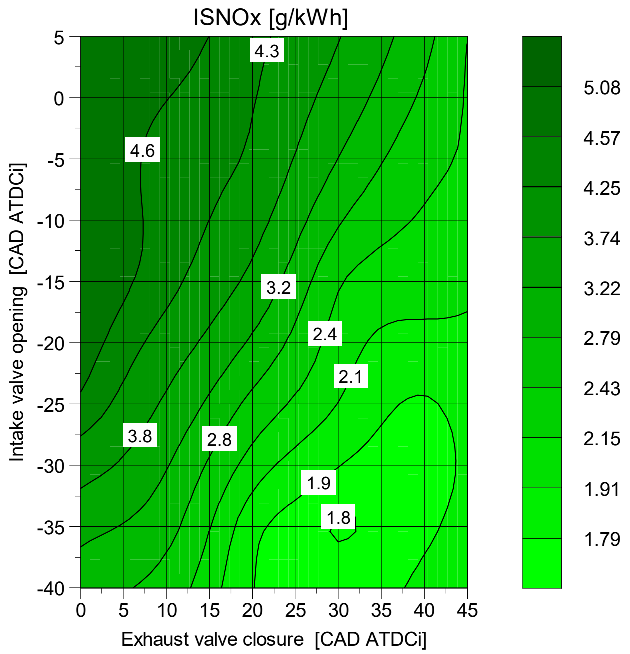

Figure 14: NO_X emissions as a function of intake valve opening and exhaust valve closure for 8 bar IMEP

EMISSIONS ANALYSIS

The NO_X emissions as a function of IVO and EVC for 8 bar IMEP is depicted in Figure 14. Large PVO durations decreased the NO_X. The residual gas decreases the combustion temperature due the combustion products from residual gas increase the heat capacity of the charge. NVO and short PVO duration showed the highest value of NO_X emissions.

Low engine load requires less air and fuel, resulting in lower in-cylinder temperature. Lower levels of NO_X are expected for 4 bar IMEP than 8 bar IMEP. However, the higher residual gas fraction with large PVO duration decreased the NO_X emission further as a resulted of the decreased in-cylinder temperature as depicted in Figure 15.

The Figure 16 shows the CO emissions as a function of variable valve timing for 8 bar IMEP. Despite the combustion efficiency was considerable constant as show in Figure 10, the CO emissions levels decreased with large PVO duration. It could be related with the increased time for oxidation since part of the combustion products resulted from combustion is readmitted due the flow characteristic during the PVO period.

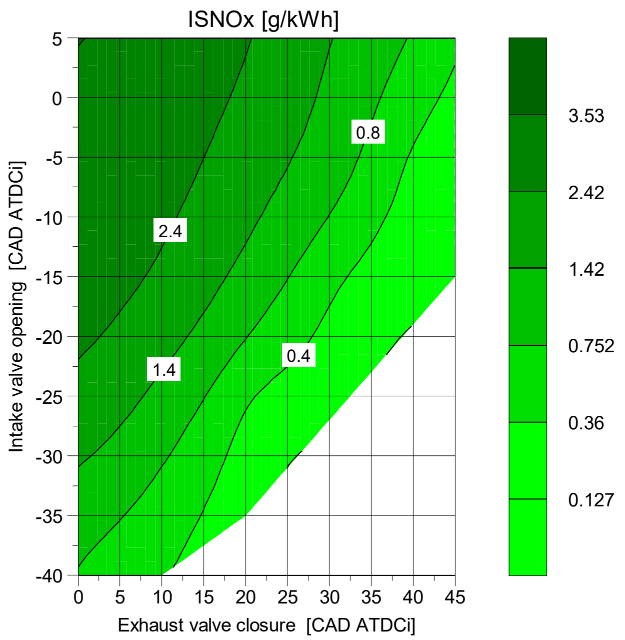

Figure 15: NO_X emissions as a function of intake valve opening and exhaust valve closure for 4 bar IMEP

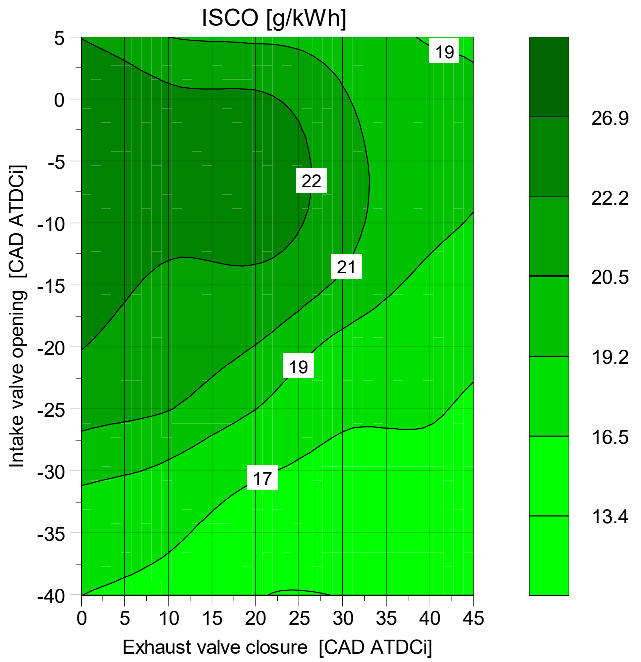

Figure 16: CO emissions as a function of intake valve opening and exhaust valve closure for 8 bar IMEP

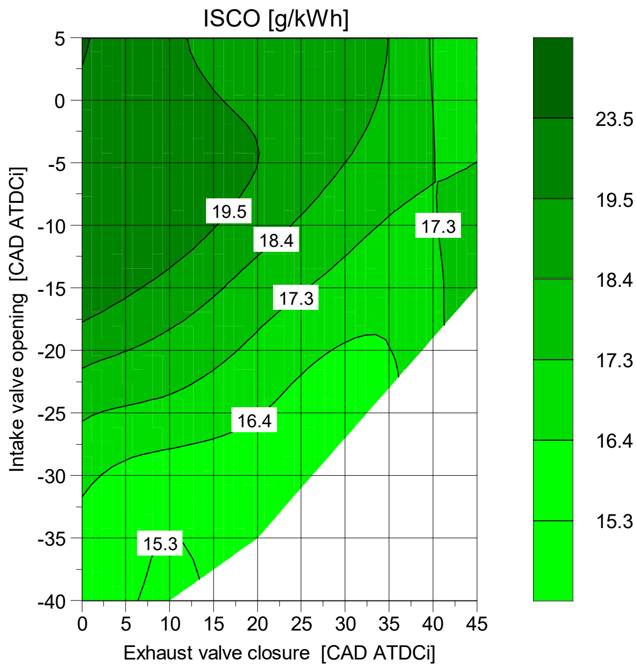

Figure 17: CO emissions as a function of intake valve opening and exhaust valve closure for 4 bar IMEP

CO emissions as a function of IVO and EVC for 4 bar IMEP is depicted in Figure 17. Similar results as discussed before was found. Large PVO durations decreased the CO emissions, since more combustion products mass are readmitted with air and fuel for the next combustion cycle. In this way, there is more time for oxidation process of CO molecules.

The indicated specific organic gas (ISOG) considers the emissions of hydrocarbons, aldehydes and alcohols. The Figure 18 shows the ISOG as a function of IVO and EVC for 8 bar IMEP. Similar results were found for all tested conditions.

In 4 bar IMEP, the ISOG increased for large PVO duration as shown in Figure 19. It could be resulted of the higher combustion instability and the lower combustion efficiency as shown in Figure 7 and Figure 11 respectively.

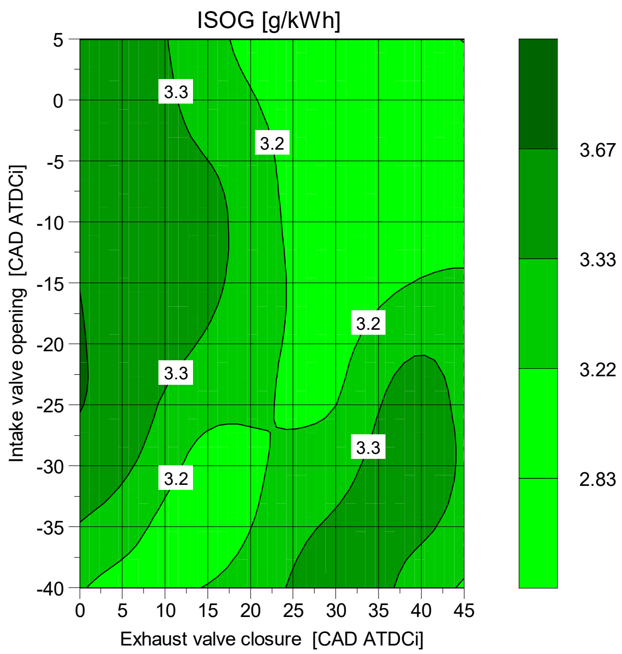

Figure 18: ISOG emissions as a function of intake valve opening and exhaust valve closure for 8 bar IMEP

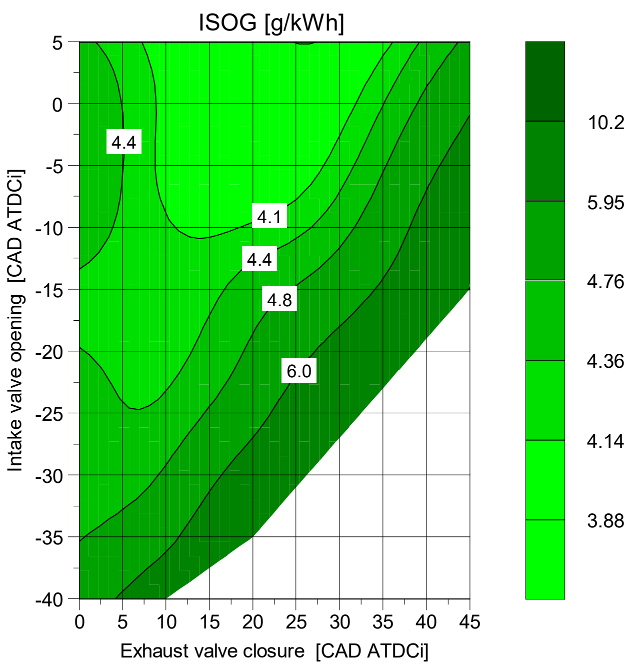

Figure 19: ISOG emissions as a function of intake valve opening and exhaust valve closure for 4 bar IMEP

CONCLUSIONS

This research investigated the variable valve timing effects in part load of a three cylinders direct injection engine fueled with hydrous ethanol. The valve timing setting was from a locked position of 5 CAD of NVO to 85 CAD of PVO in 2 different engine loads and CA50 fixed in 10 CAD ATDCf. The following conclusions could be drawn:

- The combustion duration and ignition delay increase with higher residual gas fraction for 4 and 8 bar IMEP.
- Lower engine loads shown higher combustion instabilities due higher residual gas fraction.
- The gas exchange efficiency increases due de-throttling required to maintain same engine load.
- The indicated efficiency is impaired by decreased combustion efficiency and increased heat transfer.
- Decreased combustion efficiency and increased heat transfer impair the indicated efficiency.
- NOx emissions decrease with large PVO durations due lower in-cylinder temperature resulted from higher heat capacity of the charge.
- CO emissions decreases with higher PVO durations due part of combustion product be readmitted to the next combustion cycle.
- IOSG emissions have not changed significantly.

ACKNOWLEDGEMENTS

The authors would like to thank the Federal University of Santa Maria, Federal University of Rio Grande do Sul, the Coordination for the Improvement of Higher Education Personnel (CAPES) and Research Support Foundation of the State of Rio Grande do Sul (FAPERGS) for the financial support of this project.

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