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BIASING A STAGED FUEL INJECTION SYSTEM OF A SINGLE CYLINDER FOUR STROKE GASOLINE ENGINE

Karlis Banis
Latvia University of Life Sciences and Technologies, Latvia
k.banis@yahoo.com

Abstract
This paper investigates the effect of fuel bias between the primary and secondary injectors of a staged fuel injection system on the performance of a high output single cylinder spark-ignited internal combustion engine. It is known that staged fuel injection systems are widely used in motorsports applications where high engine speeds are coupled with high power output, therefore, the aim of this study is to evaluate the effect of a secondary fuel injector installed on a Honda CRF450R single cylinder four-stroke gasoline engine. The said engine was equipped with a programmable Performance Electronics PE3-SP0 control unit and a secondary fuel injector identical to that of OE. Power measurements were carried out on a Dynojet-200ix chassis dynamometer in four different modes with altered fuel proportion between injectors, with each measurement being repeated three times. Ambient conditions were monitored with Performance Electronics Pe3Monitor software and the fuel map was adjusted to produce a stable air-fuel ratio. The results were averaged and compared numerically and by coefficient of correlation. It was observed that the data as obtained from the chassis dynamometer software SportDyno 4 contains a lot of noise, both mechanical and electrical in nature, and the changes in power output are highly dependent on engine and equipment temperature. The best results were obtained by using both injectors with fuel proportion biased to the front of the system.
Key words: electronic fuel injection system, port injection, volumetric efficiency, chassis dynamometer.

Introduction

Since the early experiments with electronic fuel injection on gasoline engines as means of improving the fuel efficiency, emissions, and power output, it has been observed that said system poses several important advantages over its predecessors, for instance, fuel efficiency. Sophisticated fuel injector design allows the fuel to be metered only by the injection pulse width (Knapp & Lembke, 1985) leading to more accurate dosing. It was very early understood that fuel injection must be carried out under low pressure to promote evaporation. Unfortunately, when injected at such conditions, the fuel breaks up not only into vapor but also liquid fuel droplets of which some are deposited on the walls and create a fuel film (Almkvist & Eriksson, 1993). However, velocity in the induction system has a significant effect on the transport and vaporization of fuel spray, as well as on the evaporation of wall deposits (Nagaishi *et al*., 1989). It was then found that fuel vaporization can be accelerated by breaking the flow into very fine droplets (Zhao, Lai, & Harrington, 1995). For this reason, high pressure injection was developed, where it was also found that high pressure fuel spray producing fine droplets minimizes soot emissions (Karl *et al*., 1997). Although a lot of research was done on GDI (Gasoline Direct Injection) where the fuel is injected directly into the combustion chamber, later developments of HCCI (Homogeneous Charge Compression Ignition) systems found that port injection is highly preferable in applications requiring different fuel atomization strategies combined with low production cost (Cao *et al*., 2005). Around the same time a different port injection strategy was tested mainly in motorsports – staged injection, consisting of a primary (downstream) and secondary (upstream) injector. On single cylinder or individual throttle body engines the secondary injector would most often be placed upstream of the throttle valve where earlier studies show that droplet size is the determining factor in their path around it – larger droplets will tend to deposit on the throttle valve surface while smaller droplets will tend to follow the air stream around it (Nogi *et al*., 1988). The secondary injector would only be deployed at engine speeds and loads with sufficiently high intake velocity under high injection pressure as to minimize the likelihood of fuel deposition. It was discovered that in order to burn the deposited fuel film, it must be vaporized by heat conduction from the walls (Hendricks *et al*., 1993). Advantage could be taken from what little deposition remained as it cooled the intake runner walls while evaporating. Thus, the injected fuel itself increases the density of mixture entering the cylinder leading to higher volumetric efficiency (Sarkar, Manivannan, & Ramesh, 2003). These advantages are mostly employed in motorsports. However, knowing that motorsports is the area of technical break-through where the limits of our technology are pushed further, some examples have been known to exist among road use motorcycles. The aim of this study is to evaluate the effect of a secondary fuel injector installed on the single cylinder four-stroke gasoline engine of a Honda CRF450R off-road motorcycle.

Materials and Methods

The object of investigation is the electronic fuel injection system of a Honda CRF450R motorcycle equipped with a secondary fuel injector identical to that of OE. The location of the secondary injector was chosen 76 mm upstream of the primary (OE) injector (Figure 1) due to packaging restraints. The power measurements were carried out in 2018, on DynoJet 200-ix eddy-current chassis dynamometer in Riga. The technical parameters of the used engine and dynamometer are given in Table 1. Four different fuel injection modes were chosen for comparison (Table 2) where the total amount of fuel injected remains unchanged, but the proportion of it is biased between the primary and secondary injectors where the maximum allowed fuel bias as dictated by the engine control unit is 65% towards either of the injectors. The fuel is supplied via electric pump with a constant pressure of approximately 3.5 – 4 bar. The amount of injected fuel is defined as the injector open time in milliseconds during each injection cycle. In order to evaluate the repeatability of the power measurements, the power delivery in each mode was measured three consecutive times resulting in a total of 12 measurements.

The Performance Electronics PE3-SP0 engine control unit is also programmed to multiply the injector open time values in fuel map with coefficients based on water and intake air temperatures. Since both of these parameters directly influence not only the amount of injected fuel but also the development of intake charge mixture, which in turn directly influences the

Table 1

Technical parameters

Honda CRF450R engine	Number of cylinders	1
	Capacity	449 cm^3
	Bore	96.0 mm
	Stroke	62.1 mm
	Fuel delivery	Gasoline EFI, 46mm throttle body
	Compression ratio	13.8:1
	Valvetrain	SOHC, four valves per cylinder
	Cooling strategy	Liquid cooled
DynoJet 200-ix dynamometer	Type	Chassis dynamometer
	Absorber	Eddy current
	Maximum power rating	750 hp
	Maximum speed	320 km h^{-1}
	Maximum wheelbase	2134 mm
	Sensors	Air temp., humidity, barometer, lambda

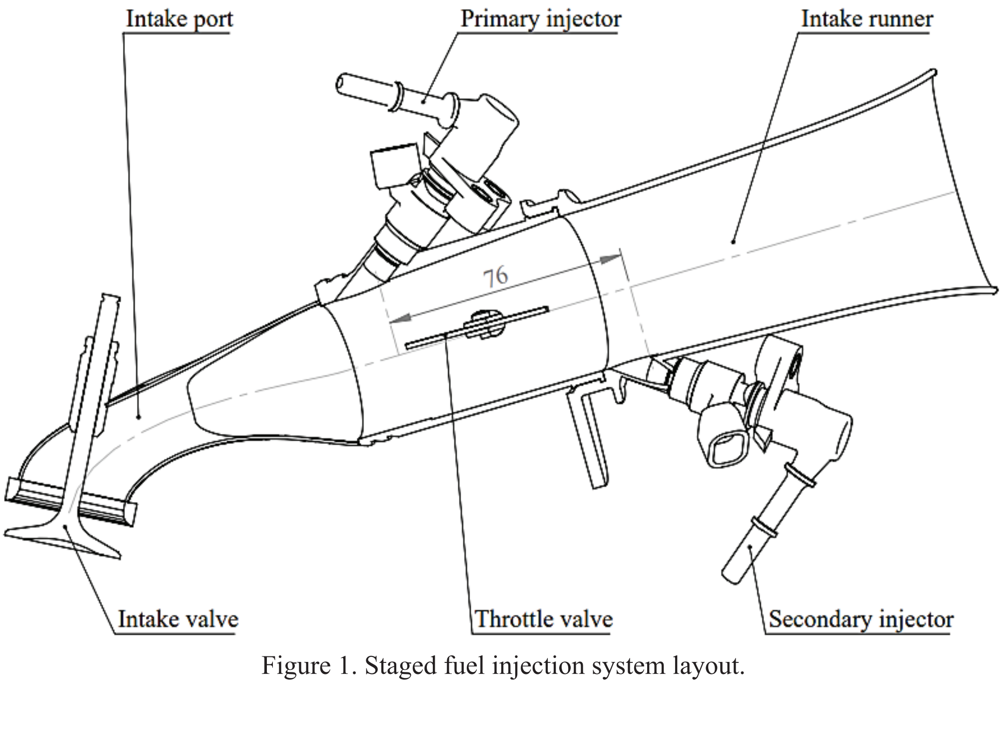

Figure 1. Staged fuel injection system layout.

Table 2

Injected fuel bias as tested in each mode

Configuration	Injected fuel bias	
	Primary injector	Secondary injector
Baseline	100%	0%
Mode 1	65%	35%
Mode 2	50%	50%
Mode 3	35%	65%

Table 3

Controlled parameters

Parameters	Range of values
Engine speed range	4,500 – 10,000 min^{-1}
Engine rotational acceleration	340 – 360 min^{-1} sec^{-1}
Intake air temperature	20 – 21 °C
Water temperature	80 – 82 °C
Air-fuel ratio (AFR)	12.5 – 13.5
Tire pressure	2.0 – 2.1 bar

power output, it is essential to control them during the measurements. This is done by connecting the engine control unit with a laptop via ethernet cable where the temperature values as read from the built-in sensors on the engine are displayed on the online screen of Pe3Monitor software. All the controlled parameters of the experiment are listed in Table 3.

The data is exported from SportDyno 4 software as mechanical power (hp) reading at a given engine speed in revolutions per minute. Since the DynoJet 200-ix dynamometer and the internal combustion engine themselves are prone to repeatability errors associated with noise, both mechanical and electrical in nature (vibration, static electricity, grounding, etc.), the exported data is very unstable and is not usable without pre-processing i.e., the data points are not aligned and therefore cannot be compared on a single horizontal axis. In order to align the data points, formula (1) is used to round the engine speed values v_i to the nearest integer V_i with the interval 100 min^{-1}.

$$V_i = 100\lfloor\frac{v_i}{100}\rceil \quad (1)$$

Average power P_j at engine speed V_j is further calculated from the measured power P_i data points corresponding with equal engine speeds V_i according to equation (2), where the number of identical engine speeds V_i after rounding according to equation (1) is denoted by n.

$$P_j(V_j) = \frac{1}{n}\sum P_i(V_i) \quad (2)$$

The torque T_j can then be back-calculated in Newton-meters from mechanical power P_j using equation (3).

$$T_j(V_j) = \frac{P_j(V_j)\cdot V_j}{7120.756} \quad (3)$$

Results and Discussion

The results calculated after averaging the data against the nearest engine speed integers according to equations (1 and 2) are given in Table 4. Coefficient of correlation is then used to evaluate the agreement between the repeated measurements in each mode. The resulting coefficients of correlation are shown in Table 5.

It is visible that the agreement of data or repeatability in each mode is noticeably higher between two specific runs, meaning that one of the three consecutive runs or measurements tends to deviate from the other two. Such runs are Baseline – Run 1, Mode 1 – Run 2, Mode 2 – Run 1 and Mode 3 – Run 1. This could be explained by variations in coolant and oil temperatures due to the runs being executed in a consecutive manner – one after the other with inconsistent cool-down times where the temperature of the intake system and combustion chamber is influencing the fuel evaporation rate, intake charge density and volumetric efficiency

Table 4

Aligned results

Configuration	Average torque, Nm	Average power, hp
Baseline – Run 1	46.81	46.96
Baseline – Run 2	44.85	45.12
Baseline – Run 3	44.38	44.65
Mode 1 – Run 1	44.96	45.21
Mode 1 – Run 2	44.31	44.60
Mode 1 – Run 3	44.60	44.87
Mode 2 – Run 1	46.01	46.20
Mode 2 – Run 2	44.84	45.09
Mode 2 – Run 3	44.30	44.59
Mode 3 – Run 1	45.36	45.59
Mode 3 – Run 2	44.52	44.81
Mode 3 – Run 3	44.25	44.54

Table 5

Coefficients of correlation

Configuration	Runs 1 & 2	Runs 1 & 3	Runs 2 & 3
Baseline	0.9931	0.9862	0.9972
Mode 1	0.9967	0.9992	0.9980
Mode 2	0.9944	0.9893	0.9981
Mode 3	0.9957	0.9939	0.9976

Table 6

Averaged results

Interval	4,500 – 6,000 min^{-1}		6,100 – 8,000 min^{-1}		8,100 – 10,000 min^{-1}	
Config.	Average torque, Nm	Average power, hp	Average torque, Nm	Average power, hp	Average torque, Nm	Average power, hp
Baseline	39.78 ± 0.29	29.59 ± 0.22	49.55 ± 0.38	49.14 ± 0.38	45.62 ± 0.15	57.74 ± 0.18
Mode 1	39.73 ± 0.34	29.54 ± 0.25	49.72 ± 0.46	49.30 ± 0.45	45.85 ± 0.16	58.03 ± 0.31
Mode 2	39.68 ± 0.29	29.50 ± 0.22	49.33 ± 0.36	48.92 ± 0.36	45.75 ± 0.13	57.89 ± 0.16
Mode 3	39.31 ± 0.20	29.22 ± 0.15	49.16 ± 0.24	48.75 ± 0.23	45.64 ± 0.11	57.77 ± 0.14

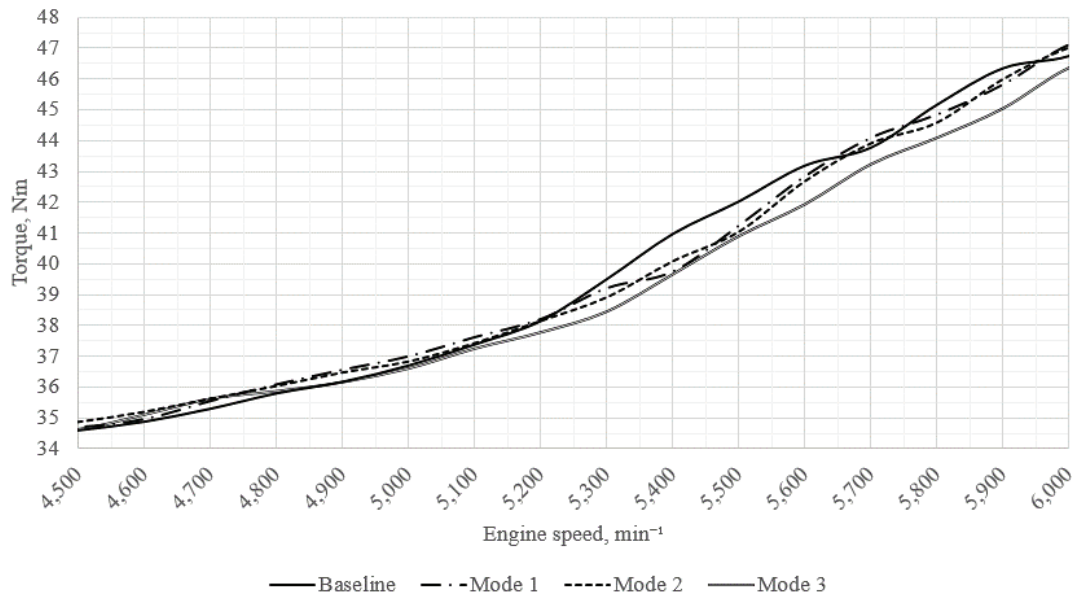

Figure 2. Torque comparison at low engine speeds.

(Sarkar, Manivannan, & Ramesh, 2003). To increase the credibility of this study, the mentioned data sets are discarded. The results are brought to the final summation by averaging the two remaining data sets in each mode. Table 6 shows average torque, split by three engine speed intervals – low speed (4,500 – 6,000 min^{-1}), medium speed (6,100 – 8,000 min^{-1}) and high speed (8,100 – 10,000 min^{-1}). The torque values are expressed as averages between repeated runs supplemented with standard error of mean.

Figure 2 indicates the tendency of increasing low speed torque loss when secondary injector is used (Mode 1, Mode 2 and Mode 3). The highest averaged torque – 39.78 Nm is produced in baseline test, using only the primary injector. This could be attributed to an insufficient intake velocity, leading to increased fuel deposition on the walls of the intake system (Nagaishi *et al*., 1989).

Figure 3 shows slight improvements of medium engine speed torque with the fuel injection biased to the front (Mode 1), meaning that only slightly more fuel can be injected from the secondary injector compared with the baseline (single injector) test until it starts to deposit on the induction system walls without evaporating, as represented by Mode 2 and Mode 3 curves.

Figure 4 shows the tendency of the previously described phenomena becoming less pronounced as the engine speed is increased higher. This could be explained by accelerated evaporation of the fuel deposits due to a higher intake velocity, meaning that at some point the engine speed could be high enough

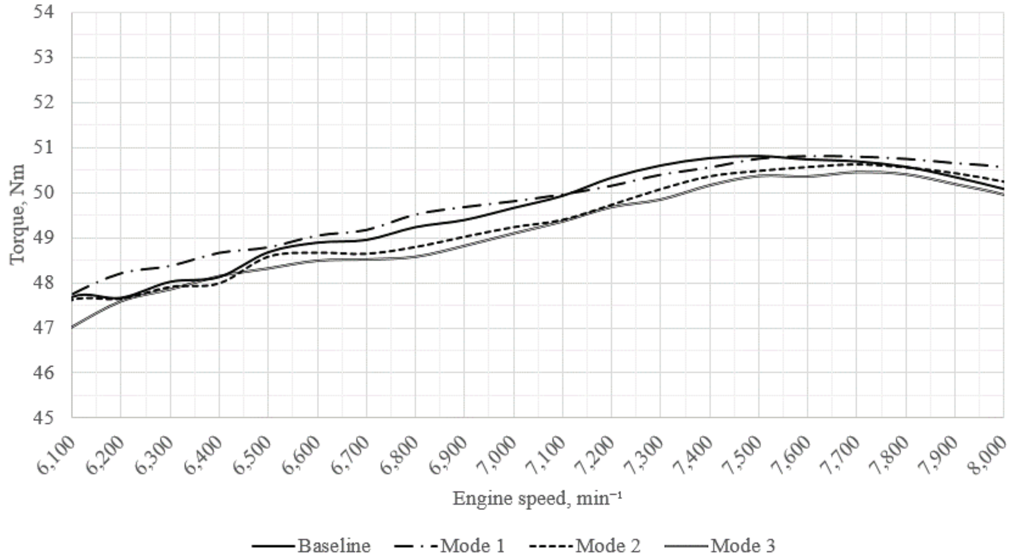

Figure 3. Torque comparison at medium engine speeds.

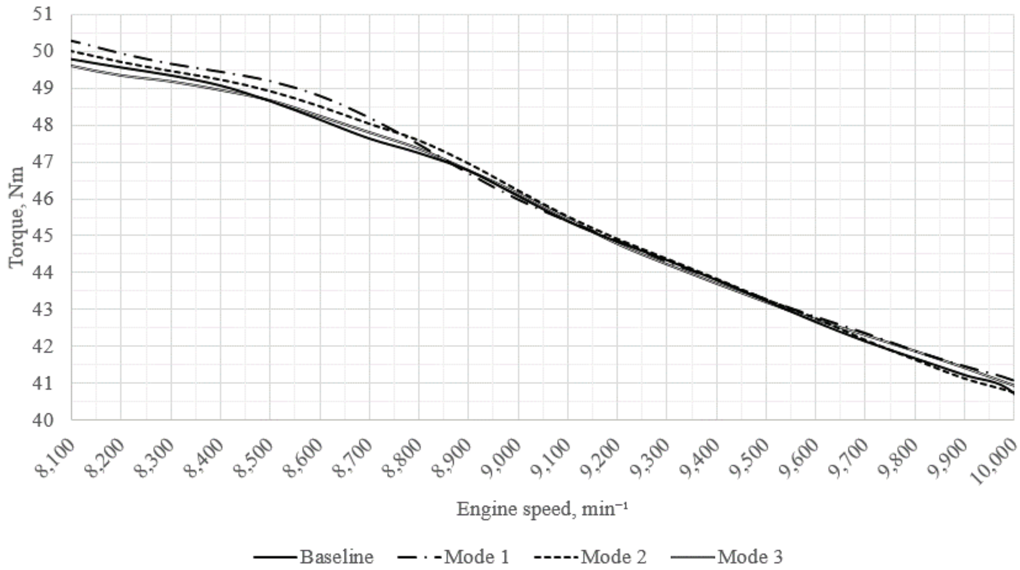

Figure 4. Torque comparison at high engine speeds.

so that biasing the injected fuel more towards the secondary injector becomes beneficial. In the case of the engine used in this study, the said point falls off the operating speed range.

Conclusions

1. Staged fuel injection systems in gasoline engines are used when high output and high engine speeds are required, where the aim of the secondary fuel injector is to aid in cooling the intake runner, while increasing the volumetric efficiency and the rate of fuel evaporation. The possible benefits in this case are lower fuel consumption, lower emissions and higher efficiency.
2. In the engine speed interval 4,500 – 6,000 min^{-1}, the best results (39.78 Nm, 29.59 hp) were achieved by using only the primary injector. At medium engine speed between 6,100 and 8,000 min^{-1}, the best results – 49.72 Nm (+0.34%) and 49.30 hp (+0.33%) were achieved by fuel bias 65% on the primary and 35% on the secondary injector. At high engine speeds from 8,100 to 10,000 min^{-1}, the best results were also achieved by fuel bias 65% on the primary and 35% on the secondary injector – 45.85 Nm (+0.50%) and 58.03 hp (+0.60%).
3. The measurements conducted on DynoJet 200-ix chassis dynamometer show a relatively low credibility as the calculated error (0.11 – 0.46 Nm and 0.12 – 0.45 hp) exceeds the differences between test modes (0.02 – 0.45 Nm and 0.03 – 0.38 hp). This could be attributed to changes in engine and possibly equipment temperatures. To increase the credibility of this study, the experiment should be repeated on engine dynamometer under tighter control of engine temperature and ambient conditions.

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