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Numerical Modelling of the Operation of a Novel Two Stroke V4 Engine

Albert Boretti[1,*]

[1]Department of Mechanical and Aerospace Engineering, Benjamin M. Statler College of Engineering and Mineral Resources, West Virginia University, Morgantown, WV 26506, USA

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Abstract

The paper presents simulations of the operation of a novel two-stroke engine for road Super Sport motorcycles. Two-stroke engines were preferred to four-stroke engines in Grand Prix motorcycle racing until the start of the Moto GP era when new rules phased out the two-stroke engines. Reasons for the change were the poor fuel economy and significant pollutant emissions. The paper discusses the opportunity of a came back at least in road Super Sport motorcycles thanks to the recent advantages in direct injection and precise lubrication for two-stroke engines, plus the opportunity to use jet ignition, based on performance simulations.

Keywords: internal combustion engines, direct injection, jet ignition, precise oiling, two-stroke engines

1. Introduction

Two stroke engines were the most popular choice in Grand Prix (GP) racing for half a century. Their phase-out was determined by new rules issued in favor of four-stroke engines in the Moto GP class of 2002 when 500 cm^3 two-stroke engines had to compete with 990 cm^3 four-stroke engines, and more recently with the replacement of the 125 cm^3 and 250 cm^3 two-stroke engines classes with the four-stroke engines Moto 2 and Moto 3 classes. The reasons of the replacement were the large pollutant emission and the low conversion efficiencies of the two-stroke engines. This was due to the lubricating oil premixed to the fuel and the reduced time to perform the same events of four-stroke engines. However, more than that, it was the interest of the manufacturers towards the much more expensive and complicated four-stroke engines rather than the very simple two-stroke engines to dictate the end of an era.

Significant improvements of injection and ignition systems and precise oiling have now made pollutant emission and fuel economy issues solvable today. The much better specific power and simplicity, reduced number of components and reduced weight of two-stroke engines may then return to be attractive. The present work demonstrated as the simple addition of direct fuel injection, precise oiling and jet ignition to an otherwise traditional design of the end 1990s with multi-port, intake reed valve, exhaust power valve and expansion muffler, may permit above 130 kW of peak power in a V4 500 cm^3 engine while permitting peak fuel conversion efficiencies approaching 40%. Two-stroke engines therefore have the potentials of a successful came-back in both track racing motorcycles and road Super Sport motorcycles

2. Latest Two stroke engines

Two-stroke engines were widespread in grand prix racing. One of the most successful Grand Prix engine of all the times has certainly been the Honda NSR500 engine [1]. The Honda NSR500 engine was produced for motorcycle Grand Prix racing by the

* Corresponding author. E-mail address: a.a.boretti@gmail.com

Honda Racing Corporation from 1984 to 2002. The engine was a two-stroke V4 of displacement half a liter. At the end of the 1990s, the engine was delivering 185-200 HP revving at 12,000-13,000 rpm. Honda won 10 titles of 500 cm^3 World Championships Grand Prix with the NSR500 from 1984 to 2001. Regulations for the World Championship motorcycle road racing 500 cm^3 class were changed drastically for the 2002 season with four-stroke engines permitted up to 990 cm^3 and up to six cylinders. The larger displacement four-stroke bikes dominated the series and the two-stroke including the NSR500 were phased out of the top Grand Prix class, with the smaller 125 cm^3 and 250 cm^3 permitted to survive for another decade.

For what concerns street bikes, the latest significant attempt to introduce in the market a two-stroke engine sport bike competing with four-stroke engine sport bikes was the Bimota V2 [2-3]. In 1996-97, Bimota, a small niche bike manufacturer, decided to introduce a new 500 cm^3 two-stroke motorcycle that would revolutionize sport bikes. The twin V-Due promised to meet the increasingly strict emissions laws while delivering power in excess of 100 HP in a bike of less than 130 kg when the four-stroke engines competitors were well above 200 kg. Up to that time, two-stroke engines used carburetors with constant fuel, oil and air premixing in the intake. This solution was releasing significant unburned hydrocarbons (HC) into the exhaust, as the exhaust ports are always open when the intake ports are open, and the exhaust ports always close last. Additionally, significant amount of residuals trapped at port closure and limited expansion to complete combustion made combustion incomplete translating in further increased HC and carbon monoxide (CO) emission. A significant amount of the oil mixed to the fuel to permit the lubrication of the reciprocating and rotating parts was burned within the cylinder, and the significant amount of fuel escaping combustion was drastically lowering the fuel conversion efficiency.

Bimota claimed they had fixed the emissions issue by developing a fuel-injected two-stroke engine with electronic ignition that had never been done in a motorcycle before. The fuel injection was not delivered by a modern direct injector located about the cylinder axis on the cylinder head, but by one port fuel injector of the time. The more precise injection was however believed to permit delivery of the fuel when needed into the cylinder mitigating the HC issue. The electronic ignition was also deemed to ensure a better spark timing for more complete combustion. Bimota also used forced lubrication for the bottom end, with a reduced amount of oil mixed to fuel to lubricate the pistons. The first production models of 1997 passed the emissions test but mostly due to the poor performances of the fuel injection system were very difficult to ride. The bike was delivering peak power of 76.6 kW / 105 HP @ 9000 rpm and peak torque of 90 Nm @ 8000 rpm [3], values in principle not that bad even if not outstanding, but the engine output was rather unpredictable. The engine also suffered from improper crankcase sealing due to crankcase castings issues. In 1998, Bimota reverted to a pair of carburetors, before going bankruptcy in 1999. All the stock sold at an auction was purchased by one of the designers who evolved the technique without too much funding. The latest models introduced in 2003 and 2004 were all based on carburetors and offered improved ride ability with even larger powers of 120-130 HP.

After the Bimota attempt, none of the major manufacturers designed two-stroke super sport street bikes. Only 125 cm^3 and 250 cm^3 street legal motorcycles were able to survive up to few years ago. The peak power 52.9 kW / 72.5 HP @ 11900 rpm peak torque 40 Nm @ 10750 rpm Aprilia RS 250 two cylinder was produced until 2004 [4]. The peak power 21 kW / 28 HP @ 10500 rpm peak torque 19 Nm @ 9000 rpm Aprilia RS 125 single cylinder was produced until 2012 [5].

While in street motorcycle applications the two-stroke engine has been phased out, however, in other applications, as marine or snowmobiles, the two-stroke engine has been able to remain competitive vs. four-stroke engines mostly thanks to the introduction of direct injection and precise oiling, techniques, however, also attempted in two-stroke scooters and small displacement motorcycles, but with a reduced level of sophistication.

The Rotax ETEC 800R engine [6-8] is an example of outstanding power density delivered within today's pollutant emissions and fuel economy constraints. This engine is an inline two-stroke two-cylinder engine featuring multi ports intake and exhaust, air-assisted central direct injection, precise oiling, an intake reed-valve a 2-into-1 tuned exhaust with a power valve. The fuel injectors are positioned adjacent to the spark plug in the cylinder head, injecting the fuel into the combustion chamber during the compression stroke at 30 bar of pressure, with air assisted atomization. The injection strategy includes full load homogeneous mixture and part load stratified mixtures playing with length and phasing of multiple injection events. The targeted oil injection supplies oil right where it is needed. The power valve on the exhaust permits three opening positions as these valves control three exhaust port openings per cylinder. By changing the height and size of the exhaust ports as RPMs change, the power valve improves fuel efficiency and torque over the speed range. The engine delivers peak power of 163.9 HP. Fuel usage, oil consumption and pollutants' emissions are drastically reduced vs. two-stroke competitors, similarly to smoke and smell.

Aim of this paper is to evaluate the advantages the adoption of modern high pressure, fast actuation gasoline injection coupled to jet ignition could deliver in an otherwise very basic design as those of the racing engines of the latest 1990s with mainly only reed valve intake and expansion chamber exhaust. Direct injection coupled to precise lubrication has emerged as a potential game changer for the two-stroke engine [9-16]. Sophisticated direct injectors with pressures up to 500 bar and piezo actuation that may deliver very well atomized fuel within very short time frames for homogeneous and stratified mixtures are here considered, together with jet ignition [17, 18]. Jet ignition is a technology now evolving to an almost off-the-shelf component. It may ignite air-fuel mixtures in the bulk with high energy for a quick combustion. The jet ignition device is a small pre-chamber with a direct injection injector that provides a small amount of fuel for locally stoichiometric or slightly rich conditions, and a spark plug to ignite that mixture. The pre-chamber is connected to the main chamber by a number of calibrated orifices which allow the jets of partially combusted products to ignite the main chamber mixture. The multiple jets of partially burned products produced by jet ignition result in shorter burn duration of premixed mixtures of air and different fuels, as proven in many experimental and computational works [20-34]. Jet ignition with various pre-chamber fuels has been successfully used with homogeneous mixtures of a variety of gaseous and liquid fuels.

3. Engine Model Development and Computational Results

To compute the advantages direct injection and jet ignition may deliver, a validated engine model is built by using the data provided in [19] for a 1999 Honda RS 125 cm^3 two-stroke racing engine. The paper [19] presents both experimental and computational results, plus details of the engine and the numerical model. Once the single cylinder engine model is validated, the V4 90 degrees engine model with individual expansion chambers is created by coupling the 4 single cylinder models through a common plenum. Replacement of continuous port fuel injection with prescribed fuel-to-air ratio direct injection is a straightforward modification, similarly to the simulation of jet ignition, simply translating in a 50% reduced burn duration vs. spark plug from prior studies.

The paper [19] presented the measured dynamometer results of a RS Honda 125 cm^3 two-stroke single-cylinder motorcycle grand prix road-racing engine operating at full throttle from 9,000 to 13,000 rpm. The engine was instrumented to provide in-cylinder and exhaust pipe pressure crank-angle histories. All the relevant engine geometry, discharge coefficients, scavenging characteristics and combustion data were published and used to simulate the engine using a one-dimensional engine performance simulation code. The experimental and computational performance parameters such as power and torque or volumetric and trapping efficiency and combustion parameters from pressure diagrams were compared. In addition to the baseline configuration, exhaust expansion chamber changes were also analyzed by both modelling and experiments.

The Honda RS 125 cm^3 fueled on aviation gasoline (Avgas 100LL) fuel delivered a peak power at the gearbox out shaft of 30 kW at 12,500 rpm. The standard carburetor was replaced with a butterfly throttle body and port fuel injection. The engine was run at a fuel rich air-to-fuel ratio of 11. The engine was instrumented with a piezoelectric pressure transducer located in the cylinder head to provide in-cylinder pressure, and two piezo resistive pressure transducers located in the expansion chamber to provide exhaust pressure. The measured combustion data allowed calculation of the mass fraction burned vs. crank angle then approximated with a Wiebe function, with the parameters then used in the model. Exhaust gas temperature and expansion chamber wall temperature were measured in the first header pipe section and the mid-section. Exhaust gas composition was also measured sampling at the mid-section. Measured flow coefficients from a flow bench rig were given for each pipe boundary. In addition to intake, exhaust and transfer ports, the throttle body and the reed valve were also carefully represented. The reed valve is a vee type unit with a total of 6 ports. The scavenging characteristics of the port layout were also evaluated using a single-cycle scavenge rig. From the measured indicated and brake mean effective pressures IMEP and BMEP, the first obtained from the measure of the in-cylinder pressure and the second obtained by the measure of the torque, the friction mean effective pressure was computed as the difference of the two. This value was used as a model input.

As performance is critically dependent on the trapped cylinder conditions, it is necessary to accurately predict the exhaust pressure waves and the trapped cylinder contents. Therefore, a heat transfer multiplier is applied to the expansion chamber walls in order to correctly phase the exhaust pressure waves. The combustion efficiency is also altered to give the correct peak cylinder pressure. After all, the combustion efficiency can be artificially adjusted to correlate well with the measured IMEP but give quite a different cylinder pressure trace. The measured fluctuating crankshaft velocities and a 21.0 mm restrictor at tailpipe entry are also modeled in the validation of the simulation program.

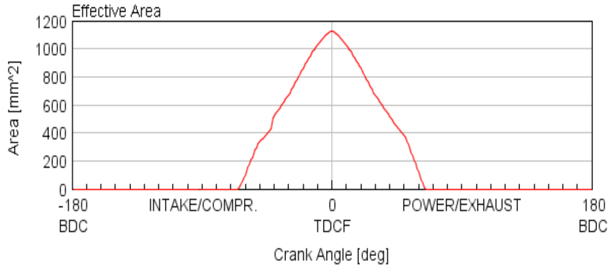
The single cylinder engine model was designed by using a commercial code [35] and the data of [19] delivering good results for the baseline configuration. The model correlates very well with the measured performance data, as it was the case of the code used in [19]. The differences between measured and computed performance parameters were (and are) well below 10%, despite some differences could have been spotted in detailed pressure traces.

With the model able to predict with reasonable accuracy the performances of the single cylinder engine, the V4 90 degrees engine model with individual expansion chambers was created by coupling 4 single cylinder engine models through a common plenum. Continuous prescribed fuel-to-air ratio port fuel injection and timed prescribed fuel-to-air ratio direct injection were introduced. Rather than fuel rich, the engine was operated stoichiometric or lean of stoichiometry. With spark ignition, same Wiebe function parameters of the validated model, 10-90% duration, 50% mass fraction burned angle and exponent were used over the speed range of interest. With jet ignition, a 50% reduced burn duration was imposed from prior studies.

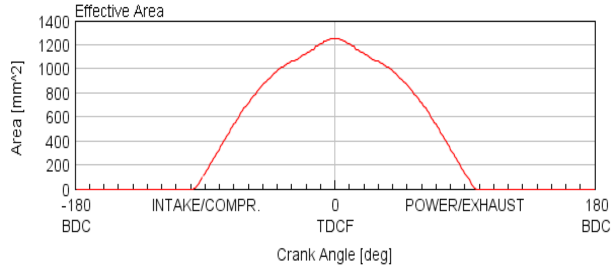
As engine performance is greatly affected by the operation of the expansion chamber with tuning strongly affected by the temperature of the exhaust gases when the exhaust port opens, maximum brake torque values may require delayed start of combustion. Further optimization of injection timings and spark timings or jet ignition timings is certainly possible. With direct injection the charge is cooled down and one-point higher compression ratio may be used. The combustion angle is taken 50% shorter with jet ignition, and the anchor angle is reduced accordingly. For sake of simplicity, lean mixtures use same Wiebe function parameters of stoichiometric mixtures. In the combustion model, all the fuel trapped within the cylinder is supposed to burn. This may not be the case with spark ignition or when significant wall wetting from direct injection may occur or due to crevices

Not present in [19], the simulations are performed by also adopting an exhaust power valve. This is a barrel valve rotating to close the exhaust duct in close proximity to the head to mitigate the effects of the off design operation of the exhaust expansion chamber. The control system of the proposed V4 engine makes use of intake throttle, exhaust power valve, injection timings and spark timing to deliver the best operation over the speed and load range. As this optimization would involve working under conditions far from those used for the model set up, this optimization is only partially performed.

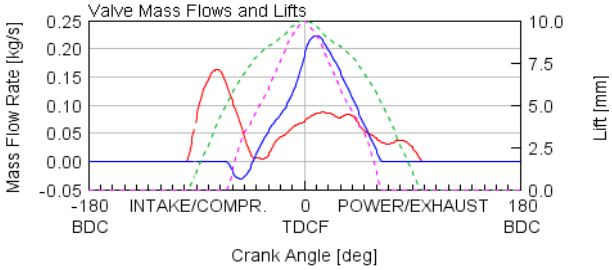
In the PFI version, the fuel injection occurs before the intake flow enters the crankcase. In the DI version, the fuel injection occurs directly within the cylinder. The jet ignition (JI) option simply translates in a faster combustion than the spark ignition version in the Wiebe model parameters. The reed valve is modelled as a matrix of flow coefficients. The exhaust and transfer ports are also defined as a matrix of flow coefficients. The throttle valve similarly to the power valve have effective areas computed as a function of an opening parameter, and the flow coefficients across the element are automatically computed.



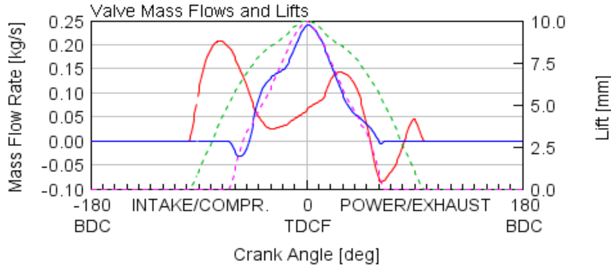
(a) Transfer ports area



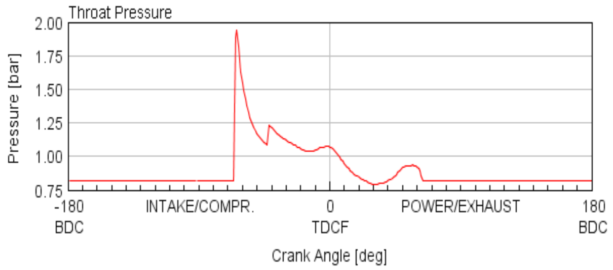
(b) Exhaust ports area



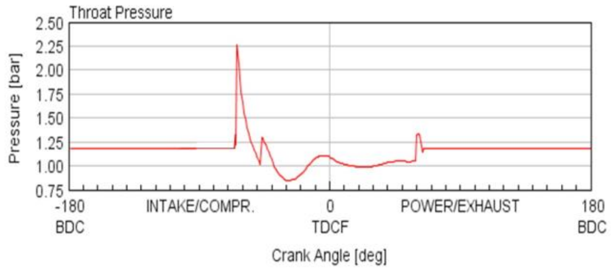
(c) Transfer and exhaust ports opening and mass flow rates, 15,000 rpm



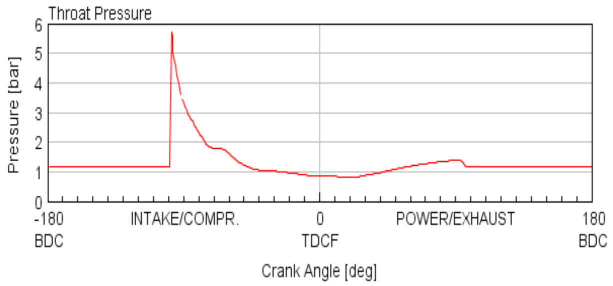
(d) Transfer and exhaust ports opening and mass flow rates, 13,000 rpm



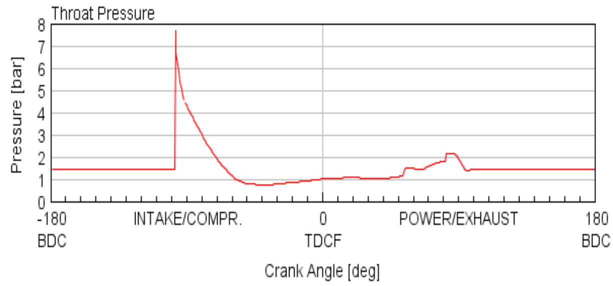
(e) Transfer ports throat pressure, 15,000 rpm



(f) Transfer ports throat pressure, 13,000 rpm



(g) Exhaust ports throat pressure, 15,000 rpm

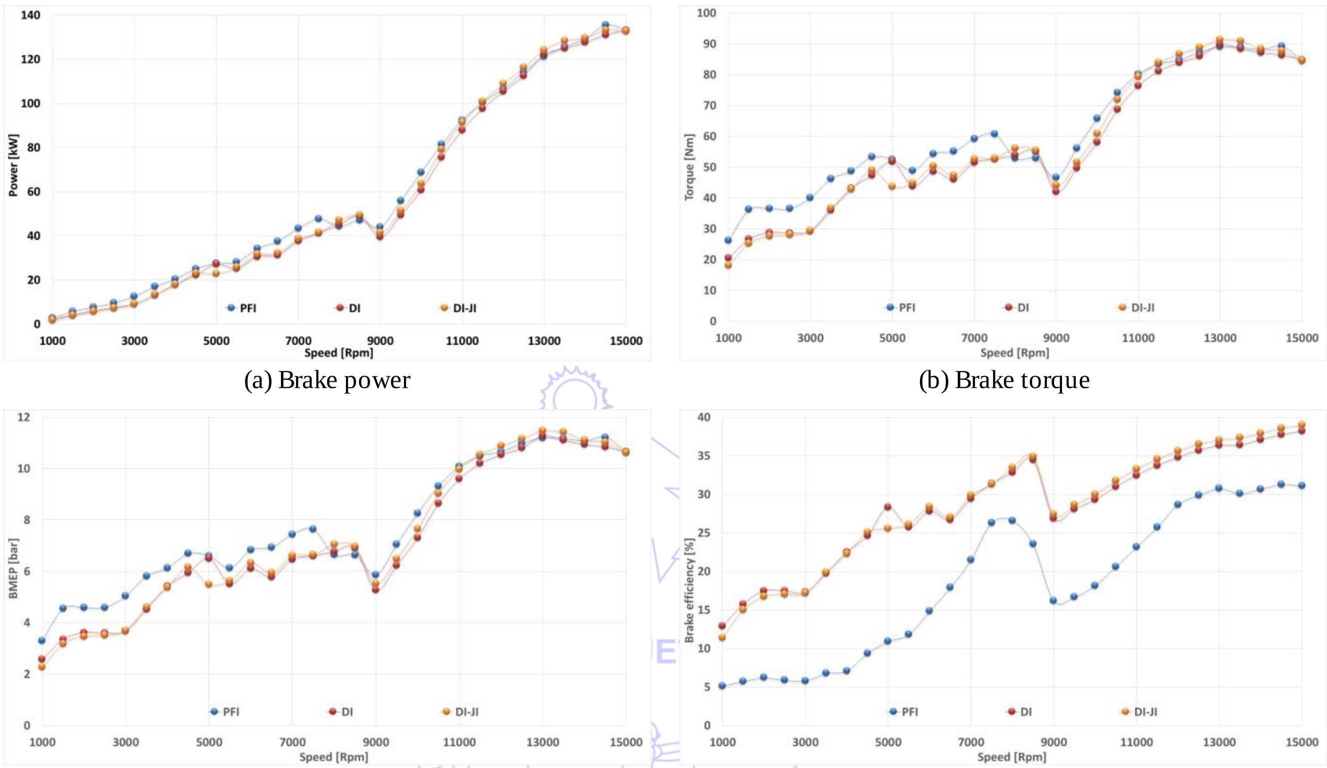


(h) Exhaust ports throat pressure, 13,000 rpm

Fig. 1 Maximum speed (15,000 rpm) and peak torque speed (13,000 rpm) WOT stoichiometric operation of the DI JI engine. Ports operating parameters vs. crank angle

Fig. 1 presents the ports operating parameters for maximum speed (15,000 rpm) and peak torque speed (13,000 rpm) wide open throttle (WOT) stoichiometric operation of the directly injected jet ignited engine. The ports operating parameters vs. crank angle are the transfer ports area, the exhaust ports area, the transfer and exhaust ports opening and mass flow rates, the transfer ports mass flow rates, the transfer ports throat pressure, the exhaust ports mass flow rates and the exhaust ports throat pressure. Clearly, despite the very simple design and the very short time to perform the gas exchange within the constraints of symmetric port timing, the two-stroke engine gas exchange process is very efficient when the expansion muffler is tuned.

Fig. 2 presents the computed brake power, brake torque, brake mean effective pressure and brake efficiency of port fuel injected spark ignited (PFI), directly injected spark ignited (DI) and directly injected jet ignited (DI-JI) engines, all working stoichiometric. In terms of power and torque outputs, the advantage of a higher compression ratio (DI) or a faster combustion (JI) is less relevant than the drop of the charge temperature produced by vaporizing the fuel in the intake. However, the port fuel injection option translates in drastically reduced brake efficiencies almost 10 points lower, as a significant amount of fuel escape combustion. With port injection, the density of the intake charge is much higher. However, during scavenging, not only air, but air plus fuel escape the combustion chamber.



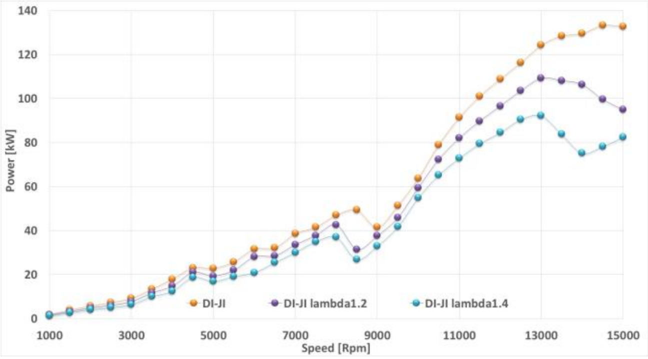
(a) Brake power (b) Brake torque

(c) Brake mean effective pressure (d) Brake efficiency

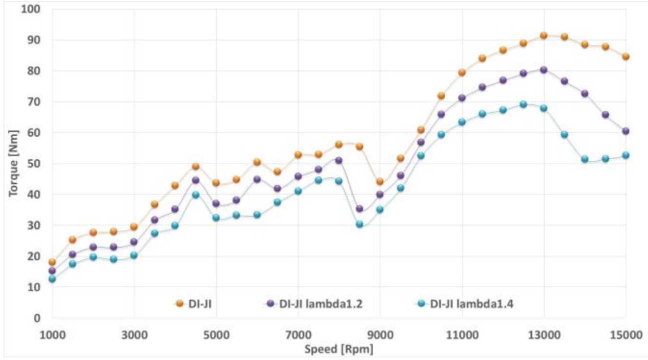
Fig. 2 Performance parameters of the port fuel injected spark ignited (PFI), directly injected spark ignited (DI) and directly injected jet ignited (DI-JI) engines, all working stoichiometric

Fig. 3 presents the operation of the DI JI engine with stoichiometric and lean of stoichiometry mixtures. The Wiebe function parameters are not changed for the lean operation. Lean of stoichiometry lambda 1.2 and 1.4 mixtures permit about same and even better fuel conversion efficiencies but clearly also reduced torque and power outputs. This option may be interesting working part load as running leaner may permit better efficiencies than throttling the intake. The control system may indeed manage the intake throttle, the injector and spark (and the exhaust power valve described later) to deliver the best efficiencies over the load and speed range. Not presented here, the comparison of the operation of the DI JI engine with stoichiometric and lean of stoichiometry mixtures made at constant torque (throttled stoichiometric vs. WOT lean mixture) shows the opportunity to achieve even larger efficiency gains.

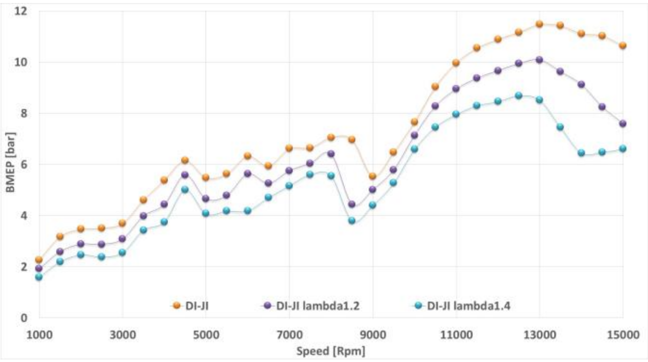
The present simulations refer to a racing bike. In case of a street bike, it must be noted that while a closed loop stoichiometric operation of the engine fitted with a three-way catalytic converter may deliver pollutant emissions within the limits, the operation of the engine lean of stoichiometry does not permit to satisfy the present emission standards. Without accounting for the changes in burn rate (homogeneous lean mixtures may require more time, but carefully stratified mixtures may reduce the difference and even permit faster combustions), running lean may permit small advantages but only in selected regions of operation



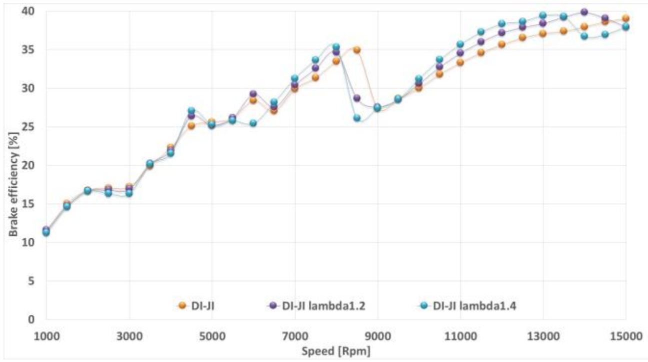
(a) Brake power



(b) Brake torque



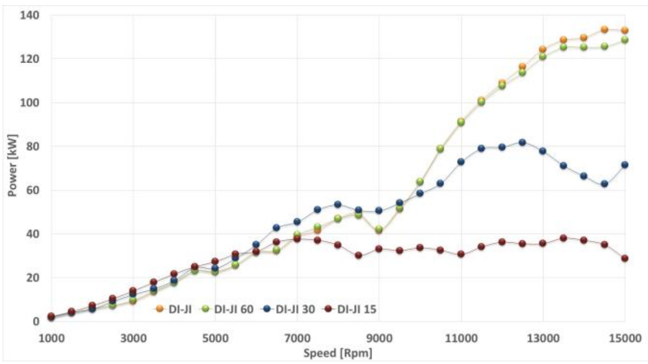
(c) Brake mean effective pressure



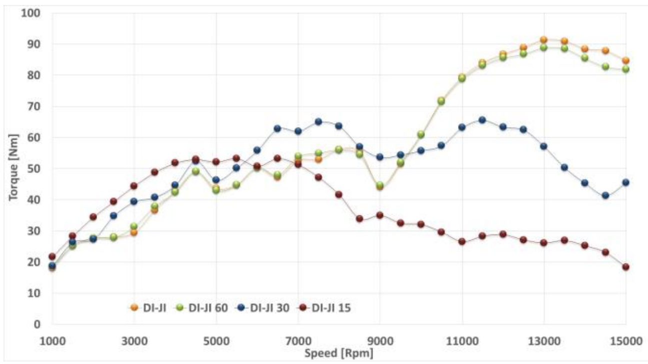
(d) Brake efficiency

Fig. 3 Performance parameters of the directly injected jet ignited (DI-JI) engines working stoichiometric and lean of stoichiometry lambda=1.2 and 1.4

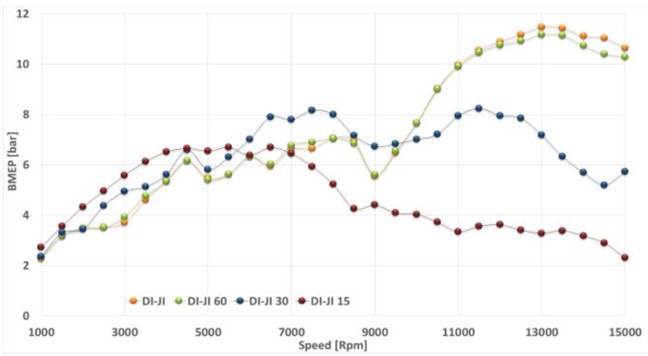
Fig. 4 shows the operation of the directly injected jet ignited (DI-JI) engine working stoichiometric with different openings of the exhaust power valve. The power valve is effective in mitigating the issues of having an expansion muffler out of tuning running at speeds below the range of optimum operation. While at high rpm the best performances are obtained with the power valve open, in the medium and low range of engine speeds there are advantages to partially close the power valve.



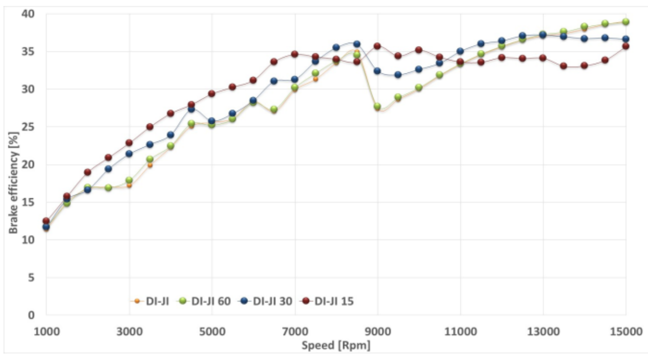
(a) Brake power



(b) Brake torque



(c) Brake mean effective pressure



(d) Brake efficiency

Fig. 4 Performance parameters of the directly injected jet ignited (DI-JI) engines working stoichiometric with different openings of the exhaust power valve

4. Discussion and Conclusions

The traditional two-stroke engine still possesses significant advantages in terms of weight, simplicity and power density vs. the four-stroke engine. The adoption of modern direct injection coupled to jet ignition and precise oiling mitigate the customary downfalls of the two-stroke engine, namely low fuel conversion efficiency and incomplete combustion with significant emissions of HC and CO, and the combustion of the lubricating oil [6-8]. Here direct injection and precise oiling is used in a racing engine to explore the potentials in terms of power output and fuel conversion efficiency.

The expansion chamber and the crankcase scavenging permit supercharging of the engine with very high brake mean effective pressures. This is achieved over the tuned range of engine speeds. The exhaust power valve may mitigate the drastic loss of performance occurring when the expansion chamber goes out of tune. With direct injection, this mitigation is reduced vs. port fuel injection or carburetor, but it is still relevant. In the case of the specific racing engine under study, direct injection, tuned muffler and exhaust power valve provides good results over the full range of engine speeds.

Over the small window of engine speeds for the tuned exhaust muffler and wide open throttle operation, a late 1990s two-stroke engines design only updated with high pressure, fast actuating direct injection injectors plus precise oiling and jet ignition is still competitive with today's best four-stroke engines. Despite the study has been performed for a racing bike engine, the most part of the advantages of direct injection, power valve and precise oiling are also obtainable for a street bike engine fitted with a three-way catalytic converter.

The novelty of the study is that with direct injection, jet ignition and precise oiling, two-stroke engines otherwise designed following 25 year-old concepts are competitive in terms of outputs and fuel economy with the latest four-stroke engines. This is demonstrated for a V four engine of 500 cc, and it is very likely true not only for racing bikes' engines, but also for street bikes' engines that must satisfy pollutant emissions and noise limits, even if more work is certainly needed to design a fully compliant street bike.

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