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PRELIMINARY STUDY OF A POPPET VALVE TWO-STROKE ENGINE OPERATING WITH CONTROLLED AUTO IGNITION APPLIED TO POWER GENERATION

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Abstract. *The simpler and more common two-stroke engines are known for their high power density and high level of pollutant emissions. The use of a poppet valve design, combined with controlled auto ignition, can improve two-stroke engines efficiency and may achieve current emissions standards. This work presents a preliminary analysis of a poppet valve two stroke engine (PV2SE), aimed for power generation, working with controlled auto ignition (CAI) combustion and ethanol as fuel. The analysis consists of a set of computational performance simulations of a parameterized PV2SE feed by values found in literature. A conventional four-stroke engine model is also created to establish a benchmark. The simulation results are presented and some configurations for operation are proposed.*

***Keywords**: Two-Stroke Engines; Poppet Valve; CAI Combustion; Power Generation; Ethanol*

1. INTRODUCTION

The simpler and more common two-stroke engines are known for their high power density and high level of pollutant emissions. This is due mainly to the fact of: performing a complete cycle for each crankshaft turn; work with intake and exhaust systems by means of ports instead of valves and need lubricating oil added to the fuel. Despite the configuration of intake and exhaust ports being the most used in two-stroke engines, there was an effort of some companies to enable the configuration with valves in the cylinder head, also known as poppet valve two-stroke engine (PV2SE). The attempt was aiming to keep the power density previously achieved but reduce drastically the level of emissions.

Not necessarily restricted to the scope of the two-stroke engines, studies were conducted seeking to improve the efficiency of internal combustion engines by mean of, among other approaches, controlled auto ignition (CAI). The occurrence of controlled auto ignition requires a relatively high level of pressure and temperature in the combustion chamber, which may be obtained by an appropriate combination of compression ratio, intake air heating and exhausts gas recirculation (EGR) [1]. What differs CAI combustion from knock is the rate that the reactions happen. The reactions rates in CAI are slower than in knock due to diluents, such as air in excess or EGR.

The use of EGR has been considered an appropriate alternative to create and control the conditions inside the combustion chamber where CAI is desired. A common practice is the use of burned gas from the previous cycle because it is hot and presents active radicals which facilitate the ignition. The percentage of EGR in the mixture is critical to the process of CAI.

It is assumed that the adoption of the CAI operation in PV2SE with ethanol as fuel, presents a unique potential application for power generators. Operating with EGR and stoichiometry (or leaner) mixtures, the demand for fresh air and the work associated with pumping losses are sensibly reduced due to the fact that the burned gas necessary for CAI simply can be retained in the combustion chamber. Ethanol should be an interesting fuel because it has a lower tendency to detonate when compression ratio is increased if compared with gasoline. For a poppet valve engine, a simple change in the valve timing allows the variation of the EGR quantity kept in cylinder. The CAI combustion is thought to result in significantly lower temperatures than those found inside the reaction zone of a spark ignition engine. As a result, the NOx emission so as the specific fuel consumption are expected to be dramatically reduced [2].

This work presents a preliminary poppet valve two stroke engine (PV2SE) analysis, aimed for power generation, working in controlled auto ignition (CAI) combustion with ethanol as fuel. The analysis consists of a set of computational performance simulations of a parameterized PV2SE feed by values found in literature. A conventional four-stroke engine model is also created to establish a benchmark. The simulation results are presented and a configuration for the operation is proposed.

2. THEORETICAL CONSIDERATIONS

2.1. Two-stroke engines

The main advantages of two-stroke engines are its greater simplicity, lower weight and higher power density. The high power density makes its operation smooth due to the greater number of power strokes in respect to the number of crank turns if compared with four-stroke engines. An issue concerning two-stroke engines comes from the difficulty in achieving the latest emissions standards. Usually two-stroke engines burn a mixture of oil/fuel increasing the emissions. This is due to the fact that usually the mixture is compressed in the crankcase in order to enhance the scavenging process. In scavenging processes, some of the mixture induced escapes to the exhaust system. If the engine has port injection, higher hydrocarbons (HC) emissions are expected [3].

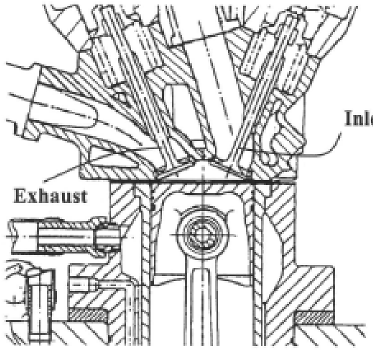

Figure 1. Flagship poppet valve two-stroke engine (Adapted from [5])

In the poppet valve concept, the traditional ports are replaced by poppet valves in cylinder head. Figure 1 shows that the poppet valve two-stroke engine is very similar to the conventional four-stroke design.

A poppet valve two-stroke engine is cleaner than the conventional two-stroke ones, especially when direct injection is used. The use of a direct injection system enables a scavenging process using only air and thus reducing the fuel loss through the exhaust system.

The main advantages of the replacement ports by poppet valves are the greater control of scavenging process [4, 5] and the possibility of variable valve timing [5]. This increases the engine speed range. Figure 2 describes the scavenging process during a two-stroke engine cycle. The process begins with the intake valve open and terminates with the intake valve close. The scavenging efficiency is very sensible to the intake pressure, cylinder and piston geometry and valve timing. The valve orientation and timing may be more important parameters in controlling the cylinder internal flow and scavenging process [6]. The use of a supercharger or a turbo-compressor instead of crankcase mixture compression eliminates the need to burn oil together with the mixture resulting in a cleaner operation. The main disadvantage of the poppet valve two-stroke concept is the increase in mechanical complexity due to a high number of moving parts and the extra weight added by them [6].

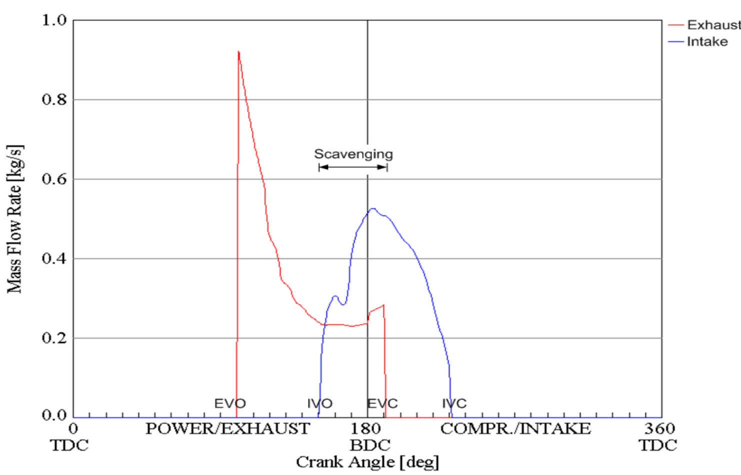

Figure 2. Scavenging process on a 2 stroke engine

2.2. Homogeneous charge compression ignition

Homogeneous charge compression ignition (HCCI) is a particular mode of controlled auto ignition combustion where a homogeneous charge consisting of gaseous mixture and vaporized fuel is compressed until the occurrence of auto-ignition. This auto-ignition occurs at the same time in several points of the cylinder resulting in rapid and uniform combustion. Engines that work with HCCI combine the advantages of spark ignition engines (SI) and compression ignition engines (CI). Combustion is performed with a homogeneous charge, characteristic of SI engines, reducing the emission of soot. At the same time HCCI operation occurs without a throttle, characteristic of CI engines, increasing the efficiency and reducing pumping losses [2, 7]. HCCI engines can operate without a throttle because charge control can be made varying the amount of EGR gas trapped in the cylinder.

The use of HCCI presents the following advantages: elimination of throttle, operation with high compression ratio since knock is unlikely for particular conditions, no need of a spark plug to start the combustion, possibility of working with lean (sometimes extremely lean) mixtures. The absence of a flame front propagating along the combustion chamber, reduced combustion time and the possibility of working with various fuels mixtures are also advantages of HCCI combustion. Emissions are reduced as well because lean mixtures results in lower maximum gas temperatures, therefore, lower NOx emissions [1].

The main disadvantages found in the HCCI application in engines are its limited load range of use and combustion control complexity. Usually HCCI control is done through compression ratio variation [8], valve timing and percentage of EGR in the cylinder [9, 10, 11]. This control is relatively simple for an engine operating in permanent regime but it turns to be a challenge during transient operation, where the parameters that influence auto-ignition change significantly. HCCI is limited by load where the percentage of EGR or some other diluents are not enough to maintain the reaction rates under suitable levels. In this condition, the combustion presents a threat to the engine integrity. An alternative for controlling the combustion on high loads is the use of water injection in the cylinder to reduce the temperature of the gases as the intensity of combustion.

It is very difficult to start an engine in HCCI mode. This is because all the components of the combustion chamber are relatively cold. It can be difficult to stabilize the HCCI operation too, so there is the need of a supplementary ignition system. During the cold start and transient loads, the engine can work as a traditional SI. When the conditions are established, the engine management system can adjust the parameters to start HCCI mode.

2.3 Power generation using piston engines

Reciprocating engines are used throughout the world in applications ranging from lawn mowers to cars, trucks, locomotives, ships and for power and cogeneration. Engines vary in size from less than 1kW to 65.000 kW [12]and they can burn a wide range of fuels including natural gas, biogas, LPG, gasoline, diesel, biodiesel, heavy fuels oil and even coal.

Applications of piston engines as a power generator are enormously varied. Small units can be used for standby power or for combined heat and power in homes and offices. Larger standby units are often used in situations where continuous power supply is critical; like in hospitals or air traffic control facilities. Others commercial and industrial facilities use medium-sized piston engine based to generate power for their own utilization, usually when no electricity network is available. Large engines meanwhile can be used for base load, network-connected power generation.

Piston engines used for power generation are almost exclusively derived from engines designed for motive applications. The smaller ones normally are based on car or truck engines while the larger engines are based on locomotive or marine engines. Performance of these engines varies. Because they are mass produced, smaller engines are usually cheap but they have relatively low efficiencies and short lives. Larger engines tend to be more expensive (per kW) but they will operate for much longer. The largest engines in the megawatt scale are probably the most efficient prime movers available, with simple cycle efficiencies approaching 50% [12].

The number of revolutions per minute that a piston engine operates will depend on its size, because of inertial forces. Usually small engines will operate at higher speeds than the larger ones. Since in most situations a piston-engine-based power unit will have to be synchronized to an electricity network operating at 50 or 60Hz, the engine speed must be a function of one of these rates.

Table 1. Engine application range

	Engine Size (MW)	Engine speed (rpm)
High speed	0.01-3.5	1000 - 3600
Medium speed	1.0 - 35.0	275 - 1000
Slow speed	2.0 - 65.0	58 - 275

The environmental considerations made to piston engines used as power units are the same as the ones applied to oil and gas-fired power plants; the emissions resulting from fuel combustion. The main emissions are nitrogen oxides (NOx), carbon monoxide (CO) and volatile organic compounds (VOCs). Diesel engines, particularly those burning heavy diesel fuels will also produce particulate matter (carcinogenic) and some sulfur dioxide (acid rain).

Piston engines used as power units usually run in continuous operation, so its performance should be predictable and less susceptible to variations. Starting and stopping, as encountered in transport applications delivers more stress on the machines.

The greatest risk attached to the operation of a piston engine power plant is related to fuel supply [12]. Oil prices can be particularly unstable and unpredictable due to geopolitical issues. The development of a large piston engine power plant will usually include fuel production control, in cases of bio-fuels or long-term fuel supply agreement.

Maintenance costs vary with engine size and type. Small, high-speed engines generally require the most frequent maintenance while larger engines can run for much longer periods without attention.

2.4 Energy scenario in Brazil

Ethanol and other bio-fuels inclusion in energy matrix is a extreme importance factor not only in the environmental point of view, but also in the energy security sense [13]. The dispute for oil and its derivatives in response to growing demand for energy makes the partial replacement of petrol a critical element in the energy systems management. The bio-fuels international market should increase in the coming years due to geopolitical and economic issues.

In the period 2003-2005, ethanol use won extraordinary boost with the European Union Directive 2003/30 and with the United States Energy Policy Act, 2005, which defined tax reduction and loan guarantees for renewable fuels. The Energy Policy Act established chronograms for ethanol minimum consumption volumes. In the United States, the volume of bio-fuels, in its most ethanol, to be mixed to petrol must reach the 28 billion liters in 2012 and 136 billion liters in 2022, in accordance with the new rules of the sector. In Europe, bio-fuels shall replace, in energy content, 5.7% of fossil fuels until the end of 2010 [13].

Energy is a fundamental input for the functioning of society. Sector performance analysis is therefore crucial to show the opportunities and obstacles which they shall input to economic activities in the coming decades. The choice of a given alternative energy font involves heavy investments, and has long term effects on industrial patterns production and society use, changing the entire production mode. Figure 3 shows a prediction of the Brazilian energy matrix from 2007 to 2030.

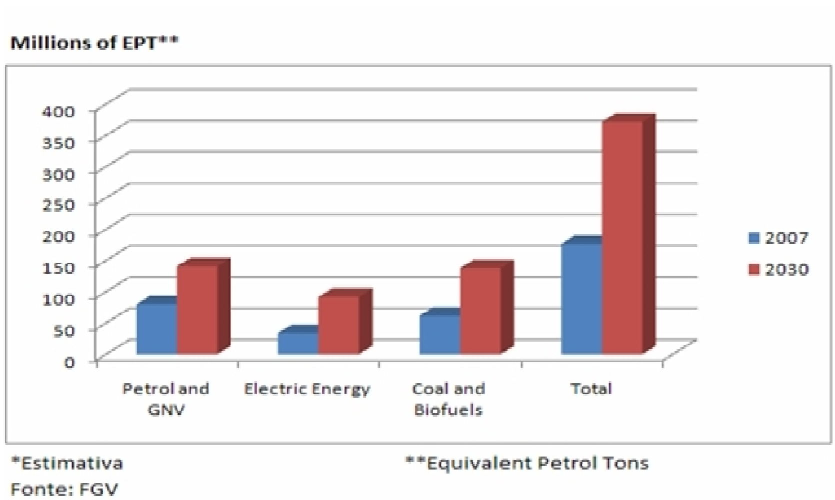

Figure 3. Brazilian Energy Matrix (adapted from [13])

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3. MODEL SETUP

The GT-Power software, from Gamma Technologies, is used for computational simulations. The software is based on one-dimensional flow and heat transfer calculations of the engine components. The computational model consists in providing the software with information about these components geometry and timing of the moving parts. Figure 4 shows GT-Power graphical interface.

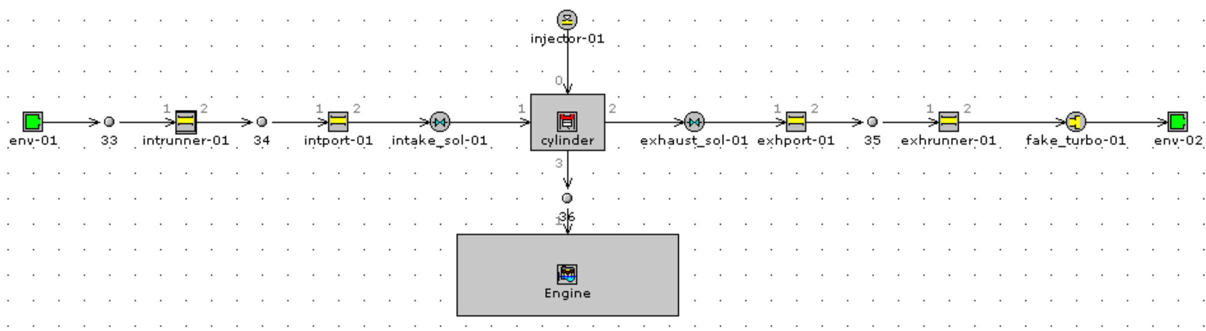

Figure 4. Computational model developed in GT-Power

A simplified model of a single cylinder engine was built, with the admission and exhaust systems consisting of two parts each, a single runner and a port. The fuel injection is done in the liquid phase directly into the cylinder. The injection model has flexible parameters such as: injection timing angle, injected fluid temperature, vaporized fuel fraction and equivalence ratio. The last one is reference to the air induced.

Solenoids are used to control the opening and closing of the valves, therefore, the intake and exhaust valves timing are completely independent. By this mean, it is also possible to vary the lift for each valve on each operation condition. Despite the complete flexibility, both valves have a fixed lift value that is reached in one millisecond. It important to note that with all parameters defined, it is possible to use conventional camshafts for valve control, especially for stationary purposes.

A simplified model for a turbocharger is used. It is done by placing a restriction in the exhaust system and then imposing a positive pressure in the intake. The restriction should create a drop in energy which has to be greater than the energy necessary to compress the intake gases up to the intake pressure.

The scavenging process has a fundamental influence on the performance and emissions control in two-stroke engines. Valve or window overlap occurs during scavenging, where fresh mixture can go through the exhaust system and leave as unburned hydrocarbons. Ideally scavenging would drive all burned gases outward and leave in the cylinder only fresh mixture. In practice this situation does not occur, so scavenging and valve timing determine the quantity of EGR left in the cylinder as well as the quantity of fresh mixture lost by the exhaust system. Scavenge modeling in GT-Power is performed with the input of an array object that relates burned gas percentage leaving the cylinder with the burned gas percentage standing inside it. There are two ways of creating this array, one is experimentally and the other is using computational fluid dynamics simulations.

The modeling of HCCI combustion may be carried out through the use of a single-temperature zone combustion model that uses a Wiebe function to impose a heat release rate. It is a simple function with only two parameters, Combustion Phasing ($T_{50\%}$) and Burn Time (BT). The combustion phasing defines the point where 50% of the combustion reaction occurs. The burn time, which can be measured in crank angle (CA), is the time needed to complete the reaction from 10% to 90%. Studies suggest that HCCI can be modeled with a short burn duration and a combustion phasing around TDC. It is possible to use a predictive model to simulate the CAI combustion, but this is something that needs not only more computational time, but a great chemical kinetics and turbulence understanding.

To compare the results and validate the model, the same single cylinder engine is used, but this time running in the conventional four-stroke configuration (PV4SE). Despite the simplicity, this is a valid way of evaluating the probing engine performance because most components that are found in the four-stroke engine (complex runners, poppet valves) would be shared with this new configuration. It is important to note that the results are valid only if the convergence criteria for mass, pressure and energy conservation is achieved.

3.1. Computational Model Parameters

The parameters that correspond to engine physical characteristics are listed below.

Table 2. Models specifications

Two-stroke Model			Four-stroke Model		
Bore:	151	[mm]	Bore:	151	[mm]
Stroke:	168	[mm]	Stroke:	168	[mm]
Connecting Rod:	240	[mm]	Connecting Rod:	240	[mm]
Intake Valve Diam.:	80	[mm]	Intake Valve Diam.:	80	[mm]
Intake Valve Lift:	20	[mm]	Intake Valve Lift:	20	[mm]
Exhaust Valve Diam.:	85	[mm]	Exhaust Valve Diam.:	75	[mm]
Exhaust Valve Lift:	20	[mm]	Exhaust Valve Lift:	20	[mm]
Exhaust Pressure:	1	[bar]	Exhaust Pressure:	1	[bar]

Table 2 shows that the only physical difference between the two engines is the exhaust valve diameter. The value in the two-stroke configuration is slightly greater because the exhaust valve opens earlier.

The operational parameters which can be varied in order to analyze their influence are listed below.

- Valve Timing (referenced in TDC and measured clockwise in degrees):
  - Intake Valve Open (IVO)
  - Intake Valve Close (IVC)
  - Exhaust Valve Open (EVO)
  - Exhaust Valve Close (EVC)
- Compression Ratio (CR)
- Gauge Intake Pressure (GIP)
- Air to Fuel Ratio (A/F)
- Combustion Model:
  - Combustion Phasing (CP)
  - Burn Time (BT)

The following indicators are used to compare the performance between configurations:

- Indicated Specific Fuel Consumption (ISFC).
- Brake Power (BP).

Both indicators listed above, plus the Indicated Mean Effective Pressure (IMEP), are used to compare the performance of the different operational configurations of the proposed engine.

Additionally, the flow indicators used to analyze breathing and scavenging performances are listed below:

Volumetric Efficiency (VE):	Mass of delivered air per cycle divided by a reference mass
Trapping Ratio (TR):	Mass of delivered air retained divided by mass of delivered air
% of EGR (EGR)	Mass of burned gas divided by trapped mass

The combined turbo-compressor efficiency (CTCE) is calculated dividing the additional energy to get the intake gases from ambient pressure up to the intake pressure by the energy lost by the exhaust gases in the restriction.

4. RESULTS

Several cases were solved in GT-Power in order to evaluate the performance of both engines. The simulation results are shown on Figs. 5-14. Figures 5-6 compares the performance between the two-stroke and four-stroke engine. Figures 7-14 relates only to the two-stroke configuration.

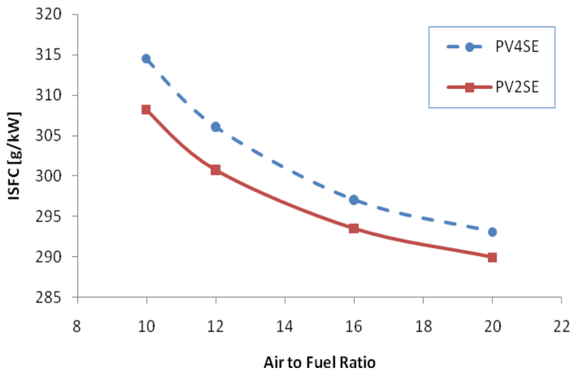

Figure 5. ISFC comparison
(CR: 12)

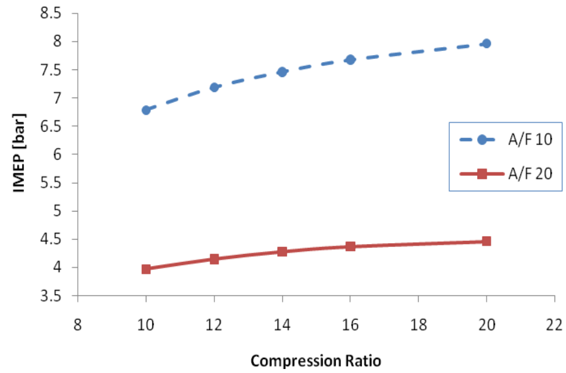

Figure 8. IMEP changes with CR (GIP: 1)

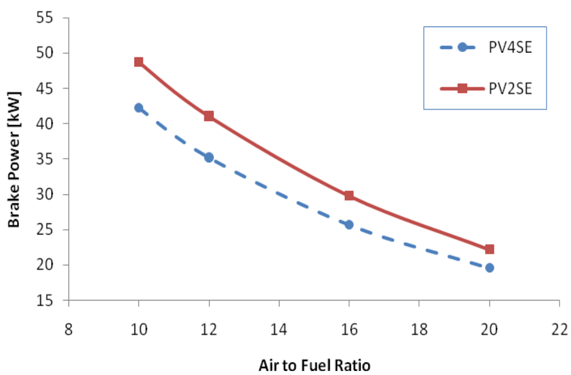

Figure 6. Brake Power comparison
(CR: 12, GIP: 1)

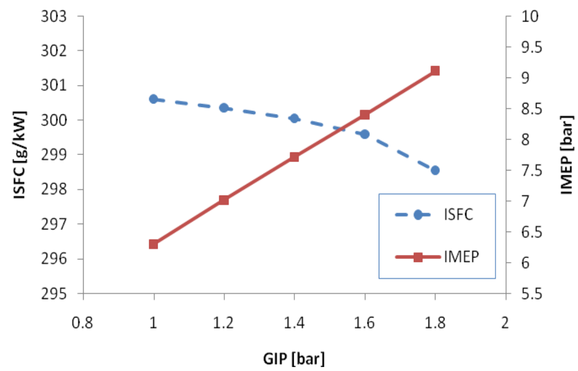

Figure 9. ISFC changes with GIP
(CR: 12, A/F 10)

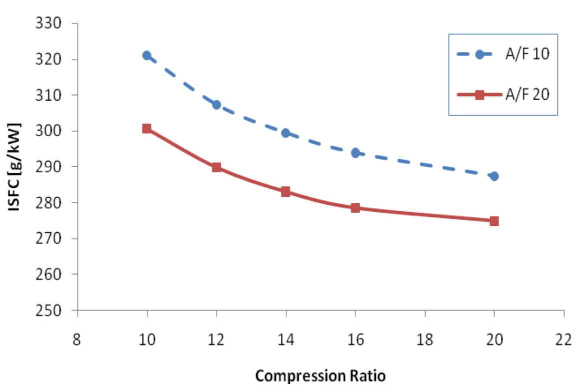

Figure 7. ISFC changes with CR (GIP: 1)

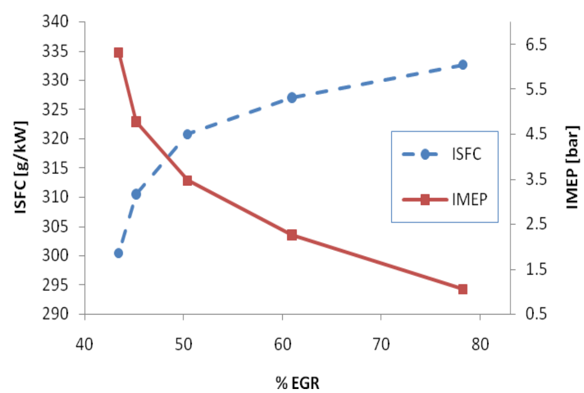

Figure 10. ISFC changes with % of EGR
(GIP: 1, CR: 12, A/F: 12)

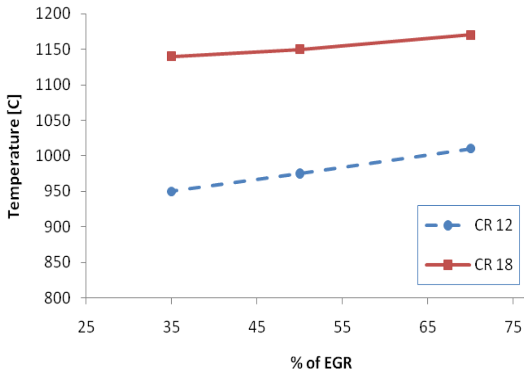

Figure 11. Temperature at the end of compression changes with % of EGR (A/F 12, GIP: 1)

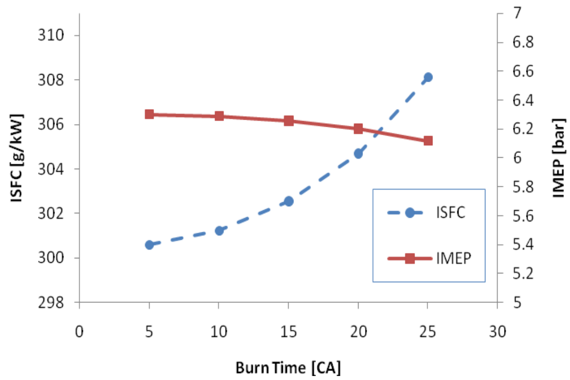

Figure 13. ISFC and IMEP changes with burn time (CR: 12, A/F 12, GIP: 1)

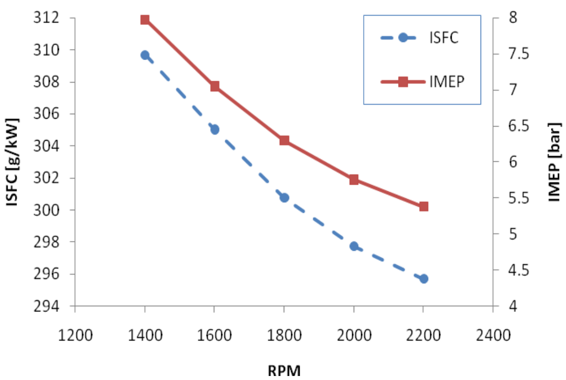

Figure 12. ISFC and IMEP changes with rpm (CR: 12, A/F 12, GIP: 1)

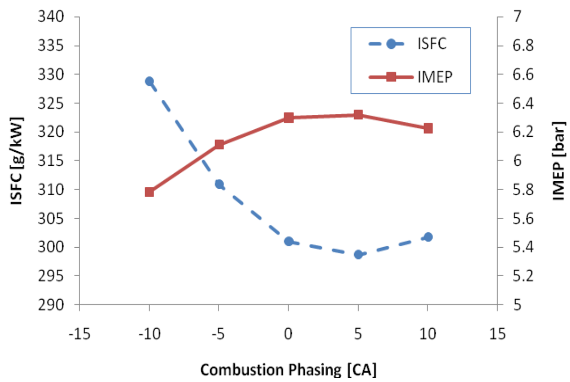

Figure 14. ISFC and Brake Power changes With combustion phasing (CR: 12, A/F 12, GIP: 1)

Figures 5 and 6 compare the performance between the four-stroke and two-stroke configuration. The two-stroke engine has a lower ISFC and higher brake power than the four-stroke one. Both indicators are sensible to A/F ratio, becoming lower for leaner mixtures. It should be noticed that these models run on different operational conditions, i.e. different: percentage of EGR, valve timing and gauge intake pressure. The two-stroke model used on Figures 5 and 6 refers to configuration (1) on Table 3. The four-stroke model is presented in the same table as well.

Figure 7 shows that for the same value of compression ratio a leaner mixture will lead to lower ISFC. IMEP has the opposite tendency and is more sensible to A/F ratios than to compression ratio, as shown in Figure 8. On both figures the lines tend asymptotically to a fixed value. This happens exactly in the same way in four-stroke engines.

Figure 9 shows values of ISFC and IMEP for different gauge intake pressures. A value around 1.6 bar is found to be an optimum value for this model, maximizing IMEP and minimizing ISFC. It is important to note that for the same valve timing, the intake pressure is dependent of the combined turbo-compressor efficiency.

Figure 10 show curves of IMEP and IFSC for different percentages of EGR. The ISFC is less sensible to EGR variations than IMEP. Figure 11 plots the temperature at the end of compression cycle for different percentage of EGR. It shows that auto-ignition temperature for ethanol may be achieved.

Figure 12 shows that the engine is very sensible for variations in the speed, so the control of valve timing would be very complex and critical for applications where speed varies. For stationary engines this challenge is minimized for obvious reasons. Burn time effects on ISFC and brake power are shown on Figure 13. Fast combustion improves ISFC and brake power values because the cycle is closer to an ideal constant volume one. It is also noted that ISFC is more sensible to slower combustion than IMEP.

Figure 14 relates combustion phasing with ISFC and brake power. It is seen that for earlier combustion phasing higher ISFC and lower brake power are expected since work during compression is increased. As the combustion phasing shifts closer to top dead center, lower values of ISFC and higher values of IMEP are expected. Higher angles cause a decrease on IMEP and an increase on ISFC because expansion cycle is less efficient.

Analyzing the data generated by the simulations, a set of configurations is proposed and the performance evaluated. The results are compared with an equivalent four-stroke engine. All configurations satisfies a project requirement of an output of 42 [kW] per cylinder at 1800 RPM.

Table 3. Performance of different configurations

	PV2SE (1)	PV2SE (2)	PV2SE (3)	PV2SE (4)	PV2SE (5)	PV4SE
Bore [mm]	156	156	156	156	156	156
Stroke [mm]	156	156	156	156	156	156
Displacement [l]	3	3	3	3	3	3
Compression Ratio	12	16	16	16	18	12
Air to Fuel Ratio	12	12	12	15	18	12
Intake Valve Diameter [mm]	80	80	80	80	80	80
Intake Valve Lift [mm]	20	20	20	20	20	20
Exhaust Valve Diameter [mm]	85	85	85	85	85	70
Exhaust Valve Lift [mm]	20	20	20	20	20	20
Intake Pressure [bar]	2	2	2.5	2	2.2	1
Exhaust Pressure [bar]	1	1	1	1	1	1
* IVO	150	150	150	150	150	20 BTDC
* IVC	200	220	181	220	220	30 ATDC
* EVO	97	110	75	92	80	30 BBDC
* EVC	180	180	180	180	180	20 BTDC
EGR	42.3%	49.6%	35.2%	43.2%	39.4%	3.9%
** Volumetric Efficiency	56.0%	53.8%	60.0%	67.6%	84.3%	83.0%
Trapping Ratio	91.5%	91.6%	91.2%	92.7%	93.4%	100.0%
*** Temperature [degC]	835	910	825	830	825	n/a
*** Pressure [bar]	38.5	61	48	63	82	n/a
Speed [RPM]	1800	1800	1800	1800	1800	1800
Turbo-compressor efficiency	74.3%	88.8%	91.2%	77.3%	76.3%	n/a
ISFC [g/kW]	303	280	315.5	284.5	288.8	314.4
IMEP [bar]	6.42	6.6	6.57	6.59	6.77	11.1
Brake Power [kW]	42	42	42	42	42	42

* referenced in TDC and measured clockwise in degrees
** referenced to intake pressure
*** value at the end of compression

It is clear in configuration (1) that for the same air to fuel ratio and compression ratio a lower indicated specific fuel consumption than in the four-stroke is achieved. Configuration (2) has the exhaust and intake valve opening later; this means that the combined efficiency of the turbo-compressor would have to be higher than in (1) in order to be feasible. This is the configuration that shows the lower indicated specific fuel consumption.

In configuration (3), a higher boost pressure is used. To achieve this boost pressure the exhaust valve has to open earlier. To limit the amount of intake gases as the power output, the intake valve has to close earlier. This strategy seems to be not interesting. Configuration (4) shows an attempt to run with relatively high compression ratio and very lean mixtures. The conditions created in the end of compression seem to be suitable for the occurrence of auto-ignition.

In configuration (5) an even leaner mixture is used. In order to achieve the output required, the boost pressure has to be slightly higher. Since the mixture is leaner, the temperature and pressure inside the cylinder during the power stroke are lower. The exhaust valve has to open earlier in order to supply the energy necessary to compress the intake gases.

5. CONCLUSION

It is expected a great increase in the use of bio-fuels in Brazil for the next decades. This is mainly due to the volatile prices of foreign oil and growing pressure to use cleaner sources of energy.

Medium sized piston-based power units are widely used in the country, burning primarily diesel. The proposed engine shows an alternative for these conventional four-stroke engines. It almost does not differ in components, but due to CAI combustion, is expected to be more flexible in terms of fuels and be more mechanically efficient.

Although the results are quite promising, there are some major practical challenges that should be overcome in order to create a fully operational engine. One is obviously to propitiate an efficient scavenging and charging, without a great loss of fresh air to the exhaust system. Another is to create a turbine and compressor that are efficient operating in the new conditions.

This paper consists of a preliminary evaluation of the performance of the proposed engine. For a deeper analysis it would be almost mandatory to have a good chemical kinetic model of CAI combustion as well as a Computational Fluid Dynamic model for the process of scavenging. The results should be compared with experimental data in order to calibrate the models, then, a complete numerical model can be constructed so it could help the development of a prototype.

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