EXPERIMENTAL AND MATHEMATICAL STUDY OF THE EFFECT OF HYDROUS ETHANOL (93,7 °INPM) AND TYPE C BRAZILIAN AUTOMOTIVE GASOLINE BLENDS ON SPARK IGNITION ENGINE PERFORMANCE

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Abstract. *This study evaluates the effect of hydrous ethanol (AEHC) and Brazilian automotive gasoline (GASC) blends on the performance of Brazilian MPFI Flex Fuel engines. These blends were investigated under several compositions, advance ignition, and air-fuel equivalence ratios. A four-stroke, four-cylinder, 1595 cm³, 16-valve cross-scavenged, water-cooled engine was used, and both fuel injection and ignition were controlled by a programmable Electronic Control Unit (ECU). The experiments were performed at constant speed of 2500 rpm and wide open throttle, while varying the operational parameters. The engine performance was evaluated for brake torque, brake specific fuel consumption, and consumption rate. The limited-pressure ideal model was used to build a mathematical model, and the set of equations were solved through the Newton-Raphson method and coupled in time by Euler numerical integration algorithm. The results have showed that blending GASC with AEHC does not cause an appreciable difference to brake torque, but the mathematical model has indicated a lower torque of AEHC operations regarding GASC operations. BSFC and consumption rate showed positive linear correlation when AEHC was added to the blend. The addition of 25% of AEHC to the blend caused an increase of approximately 10% for consumption fuel, and the difference in consumption rate to AEHC regarding GASC was greater than 40%. The advance ignition proved a strong relation to brake torque, and MBT was verified to 20º BTDC and $\lambda = 0.9$ independently on the blend composition.*

Key words: *hydrous ethanol, flex fuel engine, BSFC, brake torque, fuel consumption.*

1. INTRODUCTION

In Brazil, in the 80's, just like other projects, such as methanol in the United States and Europe, there were experiments with "Hydrous Ethanol" (AEHC), a renewable fuel produced from sugar cane wine fermentation. The product obtained through this process is a hydro-alcoholic blend, with minimum graduation of 92,6º INPM. The success of this project, whose scale was unknown, showed the technical viability of the use of this fuel in vehicles. However, the international commerce of products derived from sugar cane made impracticable the alcohol production in Brazil, and this project was abandoned for economic and market reasons. Presently, as an alternative to AEHC produced in Brazil when the international market is unfavorable, by enhancing marketing appeals so that consumers may feel free to choose the fuel for their vehicles, the flex fuel vehicles have been launched. These vehicles can run at any proportions of AEHC and/or "Type C Automotive Gasoline (GASC)". The latter is a blend of unleaded gasoline and "Absolute Ethanol (AEAC)", in fractions ranging from 20%vol. to 25%vol.

In the Flex Fuel engines used in Brazil, both the mixture formation and the combustion process undergo significant changes due to differences in the physical properties of the fuel, depending on its formulation. The vapor pressure of ethanol is generally lower than the vapor pressure of the typical gasoline. However, additions of this oxygenated to this type of gasoline considerably increase its vapor pressure when 10% to 20%vol. is blended to fuel, and diminish in contents higher than those ones. In Brazilian Flex Fuel engines, the best condition of vaporization must be obtained when they operate with GASC. Yet, greater vaporization enthalpy of AEHC causes both increased cooling of the charge and increased volumetric efficiency of the engine. On the other hand, this characteristic, together with its low vapor pressure, may hinder evaporation and contribute towards fuel condensation on the walls of the admission runners.

The flex fuel engines are very similar to the conventional engines. Few external differences can be visualized, which may be unnoticed by a casual observer. The only external characteristic that can be perceived is the cold-start system annexed together with admission runners and necessary when external temperatures are very low, due to raised boiling point of AEHC. The compression ratio of flex fuel engines are slightly higher than the compression ratios of conventional engines and lower than in AEHC engines, to avoid knock when they operate with GASC and torque is not lost when they operate with AEHC, promoting a steady running for any fuel blend. The solenoid injector had its outflows increased up to 30%, and the injection pressure was increased in 0.5 bars. This is necessary to supply AEHC in available times when running in Wide Open Throttle regimen (WOT - full load). In fact, this is needed because the air/fuel stoichiometric rate of AEHC ($\approx$ 8.5:1) is lower than the same relation for GASC ($\approx$ 14:1).

From the 60's and 70's, when this type of engine was introduced, the first concerns about the waste of oil reserves and the damages that their use was causing to the environment, a number of alternatives have been proposed. Some of them, with more immediate consequences and implantations were adopted and they presented satisfactory results for that short period. The option for engines with fuel injection and management control in closed-loop, as substitutes for carbureted engines, and the reformulation of the automotive gasoline with oxygenates and other hydrocarbons as substitutes for lead were the most common solutions. On the other hand, numerous programs of alternative fuels have been tested and the most popular ones are those that promote the use of biofuels. In Brazil, a program for production of ethanol from the sugar cane presented great success in the 80's. However, the impossibility of the engines to operate with another fuel caused a collapse in that project since there was no guarantee that the fuel would be available. Besides that, there was a strong international commerce competition for products derived from sugar cane. Hence, adopted in Brazil from 2003, the new technology FLEX FUEL made the engines flexible, freeing consumers from the market uncertainties and giving Brazilian ethanol a new breath.

2. STATE OF THE ART

In accordance with CFDC (2006), alternative fuels are those that can be used in vehicles with engines of internal combustion and that are not typical automotive gasoline or diesel oil. Literature has presented methanol, ethanol, natural gas, LPG, hydrogen and liquid fuels derived from mineral coal as examples of these fuels. There are several reasons to do researches about alternative fuels, such as economic, environmental and health issues. In the context of alternative fuels, the biofuels have been presented as the strongest substitutes for conventional fuels in short-term. According to IEA (2006), biofuels are liquids or gases derived from biological resources such as cereals, wood or vegetable oils. Among other biofuels, ethanol is the most promising and popular fuel. This oxygenated fuel has been used blended with automotive gasoline for octane improvement; this option is due both to prohibitions of use of leaded gasoline and discouragement by MTBE for its controversial polluting power.

According to MacLean and Lave (2003), ethanol has high octane number, great inflammability band, greater flame speed and greater enthalpy of vaporization in comparison to conventional automotive gasoline. These characteristics allow for greater compression ratio, minimum combustion time and both cleaner and more efficient burning. The disadvantages of this fuel, according to these authors, are its low heating value, high corrosive action, water miscibility, and low vapor pressure, which make more difficult the cold start. However, gasoline and ethanol blends in small fractions improve volatility to fuel, also increasing vapor pressure and evaporative emissions.

Al-Hasan (2003) has investigated the effect of blends of automotive gasoline and ethanol with 99% of pureness in engine performance and emissions. The researcher has claimed that using 0%vol. to 25%vol. ethanol has improved performance, with brake torque up to 8,3%, thermal efficiency up to 9% in, and fuel consumption up to 5.7%. These favorable results were measured with 20% ethanol in the blend, and the engine did not present any operation problem caused by the addition of this fuel.

Wu et al. (2004) have studied several relative air/fuel ratios to verify their influence on performance and emissions of an MPFI engine fuelled by automotive gasoline and ethanol blends. Their tests were carried out in two speeds (3000 min^{-1} and 4000 min^{-1}), partial load and full load regimens, and with ethanol additions of 0%vol. until 30%vol. Regarding the brake torque, they have reported that the full load, for any relative air/fuel ratio and all speeds tested, the difference was so small that it was not conclusive for any ethanol addition. The authors have also claimed that no differences were noticed in brake torque by varying the ethanol content in the fuel blend, since for the same throttling valve opening and the same speed, the admitted air volume is constant, but bigger fuel consumption takes place when the fuel blend contains ethanol, compensating its minor heat value.

Hsieh et al. (2002) have also studied the behavior of an MPFI engine operating with gasoline and up to 30%vol. of ethanol blends in partial loads and full load regimen. Their results have shown little difference in brake torque and full load to 3000 min^{-1} and 2000 min^{-1} when the engine is fuelled by gasoline and ethanol blends.

Yücesco et al. (2006) have investigated the performance of MPFI engines operating with gasoline and ethanol with 99,5% pureness. They have reported that, in their experiments, the maximum brake torque (MBT) was obtained with the same ignition timing for all the fuel blends tested and that the addition of oxygenated increased brake torque and BSFC for all the ignition timings tested. They have also reported knock occurrence when ignition timing was 24º BTDC with unleaded gasoline without ethanol in the blend, what did not happen when ignition timing was up to 36º BTDC in blends containing 60% or 40% of ethanol. When they analyzed the relative air/fuel ratio influence, the maximum "MBT" brake torque occurred for λ = 0.9 for all fuel blends, and brake torque was greater in both the poor and rich regions, in relation to the stoichiometric one for any blend with ethanol. BSFC was minimum for λ = 1.05, and presented 30% increases for fuel blends with 60%vol. of ethanol in relation to gasoline without oxygenate addition. Finally, the authors have reported that the greatest brake torque occurred for 13:1 compression ratio and ethanol content of 40%vol. and 60%vol. in the fuel blend.

Taylor et al. (1996) have evaluated performance in terms of brake torque and brake specific fuel consumption, quality of combustion through parameters like combustion angle, induction timing and intensity of the pressure peak in combustion chamber, and emissions. They have concluded that, in MBT, brake torque did not vary more than 0.4% and 1.3% for changes of 0.14 points in relative air/fuel ratio and 8º in ignition timing, respectively. They have also reported that the pressure peak at combustion chamber linearly increases with ignition timing, but the combustion stability, evaluated through standard deviations of the pressure lines of cycles, presented its best results for ignition timing equal to the MBT ones and it was unfavorably affected when longer ignition timings were used. Regarding combustion, the authors have stated that the induction timing, an indirect measure of combustion delay, increases with the advance of ignition timing due to lower mixture temperatures at the moment of ignition. They have also claimed that the combustion period diminishes with the advance of ignition timing due to higher temperatures and pressures obtained in these combustion conditions. However, both diminish with the mixture enrichment.

In Brazil, absolute ethanol (AEAC), as well as hydrous ethanol (AEHC), has been used as additive to automotive gasoline (GASC). AEHC has application in vehicles with engines adapted to the use of this fuel or in modern flex fuel vehicles, which can operate with any mixture of GASC and AEHC. AEHC is produced in Brazil through sugar cane wine distillation; sugar cane is quite well adapted to the climate in the Brazilian southeastern region. The fuel obtained through this distillation is the hydrous ethanol, with minimum alcoholic degree equal to 92,6º INPM.

The motivation for the use of this fuel in Brazil was to offer an answer to oil crises in the 70's and 80's. In 1986, 76% of the new vehicles commercialized were fuelled by hydrous ethanol, and a larger distribution structure was developed in that country. In 2002, only 3% of the new vehicles commercialized were fuelled by AEHC, and in 2003, the so-called flex fuel vehicles were launched with great acceptance in the market. According to Manzini (2005), these vehicles can be supplied with both AEHC and GASC in any ratios. Table 1 shows the specifications of ethanol (either hydrous or absolute) in accordance with ANP (2005).

Table 1. Specifications of hydrous ethanol (AEHC) and absolute ethanol (AEAC) commercialized by diverse economic agents in the entire domestic territory (ANP, 2005).

CHARACTERISTIC	UNIT	SPECIFICATIONS		METHOD ABNT/ASTM
		AEAC	AEHC	
Aspect	—	Limpid and exempt of impurities	Limpid and exempt of impurities	Visual
Color	—	Colorless	Colorless	Visual
Specific mass 20ºC	kg/m³	791.5 máx.	807.6 a 811.0	5992/D 4052
Alcohol degree	ºINPM	99.3 mín.	92,6 a 93.8	5992

In accordance with *Agência Nacional do Petróleo, Gás Natural e Biocombustíveis*, a Brazilian agency of the Ministry of Mines and Energy of the Brazilian Government, whose function is to promote the regulation, contracts and surveillance of economic activities of the oil industry, automotive gasoline is any appropriate fuel to power spark ignition, internal combustion engines in passengers vehicles, utilitarian vehicles, light vehicles, agricultural equipment and boats. These gasoline, produced in Brazil or imported, are classified as Type A, and are exempt of oxygenates or lead. They are used as fuel base to reformulation of unleaded gasoline and they are not available for the end consumer. Gasoline classified as Type B is the one reformulated from automotive gasoline Type A, and its oxygenating agent is MTBE. It is used only in the south region of Brazil, but it is currently forbidden.

The automotive gasoline classified as Type C is the reformulated from Type A automotive gasoline; its oxygenating agent is AEAC, in contents varying from 20%vol. to 25%vol. in accordance to Brazilian legislation. At the time this study was carried out, this percentage was established as 20%, with permissible margin of error of ± 1%. The octane property of this fuel is equal to its American and European equivalents, with antiknock index ≥ 87. The specifications of the Type C automotive gasoline are presented in Table 2.

Table 2. Specifications for commercialization of Type C automotive gasoline in the entire domestic territory (ANP, 2001).

CHARACTERISTIC	UNIT	SPECIFICATIONS	METHOD ABNT	METHOD ASTM
Color	—	Colorless or yellowish	—	—
Aspect	—	Limpid and exempt of impurities	NBR 7148	ASTM D 1298
Absolute ethanol	%vol	20 a 25	NBR 13992	
Specific mass 20ºC	kg/m3	To write down only	NBR 7148 NBR 14065	D 1298 D 4052
Antiknock Index – IAD, min.	—	87.0	MB 457	D 2699 D 2700

3. MATERIAL AND METHODS

The MPFI engine used in this work is an alternative, propeller with regimen of work per cycle Otto for use in light vehicles. It is a project of four cylinders in-line and cross-scavenged porting of 16 valves over head. It has spark ignition and mixture preparation by fuel injection. It was originally developed to use Type C automotive gasoline; it has 9.5:1 compression ratio and open chamber. The method of load control is throttling and cooling system by water.

Table 3. Engine technical review.

Number of cylinders	4
Number of valves	16
Bore x Stroke (m)	8,05E-2 x 7,84E-2
Bore/Stroke ratio (m/m)	1,0268
Swept volume/ Total volume (m³)	3,99E-4/1,596E-3
Compression ratio	9,5:1
Maximum Power (NBR 1585) (kW/min^{-1})	78/5500 (GASC)
Maximum Brake Torque (NBR 1585) (N.m/min^{-1})	151,1/4500 (GASC)
Method of mixture preparation	Multi point fuel injection
Ignition	Electronic
Fire order	1 – 3 – 4 – 2

The fuel injection and ignition timing management was performed by an Electromotive TEC-II electronic control unit, composed of an electronic ignition system and an electronic fuel injection system. Input signs from a set of sensors distributed over the engine feed the ECU. The ECU manages the actuators, i.e. fuel bomb, ignition bobbins, idle actuator and fuel solenoid injectors, and allows its use as data-logger.

The fuels used are commercialized in Brazil, in accordance with the specifications of ANP (2001) and ANP (2005), i.e. hydrous ethanol (AEHC) and Type C automotive gasoline (GASC). They were used either in their natural commercialization form or in blend form, in order to investigate the flex fuel technology. Those blends have been denominated as “Flex Blends” in this work. The quality of these fuels acquired in service stations was evaluated through the ANP 3/2000 Technical Regulation (ANP, 2000) to certify the AEAC fraction in Type C automotive gasoline, INPM degree of AEHC, as well as their densities to 20^{o}C, color and aspect. The compositions of flex blends used are presented in Table 4.

Table 4. Formulation of Flex Blends investigated.

FLEX BLEND	**FORMULATION**
GASC	100% GASC + 0% AEHC
AEHC25	75% GASC + 25% AEHC
AEHC50	50% GASC + 50% AEHC
AEHC75	25% GASC + 75% AEHC
AEHC	0% GASC + 100% AEHC

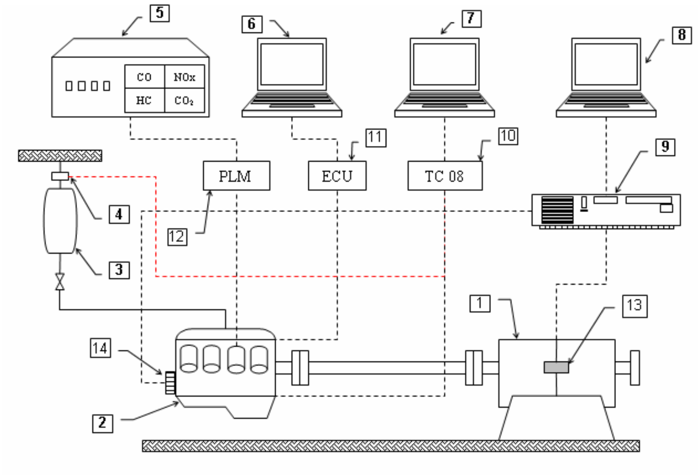
The tests were carried out in accordance with NBR ISO 1585 (ABNT, 1996), whose objective is to specify the method of engine tests for vehicles at full load regimen and as a function of speed for brake power evaluation. Data were collected during only one minute when brake torque, speed, fuel consumption, temperatures, and exhaust emissions remained substantially constant. Data represent the engine performances during approximately 1250 cycles and are the average of consecutive and stabilized readings during one minute for three tests, in three ignition timings around the MBT advance, and one test for all the other ignition timings. The tests were carried out with ignition timings from 10°BTDC to 40°BTDC with steps of 5° and relative air/fuel ratio from $\lambda = 0.8$ to $\lambda = 1.0$, with intervals of 0.1, and speed equals to 2500 min^{-1}. Fig. 1 presents the diagram of both the test bed and the instrumentation used.

The performance investigation of flex fuel engine in terms of brake torque was carried out for diverse flex blends and operational parameters. This parameter represents the engine’s capability to produce work and it is measured by dynamometers connected to the crank of the engine. The equation that determines the brake torque in dynamometer can be written as:

$$T_b = P \times b \tag{1}$$

Where,

T_b – Brake torque (N.m);
P – Force measured by load cell (N);
b – Toggle arm of brake length (m).



1. Electric brake
2. Engine
3. Tank of fuel
4. Load cell
5. Gás Analyzer Multigas 488 Tecnotest
6. Desktop ECU TEC-II Electromotive
7. Desktop – Fuel consumption and temperatures
8. Desktop – Brake torque and velocity
9. Conditioning circuit device to thermocouple and load cell
10. ECU
11. Professional Lambda Meter Motec
12. Load cell Z SV 100 Alfa Instruments
13. Pick-up sensor and trigger wheel

Figure 1. Project of test bed.

The experimental error of the brake torque was determined by the partial derivatives of Eq. (1) as a function of *P* and *b* variables in accordance with the Method of Kline and McClintok (1953). The calibration of the electric signal acquired by the load cell allowed for the determination of the developed brake torque, with experimental error of ± 0.52 N.m. The measurement of engine speed was performed by conversion of sinusoidal signal generated by the magnetic sensor to linear analogical signal calibrated as a function of the rotation. A frequency converter was used for this function. The linear signal was simultaneously acquired with the electric signal of brake torque through the same electronic data acquisition system, with an experimental error of ± 10 min^{-1}.

Fuel consumption was checked by gravimetrical method using a load cell calibrated and submitted to the weight of the fuel reservoir. The experimental error was ± 6g. Absolute consumption was determined through the relation between the variation of electric signal generated in the load cell and the time waste by the test, in accordance with Eq. (2), and was registered by a data acquisition system.

$$\dot{m}_f = \frac{m}{t} \tag{2}$$

Where,
$\dot{m}_f$ – Fuel consumption (g/s);
m – Absolute consumption (g);
t – Time waste to test (s).

The brake specific fuel consumption (BSFC) represents the efficiency with which the engine converts the fuel chemical energy into mechanical work. It is obtained through the fuel consumption and brake power ratio:

$$BSFC = \frac{\dot{m}_f \times 3{,}6}{P_b} \tag{3}$$

Where,
BSFC – Brake specific fuel consumption (kg/kW.h);
Pb – Brake power (kW).

Brake power represents the rate of work in one unit of time in dynamometer and is obtained through the product of brake torque and angular speed of the engine:

$$P_b = \frac{T_b \times \omega}{10^3} = \frac{T_b \times 2 \times \pi \times N}{10^3} \tag{4}$$

Where,
ω – Angular speed (rad.s^{-1});
N – RPM (s^{-1}).

4. RESULTS AND DISCUSSIONS

The ignition timing is an operational parameter that has significant effect on the performance of an MPFI engine. According to Heywood (1988), in steady speed regimen and for the same relative air/fuel ratio, there will be, for each engine and each fuel, an ignition timing that will produce maximum brake torque. The behavior of the engine, investigated in terms of brake torque, full load and steady speed regimen, as a function of ignition timing, relative air/fuel ratio and AEHC contents in flex blends, is presented in Fig. 2. Severe influence of the ignition timing on brake torque is noticed, and maximum performance data were obtained with ignition timing equal 20º BTDC for all the flex blends examined. This behavior was also noticed for all the relative air/fuel ratios investigated, and it is related to both the combustion period and the flame propagation. This behavior also indicates that processes of ignition beginning around 20º BTDC theoretically produce a chamber pressure peak approximately about 15º ATDC, resulting in maximum torque for these ignition timings.

When ignition timing is minor than 20°BTDC, it is possible to notice that brake torque diminishes with the ignition timing delayed. For these advances, the pressure peak is displaced for angular positions bigger than the 15°BTDC recommended, thus diminishing its intensity and net work per cycle. When ignition timing is bigger than MBT timing, it is also possible to observe reduction of brake torque. The cause of this effect is the increased negative work over piston at the end of compression stroke. Knock for GASC operations with higher than 30°BTDC ignition timings was also observed.

Yücesu et al. (2006) carried out similar experiments dealing with engines and fuel blends. In distinct load regimen, blends of gasoline and AEAC from 0%vol. to 60%vol, they have reported that the MBT, for speed of 2000 min^{-1}, 10:1 compression ratio e stoichiometric relative air/fuel ratio, also occurred for the same ignition timing for all the blends examined, which were equal to 22º ADTC in that case. These reported behaviors have evidenced little difference in the speed of flame propagation among those blends, and no differences were noticed in MBT ignition timing in operation for different GASC and AEHC blends. However, Taylor et al. (1996), in similar experiments, have pointed out the necessity of advancing the ignition timing to get MBT when the engine operated with ethanol, in relation to automotive gasoline. This behavior can also be suggested by Fig. 2, when it is observed for the largest content of AEHC in the blend.

Relative air/fuel ratio is also an operational parameter that imposes strong influence on the brake torque. In this experiment, maximum brake torque took place for $\lambda = 0.9$ for all the flex blends evaluated. The same behavior was also verified for others ignition timings investigated. These findings are similar to those found by Wu et al. (2004), who have investigated the influence of relative air/fuel ratio on engine performance. These authors reported that MBT was checked in all the conditions investigated for lightly rich relative air/fuel ratio and equal to $\lambda = 0.95$. Guibet (1999) has justified this behavior of multiple-cylinder engines claiming that since the relative air/fuel ratio is measured at downstream of the escape runner, it represents an average ratio between all the cylinders. This means that one or more cylinders are operating with stoichiometric ratio and developing its maximum capacity of work in these conditions.

Then, by admitting that the operational conditions of relative air/fuel ratio and ignition timing to MBT of this experiment are 20º BTDC e $\lambda = 0.9$, the behavior of the brake torque for these operational conditions regarding AEHC content in flex blends and the curve extracted by simulation are presented in Fig. 3. On analyzing this figure, it is possible to observe that the MBT behavior, in the conditions of investigation, presents little variation with AEHC addition in flex blend and, therefore, it is practically a horizontal line. It is possible to affirm that brake torque is not significantly influenced when AEHC content is increased in flex blend. Wu et al. (2004) have reported, however, a small improvement in the performance with ethanol addition only when the engine operated at partial load regimen with throttling openings into 60%; they did not notice any improvement in full load regimen (WOT).

Al-Hasan (2003) has also investigated the effect of ethanol addition up to 25%vol. on the brake torque and reported improvement of this parameter when ethanol content is increased from 0%vol. to 20%vol. in fuel blend where the engine presented maximum performance – the latter blend would correspond to GASC used in this work. From this content up to 25%vol., the author has noticed a slight tendency towards brake torque loss in all the examined conditions. Qualitatively, the behavior described by this author has also been found in the indicated torque curve obtained through simulation, whose behavior presents maximum performance for GASC (20%vol. AEAC) and decreases when AEAC content increases in the fuel blend. This occurs, in the simulated case, because of lesser energy availability in the fuel blend, and increased fuel consumption is necessary. There was a quantitative difference between measured (brake) and simulated (indicated) values because, in the simulation, the adopted polytrophic coefficient admitted are typical values presented by literature and was kept constant for all the conditions and during the cycle. However, in the real case, the environment of reaction characterized by this coefficient varies according to the fuel, relative air/fuel ratio and ignition timing, as well as to the compression stroke (fresh mixture) and the expansion stroke (burned gases), what justifies this difference in behavior.

By using simulation, it is also possible to observe that net work per cycle is the maximum when the engine is fuelled by GASC, and is minimum when the engine is fuelled by AEHC. However, these differences are relatively small. It is possible to state that maximum pressures in combustion chamber and flame temperatures diminish with AEHC addition to the fuel blend, as it can be seen in Table 5. This conclusion is very important, because, in experimental measurements, for any addition of AEHC to the fuel blend, considering the worst possibilities, the engine performance concerning brake torque is equal to the engine performance with GASC. This difference between ideal and real cases can be explained, because, in the former, the inefficiencies generated by energy losses in the chamber walls are not considered, therefore, the cycle processes are assumed as adiabatic (Heywood, 1988). In a real situation, losses are inevitable caused, and there is an intense exchange by convection and radiation between the burned gases and the combustion chamber walls. In operations with AEHC, the greater water presence, together with combustion products, causes reduction of the escape temperature and increases the specific heat (Taylor et al., 1996); the lower flame temperature of this fuel (Yüceso et al., 2006) turns those losses even lesser, thus justifying the performance similarity between this fuel and GASC in real cases.

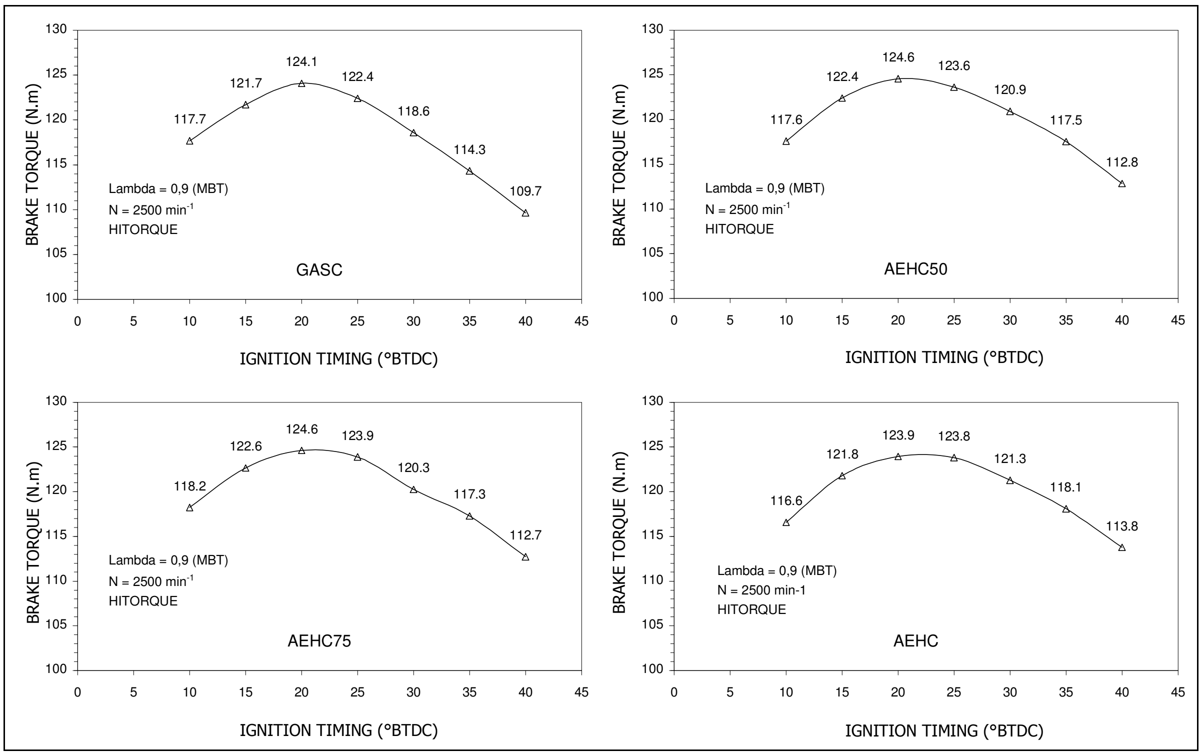


Figure 2. Behavior of brake torque as a function of ignition timing at °BTDC. $\lambda = 0,9$.
Maximum error of measurement ± 0,52 Nm.

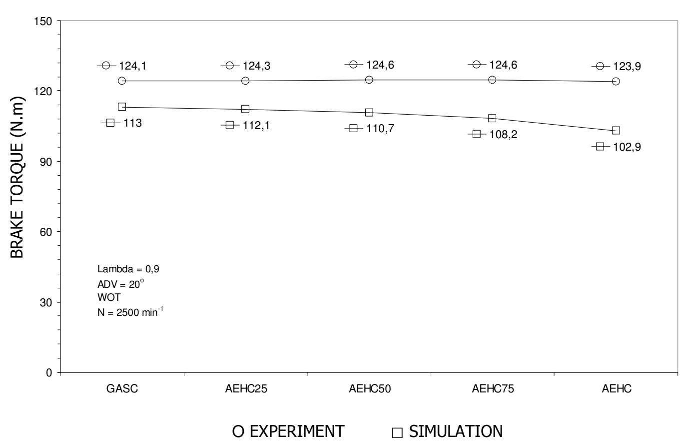


Figure 3. Behavior of brake torque as a function of flex blends composition. Maximum error of measurement ± 0,52 Nm.

Table 5. Heating value, combustion temperatures and indicated performance parameters obtained by simulation for investigated flex blends.

	GASC	AEHC25	AEHC50	AEHC75	AEHC
Heating value (kJ)	40.900	38.800	35.900	31.700	25.100
T3a (°C)	1.150	1.142	1.130	1.109	1.063
T3b (°C)	2.178	2.164	2.143	2.107	2.028
Indicated torque (N.m)	113,0	112,1	110,7	108,2	102,9
Indicated power (kW)	29,58	29,34	28,96	28,31	26,91
Indicated specific fuel consumption (kg/kW.h)	0,318	0,358	0,401	0,449	0,513

The literature has pointed out the possibility of higher flame propagation speed of ethanol in relation to gasoline, which could favor fast combustions and little time for energy losses. However, according to Fig. 2, in the cases investigated here, the difference of the flame propagation speed is very small, because the MBT ignition timings measured are relatively equal, which also indicates similar flame propagation.

The use of ethanol could improve the engine efficiency, since its antiknock characteristic is superior in comparison to automotive gasoline, allowing for engines conceived for higher compression ratio. Yüceso et al. (2006) have investigated the influence of compression ratio on performance and reported that greater brake torque was obtained when higher compression ratio and larger AEAC content were used. Fig. 4 represents the Diagram $p \times V$ of the engine operation cycle obtained through simulation and fuelled by AEHC with 9.5:1 and 14.5:1 compression ratios; the first one is the rate of the engine investigated and the last one is slightly higher than the compression ratios of MPFI engines fuelled by AEHC commercialized in Brazil. In this figure, it is possible to notice that the closed area of the diagram is larger for the higher compression ratio, indicating that the largest net work indicated per cycle is obtained with 14.5:1 compression ratio. This has enabled us to affirm, according to what is seen in Table 6, that, theoretically, there has been little or no advantage concerning net work in engines with low compression ratios, fuelled by AEHC, as it is the case of Brazilian Flex Fuel engines. Besides that, this simulated 14.5:1 increase in compression ratio allows for a 12.7% improvement in terms of power developed by this engine.

Table 6. Indicated performance parameters obtained by simulation for AEHC e rc = 9.5:1 e rc = 14.5:1.

COMPRESSION RATIO	9,5:1	14,5:1	%
Power indicated (kW)	26,91	30,34	
Torque indicated (N.m)	102,9	116,0	+12,7%
Net work per cycle (kJ)	3,23E-4	3,641 E-4	

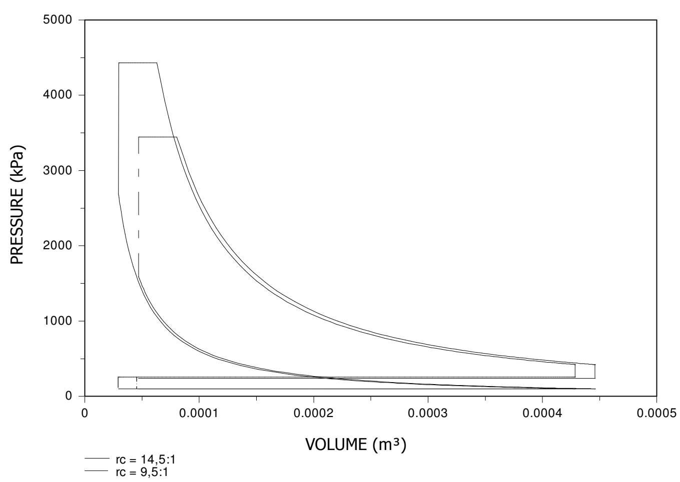


Figure 4. Pressure-volume diagram of ideal limited pressure cycle obtained by simulation for AEHC, $\lambda = 0.9$, 2500 min^{-1}, 20°BTDC, rc = 9.5:1 e rc = 14.5:1.

The main disadvantage of the engines fuelled by ethanol, given its lesser stoichiometric air/ fuel ratio, is the increased fuel consumption needed to keep the relative air/fuel ratio between the limits required. In this work, the parameter brake specific fuel consumption (BSFC) was used to measure the engine performance when AEHC was blended with fuel. In the Brazilian Flex Fuel policy, increased fuel consumption, when an engine is fuelled by AEHC blends in comparison to GASC, is particularly serious, since no significant changes in work produced with change of fuel composition have been found.

Brake specific fuel consumption (BSFC) can be understood as a performance parameter that measures the necessity of fuel mass of an engine to produce one unit of mechanical work. The engine behavior at full load regimen investigated in relation to BSFC as a function of relative air/fuel ratio and all the flex blends examined is presented in Fig. 5 at 2500 min^{-1} and 20ºBTDC. It is possible to observe that BSFC is inversely proportional to relative air/fuel ratio and directly proportional to AEHC additions in the investigated region. The greatest increase of BSFC when engine is fuelled by AEHC in relation to operations with GASC occurred for $\lambda = 1.0$ and the smallest for $\lambda = 0.8$; they represent, respectively, increases up to 51,4% and 40,9%, indicating that BSFC increase for AEHC was more significant in the stoichiometric region than in the rich region. This has allowed us to affirm that, by adding AEHC to the fuel, the engine needs a greater fuel mass to produce the same work, especially in the stoichiometric region. Yüceso et al. (2006) have reported similar behaviors in percentile terms for BSFC when investigated as a function of relative air/fuel ratio and AEAC addition to the fuel blend. However, pureness of ethanol used in those mixtures was of 99,5%, different from the AEHC (95,4 ºINPM) used in this work. The same can be said regarding results in terms of fuel consumption found by Al-Hasan (2003) when AEAC content was increased in the fuel blend.

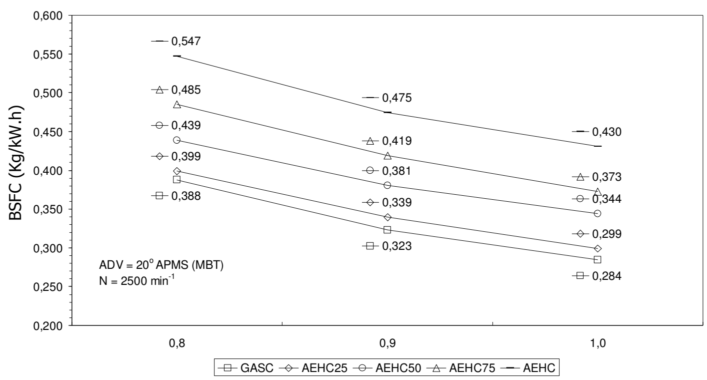


Figure 5. Behavior of BSFC as a function to relative air/fuel ratio and AEHC content.
Maximum error of measurement ± 0.01 kg/kW.h.

5. CONCLUSIONS

From the results presented and discussed, it is possible to draw the following conclusions:

The brake torque showed to be strongly influenced by both the ignition timing and relative air/fuel rate; the maximum performances found were $\lambda = 0.9$ and 20°BTDC, independently on the AEHC content in the composition of the fuel blends considered here. It is possible to conclude from the results displayed that the flame propagation is relatively similar in all the flex blends examined. We have also found that the best engine performances have not presented quantitative differences when the engine is fuelled by either GASC, AEHC or any blend of these fuels; thus, the option for a determined blend presents little influence either on net work, net power or brake torque. However, in the simulation, it has been noticed that net work per cycle can significantly increase when the engine is fuelled by AEHC, in these same conditions, but only if the engine has its compression ratio increased as well. This is perfectly possible for this fuel, by its better antiknock characteristics, and a 12.7% increase in brake torque for 14.5:1 compression ratio was obtained, compared to 102,9 N.m obtained in simulation for 9.5:1 compression ratio. These results have reaffirmed the knowledge that there will be significant technical advantage of AEHC as a substitute for GASC in terms of net work if compression ratio is increased.

In terms of fuel consumption, BSFC was evaluated (kg/kW.h). The BSFC behavior has shown positive correlation with AEHC addition in flex blends. This indicates that the Brazilian flex fuel engines, in these conditions of regimen, need to burn greater fuel mass to produce the same work when they operate with increasing additions of AEHC in fuel; these increases varied from 41% to 51%, concerning GASC operations. We have also concluded that fuel consumption increase is proportional to relative air/fuel ratio, being greater when the mixture is richer. We have noticed no significant behavior changes in this operational parameter as a function of relative air/fuel ratio and composition of flex blends, that is, the increased fuel consumption rate as a function of the relative air/fuel ratio is practically constant for all flex blends investigated and was around 14% in these experiments, for steps of 0.1 in band of evaluated relative air/fuel ratio.

Finally, it is possible to conclude, from all the aspects analyzed, that the Brazilian flex fuel engine fuelled by Type C automotive gasoline, hydrous ethanol or blends of these fuels, presents the best performances with the first fuel. We have noticed that AEHC fuelling did not present significant differences in terms of net work, however, it presented a great disadvantage in terms of fuel consumption. This parameter of performance increases around 10% in mass for each 25% in volume of AEHC added to flex blends, reaching, in the worst results, 51% of increase for unit of work produced. For these reasons, the consumer option for AEHC fuelling in flex fuel vehicle is only restricted by economic criteria, as it has not been possible to show, in this work, technical superiority of this fuel in comparison to GASC. From this criterion, the economic viability of AEHC is compensatory when the price of a liter of AEHC is 66% lower than the price charged for a liter of GASC, that is, in accordance with the following relation and considering the worst condition to BSFC (51%) verified in this work:

$$\frac{price_{AEHC}}{price_{GASC}} < \frac{1}{1{,}51} < 0{,}66$$

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5. RESPONSIBILITY NOTICE

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