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EVALUATION OF THE POSSIBLE BENEFITS OF DEVELOPING ENGINES THAT USE ONLY ETHANOL AS FUEL

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Abstract. *Over the years, the automotive industry has faced increasing restrictions related to pollutant emissions generated by the combustion of fossil fuels. In order to try to reduce the environmental impact, some strategies have been proposed, including the use of biofuels and the use of electric and hybrid motors instead of internal combustion engines. Regarding electric and hybrid motors, however, there are still great doubts about the environmental impacts that the production of these new propulsion systems would generate, and what would be the amount of electric energy needed to maintain the current vehicle fleet within Brazil. In the case of renewable fuels, mainly ethanol, these have been used and studied for a long time in Brazil, and this alternative seems to have a smoother transition for both industry and the market. However, with the advent of flex engines, the development of internal combustion engines that operated exclusively with ethanol has been halted in Brazil, and the maximum potential of this fuel is not being used. This work presents a study considering what are the possible benefits to be achieved with the development of an internal combustion engine that uses only ethanol as fuel. The results obtained show that this type of approach would be able to provide an average increase in the torque of the current engines by up to 5.36%, or a reduction in the net fuel consumption of up to 5.6%, if the torque and power were maintained at the current levels and the total engine displacement was reduced.*

Keywords: *Ethanol, Knocking, Compression Ratio, piston engine, engine*

1. INTRODUCTION

Biofuels may reduce the greenhouse gas emissions and pollutions in cities, and part of this pollution is caused by vehicles that have internal combustion engines as the propulsion system, some international organizations began to impose restrictions on the levels of emissions generated by the operation of cars, buses and trucks (Ministério do Meio Ambiente, 2020).

Within this scenario, the automotive industry has started to develop strategies to meet the requirements. Initially, the strategies involving modification in engines operational parameters or adaptations in the projects are used these may be enough to meet the requirements. However, new techniques needed to be developed in order to be able to meet the established limits. Currently, mainly regarding heavy-duty vehicles, entire systems are necessary to control emissions, such as, for example, the exhaust gas recirculation systems (EGR).

Currently, new strategies are being developed, such as the introduction of hybrid and electric vehicles in the world vehicle fleet, and also the use of biofuels instead of fossil fuels. The first alternative has great strength mainly in Europe, with a series of tax incentives for the use of electric vehicles in almost all European countries. These tax incentives includes the subsidies given by Sweden (60,000 Kronor), the United Kingdom (3,000 British Pound) and Ireland (5,000 British Pound) (Ribeiro, 2018).

However, this approach still faces a series of barriers that make the option of replacing internal combustion engines by electric motors not unanimous. The related problems and this solution involve both economic and environmental aspects. Analyzing the American scenario, according to Kintner-Meyer et al. (2007), if the fleet of cars, pickup trucks and SUV's were replaced, the North American electric energy system would be able to supply only 84% of all the energy needed to these vehicles. In the same article, the authors stated that, if the analysis extended to all light vehicles the percentage would drop to 73%. This study shows that the electricity generated is not enough to supply the entire fleet of a country of continental dimensions such as the United States. In addition, most part of the energy generated in the American territory comes from sources that are not considered clean, such as, for example, thermoelectric power plants.

In an analysis considering the Danish scenario, Herynkova (2009) classified as "insignificant" the economic benefits that would be generated by electric vehicles, in view of the fiscal waivers necessary to stimulate the internal market to adopt this strategy. The author concludes that the social benefits obtained with electric vehicles would amount to between 17 and 161.1 million euros, while the tax waiver necessary for this would generate a loss between 225.6 million euros and 1.86 billion euros. In other scenarios, such as in Germany, where much of the electricity is generated by coal and oil, the new amount of electricity that would need to be produced could maintain the levels of CO_2 and NO_x emissions and increase the levels of SO_2 emissions by 900% (Wilkins, 1997).

Regarding the Brazilian territory, the alternative to biofuels is apparently more feasible, given that there are vehicles being produced running using ethanol since 1978 (Montoia, 2019). This kind of prior knowledge of the fuel and the fact that its distribution is already fully established makes this type of alternative more attractive. However, this alternative also has its cons, the main one being the fact that the production of these types of fuels requires the use of an extensive agricultural area, which for small countries can generate a major problem related to food supply, since the space occupied will compete with food production.

Despite the benefits in engine performance, the use of ethanol as fuel presents some risk. The ethanol combustion increases the emission of some undesirable species, as formaldehyde (HCHO) and acetaldehyde (CH3CHO), due to the presence of the hydroxyl functional group on the fuel (Zarante and Sodré, 2018). Comparing with the gasoline, the ethanol also has a higher tendency to the pre-ignition, that can be related to surface ignition (Hamilton et al., 2008).

This paper presents a numerical analysis considering the development of an internal combustion engine that use only ethanol as fuel to evaluate the possibilities that can be explored to extracted the greatest potential of this fuel.

The engine used as a reference in the analysis is a four stroke engine, spark ignition of 1,598 cm³ volumetric displacement, 4 cylinders and water cooled. This engine is used in the Logan and Sandero cars, distributed by the Renault manufacturer, in their 2013 versions.

The numerical analyzes presented were developed using a single-zone internal combustion engine simulation code, an in-house code developed at Aeronautics Institute of Technology (ITA, 2020). Differences between single-zone, two-zone and multi-zone models can be found in Caton (2016). Some results obtained by the modifications proposed in this paper, show that it is possible to reach an increase in torque by up to 5.36%, using different Compression Ratios (CR). In other analysis, is shown that changing the CR and the swept volume, in order to mantain the engine performance at the same levels of the experimental data, it is possible to reduce the net fuel consumption by up to 5.6%.

2. FUELS AND THEIR PROPERTIES

In Brazil the main fuels used in spark-ignition internal combustion engines are Brazilian gasoline (E27) and hydrated ethanol (E100). Table 1 presents a comparison between the main properties of pure gasoline and hydrated ethanol.

Table 1. Properties of Gasoline and Ethanol.

	Gasoline	Ethanol
Lower Heating Value (MJ/kg)(1)	44.0	27.0
Air-Fuel Stoichiometric Ratio(1)	14.6	9.0
Energetic Parameter (MJ/kg)(1)	3.01	3.00
AKI(2)	90.0	100.5

(1)Millo (2013)
(2)Mattos (2018)

Looking at Tab. 1 it is possible to verify the differences between the properties of these two fuels. Note that the lower heating value of the gasoline is much greater than that of the ethanol, a difference of 17 MJ/kg, which represents a heating value of 38.6% lower for the ethanol compared to pure gasoline (Millo, 2013). This information is extremely important, however, it can lead to wrong conclusions about the two fuels.

Despite having a much smaller lower heating value, it is important to note that ethanol has oxygen in its molecule. This oxygen, during the combustion process, will be used for the oxidation of hydrocarbons, and therefore a smaller amount of oxidant will be need to be mixed with the fuel in order to have its complete combustion, as can be confirmed by observing the values of Air-Fuel Stoichiometric Ratio of each fuel. For this reason, when the lower heating value is correlated with the stoichiometric relationship, it is noted that the resulting energy parameter for each fuel has very close values, with a slight advantage for the gasoline (E0), as shown in Tab. 1 (Millo, 2013).

However, there is still one last parameter that must be considered in the analysis of these fuels, which is the knocking resistance of each one of them. The knocking resistance of the fuels is generally measured using two different types of scale: the RON (Research Octane Number) and MON (Motor Octane Number) scales; with the AKI (Anti-

Knock Index) parameter, showed in Tab. 1, being the average between RON and MON. It is noted that the knocking resistance basically shows how much the Air-Fuel mixture, in a spark ignition engine, can resist to temperature and pressure conditions, without autoignition start. Thus, the higher the AKI ((RON+MON)/2) value, the greater the knocking resistance of the fuel.

It is known that in an internal combustion engine, the higher its compression ratio (CR), the greater its brake efficiency will also be (Ferguson, 2016), and therefore it is always desired that this value be as high as possible. However, a higher compression ratio will also cause the temperature and pressure conditions inside the cylinder to be more severe, so there will be a greater tendency for knocking to occur.

Note, however, that the knocking resistance of ethanol is greater than that of gasoline, so that it is possible to use higher compression ratios for ethanol than for gasoline. Based on this, and thinking about increasing the knocking resistance of the gasoline, in Brazil, mixtures of gasoline and ethanol were used as fuel, and currently, this mixture has a proportion of 27% of ethanol by volume of gasoline (E27). This type of strategy is still used today, and allows automakers to use higher compression ratios than those that would be supported by gasoline.

This type of strategy allowed manufacturers to start placing vehicles on the market that could use both E27 and E100 (Flex vehicles), however, this type of approach caused the engines designed only for use with Ethanol to be stopped, and the maximum potential of this fuel is no longer explored.

3. KNOCKING

The combustion process in a spark ignition engine can be divided into three phases: in the first phase, there is the development of the first nucleus, which will start combustion; in the second phase, the flame front spreads until it meets the walls. This flame front is turbulent, and the speed with which the flame propagates is proportional to the intensity of the turbulence, which in turn is proportional to the rotation; finally, in the third phase, the fuel oxidation process is completed (Ferrari, 2008).

The first stage of the combustion process is a spark that will start combustion. The moment when this spark is delivered is called the ignition point. However, this spark cannot be provided at any time in the cycle, but in an optimized point, so that the peak pressure reached during the second phase of the event, is present after the Top Dead Center (TDC), in order to produce more work (Heywood, 1988). The ignition point does not only change the crankshaft angle where the maximum pressure will occur, but also its module. Figure 1 shows the pressure behavior of the indicated cycle for different ignition points.

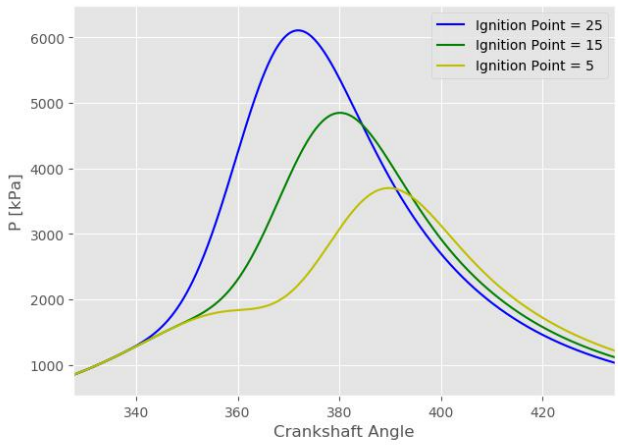

Figure 1. Ignition point effect under pressure behavior of the indicated cycle of an internal combustion engine.

In the same way that the ignition cannot start with a very small advance, impairing the performance of the thermal machine, it also cannot be done far away from the TDC, as it increases the risk of knocking. The latter is probably the most common combustion anomaly, and occurs when part of the air-fuel mixture reaches a certain condition capable of igniting itself. This can occur at more than one point at a time.

It is possible to verify the occurrence or not of knocking through experiments using accelerometers. The characteristics of the signal measured by the accelerometer and the pressure in the combustion chamber, during a cycle with the occurrence of this anomaly, can be seen in Fig. 2 (Ferguson, 2016). It is observed that, when knocking occurs, the pressure wave undergoes a great oscillation. This oscillation will be greater or less depending on the knocking intensity.

The knocking phenomenon is favored with the increase of the ignition advance, since the greater the ignition advance, the greater the pressure conditions, and consequently temperature, inside the cylinder chamber, as shown in Fig. 1. This is explained by the fact that, with the start of combustion, and the increase in pressure and temperature inside the cylinder, the air-fuel mixture that has not yet been reached by the flame front will begin to experience a thermodynamic conditions that can cause the unburned mass to start the oxidation process (combustion) at different points. Each these points that started the combustion process, will give rise to a new flame front. The encounter between different flame fronts inside the cylinder will cause the pressure oscillatory behavior shown in Fig. 2. As with greater ignition advances, more severe conditions are observed in the combustion chamber, the increase of this parameter will be a factor that may facilitate the occurrence of knocking.

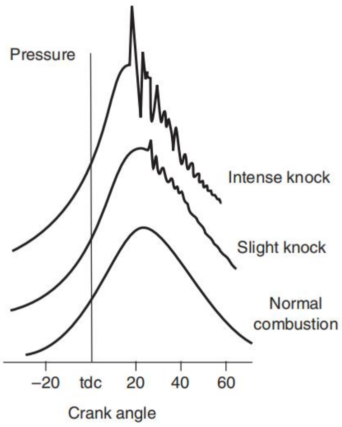

Figure 2. Pressure profiles for knocking conditions (Ferguson, 2016).

The effects of knocking on an internal combustion engine can be disastrous, and it must be avoided as much as possible, once this phenomenon may completely destroy the mechanism if it operates in these conditions for long periods of time.

Thus, it is important to be able to assess the occurrence or not of this anomaly during the spark ignition engines simulations, for that there are a number of methods for this assessment. These methods will be different depending on the type of model that is used to determine the thermodynamic cycle, be it a single-zone model or with more than one zone. According to Moses et al. (1995), any of these models can be used and present excellent results.

The model used in this work is a single-zone model, and therefore, a compatible formulation should be chosen for knocking modeling. The chosen method was developed by Yates and Viljoen (2008), and later underwent some modifications made by Yates et al. (2010), it is based on the calculation of the ignition delay time, which is given by Eq. (1).

$$t_2 = t_1 + \tau_{h,CF}\left(1 - \frac{t_1}{\tau_{h,i}}\right) \quad (1)$$

Where: t_1 is the time relating with the pre-cool-flame phase, $\tau_{h,CF}$ is the auto-ignition delay after the cool-flame appearance and the $\tau_{h,i}$ is the auto-ignition delay related to the initial conditions. These values are calculated using Eqs. (2), (3) and (4).

$$t_1 = \phi^{\beta_1} \cdot A_1 \cdot P^{n_1} \cdot e^{\frac{B_1}{T}} \quad (2)$$

$$\tau_{h,i} = \phi^{\beta_h} \cdot A_h \cdot P_i^{n_h} \cdot e^{\frac{B_h}{T_i}} \quad (3)$$

$$\tau_{h,CF} = \phi^{\beta_h} \cdot A_h \cdot P_{CF}^{n_h} \cdot e^{\frac{B_h}{T_i + X \cdot \Delta T_{CF}}} \quad (4)$$

P, T and ϕ are the pressure, temperature and the equivalence ratio, respectively. The coefficients β, A and n are calibration factors. The subscripts i, CF and 1 are related with the initial conditions, the conditions after the cool flame appearance and pre-cool flame appearance, respectively. The parameter X is a function of the Anti-Knock Index of the fuel. All of these coefficients and parameters are calculated by relations described in the work of Yates and Viljoen (2008) and Yates et al. (2010).

The T_{CF} and ΔT_{CF} values, are calculated by Eqs. (5) and (6), respectively.

$$T_{CF} = T_i + 0.5 \cdot \left(X \cdot \Delta T_{CF} + \sqrt{(X \cdot \Delta T_{CF})^2 + C_0} \right) \tag{5}$$

$$\Delta T_{CF} = \omega \cdot \left(T_i - T_{EQ} \cdot P^{\kappa_{PRESS}} \cdot \phi^{\mu} \cdot \left(\frac{100}{99 + \phi} \right)^{\sigma} \right) \tag{6}$$

Where, ω, T_{EQ} and κ_{PRESS} are functions of the Anti-Knock Index, and C_0, μ and σ are constants. Also, these coefficients and parameters are calculated by relations described in Yates e Viljoen (2008) and Yates et al. (2010).

The result of equation (1) is given in milliseconds, and therefore, its meaning only indicates the minimum time necessary for the mixture under the given temperature and pressure conditions to ignite. However, this result alone does not have much meaning, as spark-ignition internal combustion engines do not work under constant thermodynamic conditions. Thus, the strategy used in this work was to use a minimum time limit, for that knocking does not occur. This time was stipulated based on experimental data available for the analyzed engine, during the calibration phase of the numerical model. An approach similar to this can be seen in Tougri et al. (2016).

4. RESULTS AND DISCUSSION

For a better understanding of the results presented in this work they will be divided into four parts: 1) validate the behavior of the knocking model, analyzing whether it presents a trend of results consistent with the theory; 2) calibration of the model in relation to the experimental data; 3) an analysis is made of the benefits that can be achieved by changing the compression ratio of the analyzed engine, keeping the same displacement volume; 4) the compression rate was modified, however the displacement volume of the engine was varied in order to keep the torque and power parameters equal to experimental data.

4.1 Knocking Model Analysis

The simulations was carried out to evaluate the behavior of the knocking model, considering the variation of the ignition point of the engine. In Fig. 3, in the left graph is shown the behavior of the pressure curve along the cycle, while in the right graph the behavior of time t_2 versus the crank angle, during the combustion phase of the cycle. It was chosen in these study three values of ignition time, ignition advance, 30°, 20° and 10° bTDC (before Top Dead Center). Since t_2 is related to the thermodynamic conditions inside the combustion chamber, when the ignition point is increased, the minimum value of the t_2 become lower and lower, because the maximum pressure in the cycle will be higher. Figure 3 also shows that as the pressure in the cylinder increases, the thermodynamic conditions become more severe and t_2 decreases. This is in agreement with the formulation presented in the previous section.

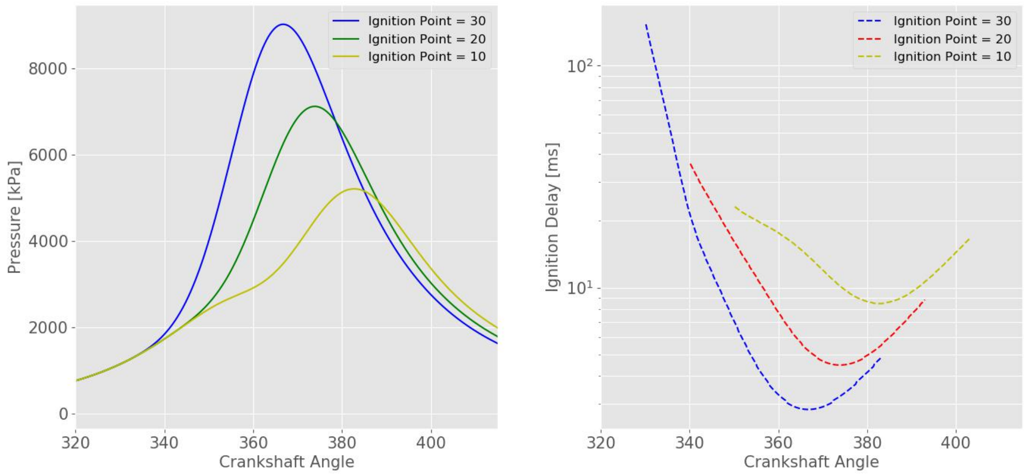

Figure 3. Knocking model behavior analysis.

This behavior is compatible with what was expected (Ferguson, 2016), since under more severe conditions of temperature and pressure, the occurrence of knocking tends to be favored, as previously mentioned. A shorter t_2 means that the air-fuel mixture need to remain at the given thermodynamic conditions for a shorter period until knocking appears.

Although this behavior is consistent with the theory, a more detailed quantitative analysis would require experimental data that are not available to the authors, limiting this analysis to the behavior of the knocking model, which, as shown in the results in Fig. 3, is consistent with theoretical concepts.

4.2 The Engine Modelling

The mathematical model used to represent the studied engine, was carried out in a single-zone internal combustion engine simulation code, an in-house code developed in Python. The torque and power curves provided by the engine manufacturer were used as reference data to validate the code. These two parameters, as well as some geometric parameters were the only information available from the engine. Control parameters, such as, ignition point and equivalence ratio were adjusted to meet the performance information reported by the automaker.

Figure 4 shows the results obtained with the computer code and compared with the experimental data reported by the vehicle manufacturer.

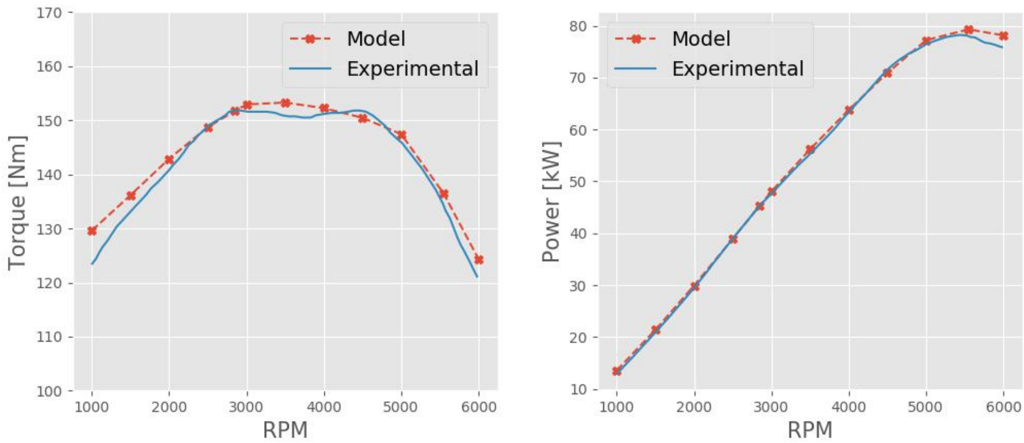

Figure 4. Comparison between the simulated results and the experimental data.

During the model calibration, the adjustment of the equivalence ratio (ϕ) was done initially considering a constant value of 1.1 for the entire rotation range. Unfortunately, at lower speeds (from 1,000 to 1,500 rpm) and for higher speeds (5,500 and 6,000 rpm) this value had to be changed. At lower speed regimes, lower ϕ values were used (1.0), while for the higher speed regimes, richer mixtures were used (1.4). This type of strategy is a good resource that can be used by automakers to control the formation of some type of pollutant.

The average error between the model and the experimental data was 1.53%, analyzing the entire speed range. The highest percentage error was 4.91%, occurring at 1,000 rpm. The authors believe that these are good results, because an error around 5% can be acceptable considering the amount of experimental data available.

Figure 5 shows the behavior of the ignition delay time (t_2) along the speed range. The lowest value of this parameter, which occurs at 6,000 rpm, is 3.51 milliseconds. It is the value that was used as a reference for minimum value that cannot be exceeded, so that knocking does not occur (since the experimental data released by the manufacturer are calibrated to avoid this type of anomaly). In this way, all analyzes made afterwards respect this limit, so that it is not exceeded.

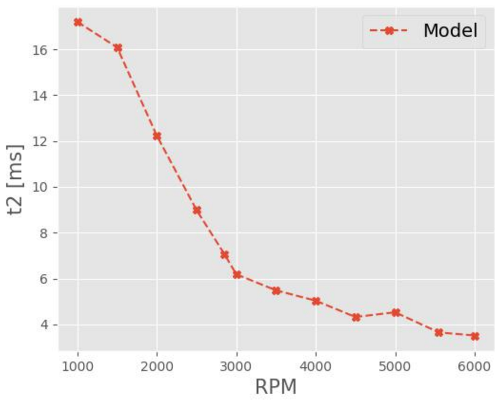

Figure 5. Estimated ignition delay time over the speed range.

In the results presented in Fig. 5 is noted that t_2 increases when the rotational speed changes from 4500 to 5000 rpm. The increase in rotational speed makes parameters related to combustion to be affected, including the combustion duration, which will change the energy release rate. This impact the behavior of the pressure inside de cylinder, which can reach lower maximum values, increasing t_2. For 5500 and 6000 rpm, t_2 decreases too. This happens due to the variation of other engine operational parameters, such as the equivalence ratio.

4.3 Influence of the Compression Ratio

After calibration of the engine model a verification of the performance improvement due to the use of ethanol started, taking as a reference the time limit founded in the previous section. Thus, the compression ratio (CR) was changed to verify its behavior. The simulations were carried out considering the same operational parameters used in the calibrated engine, being modified only when the minimum ignition delay limit is violated.

Figure 6 shows the results obtained with varying compression ratio from 12 to 17, the standard is 12. It is noted that, as the compression ratio increases, there is also an increase in torque and power for the entire speed range, as expected. Considering the compression ratio of 17, the average increase obtained in the torque and power curves was approximately 5.36%.

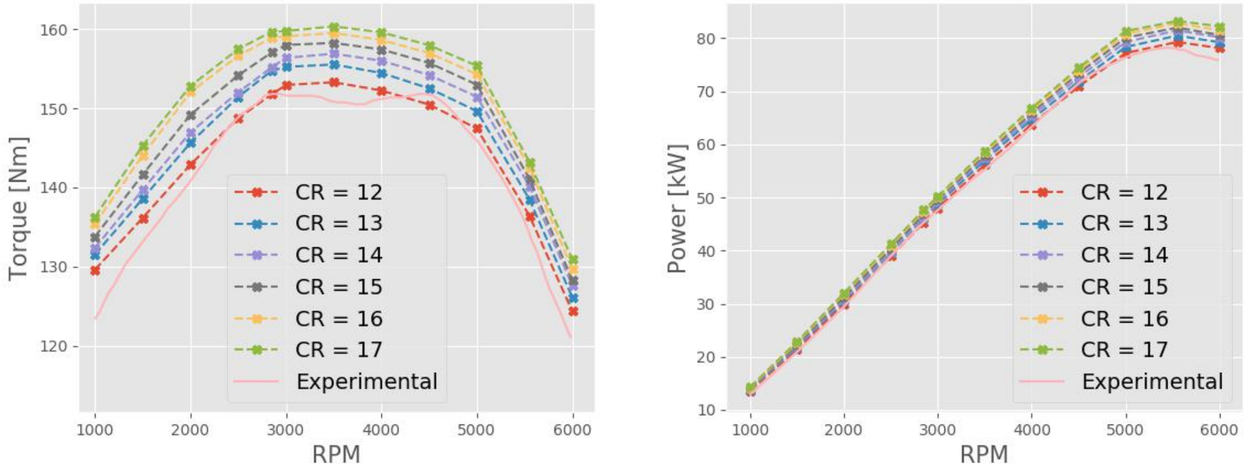

Figure 6. Torque and Power versus RPM for different compression ratios.

Figure 7 shows the results of the specific fuel consumption (left graph), and the net fuel consumption (right graph). It is possible to verify that, although there are changes in the specific fuel consumption, the net fuel consumption remains the same for all the compression ratios analyzed. This result is expected, since the displacement of the engine was not changed.

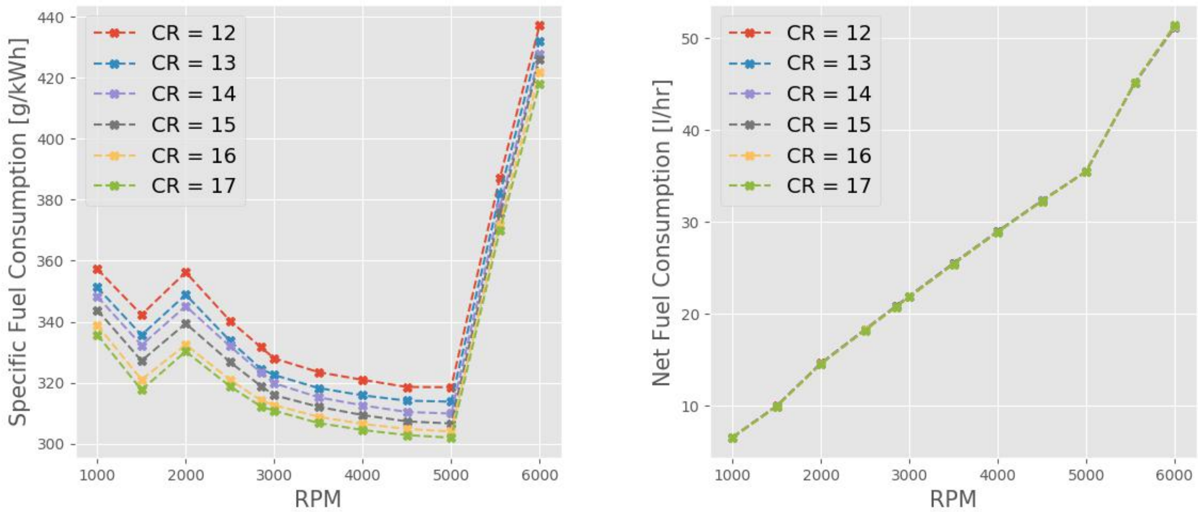

Figure 7. Specific Fuel Consumption and Net Fuel Consumption versus RPM for different compression ratios.

4.4 Influence of the Displacement

The analysis developed in the previous section consider only the engine compression ratio. As was possible to see at that section, the modification of the compression ratio has impact on the performance parameters of the engine, showing the possibility to extract more power of the thermal machine. Nevertheless, these performance parameters does not represent a problem to the customers, since the torque and power delivered by the current vehicles is enough to meet the traffic requirements. On the other hand, one huge issue that customers face is the fuel consumption. The results presented in Fig. 7 showed that the net fuel consumption remains basically the same changing the compression ratio, although there is reduction in specific fuel consumption. Knowing that the current performance parameters could be maintained, a strategy to reduce the net fuel consumption is change the swept volume of the engine.

The final analysis developed in this paper, was carried out changing not only the compression ratio of the engine, but also its displacement. These modifications were made in order to kept the performance parameters at the same levels of the original engine, and reduce the net fuel consumption.

The approach to the variation of the displacement was carried out keeping the ratio between the stroke and the bore of the cylinder constant, so that the geometry of the chamber was not altered.

Torque and power for this analysis were kept approximately constants in relation to the calibrated model, and these results are shown if Fig. 8. Figure 9 shows the results obtained for the specific fuel consumption over the speed range (left graph) and the net fuel consumption over the speed range (right graph) for various pressure ratio. Regarding the graph of the specific fuel consumption (*sfc*), these results are the same as those found in the analyzes developed in the previous section. It is noted that it does not present a smooth behavior, as might be expected. This can be explained due to the difference in the equivalence ratio at the extreme points of the speed range (Tonon, 2018).

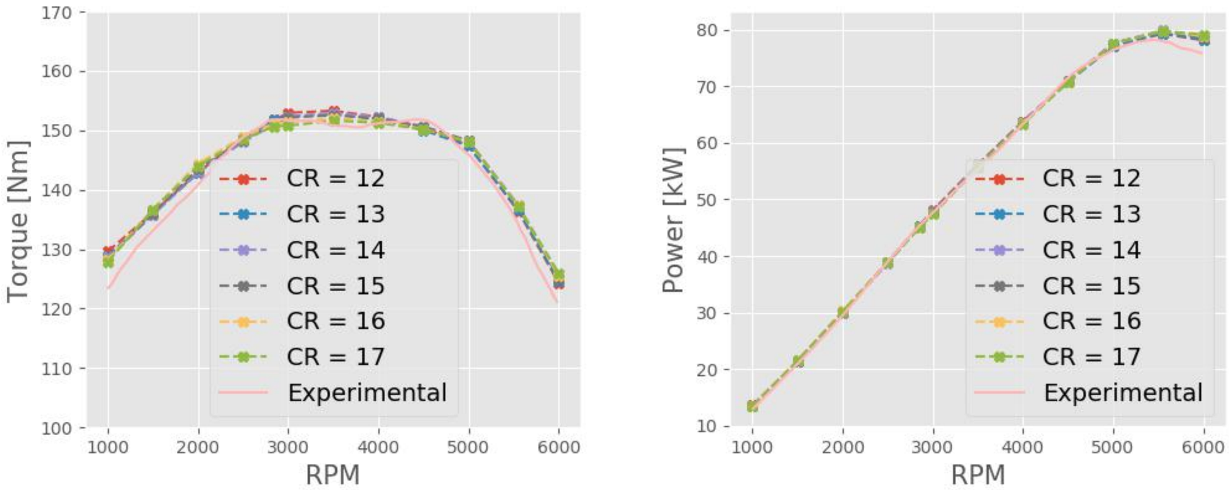

Figure 8. Torque and Power versus RPM varying CR and the displacement.

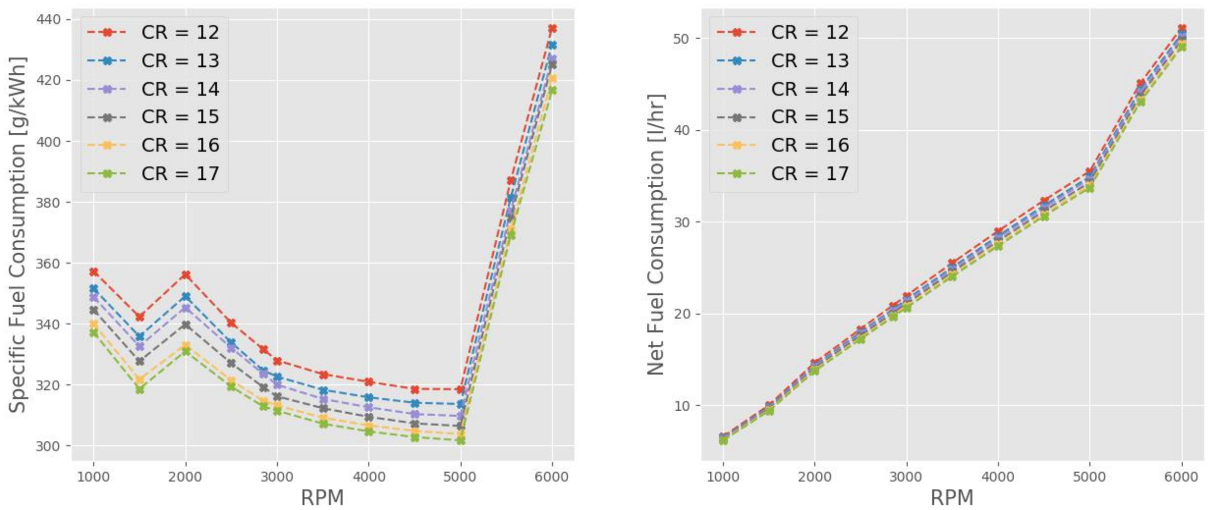

Figure 9. Specific fuel consumption and Net Fuel Consumption versus RPM varying CR and the displacement.

It can be seen in Fig. 9 that both the *sfc* and the net fuel consumption tend to decrease with the increase in the compression ratio and decrease in the displacement. The average reduction, considering the entire speed range, for the *sfc* was 5.51% and for the net fuel consumption was 5.60%. Comparing these results with those of the previous item (Fig. 7), on the analyzes changing only the CR, just the *sfc* was reduced. The reduction in net fuel consumption was possible due to the change in the displacement for each compression ratio, so the engine intakes less air-fuel mixture. Table 2 shows the displacement used for each of the analyzed compression ratios.

Table 2. Displacement used for each compression ratio settings.

Compression Ratio (CR)	12	13	14	15	16	17
Displacement (cm^3)	1,598	1,568	1,557	1,539	1,522	1,510

The engine displacement reduction generates positive effects not only on the thermal machine performance, but to the entire vehicle. This is a trend called downsizing, that allow the thermal machine and the vehicle to be lighter. Reductions in resistance to movement are achieved with a lighter vehicle, providing improvements in its overall energy balance. These improvements could be evaluated in future works.

5. CONCLUSIONS

The present work aimed to evaluate the behavior of an internal combustion engine using ethanol as a fuel. The study was conducted to foresee the potential advantages that the use of ethanol provides. The main characteristic evaluated is related to the high degree of knocking resistance that this fuel provided when compared with flex fuel and gasoline.

In order to adjust the computational model to ethanol, a previous analysis of the behavior of the knocking model was performed. In this analysis, it was possible to verify that the chosen model presents a good trend, coherent with what it was expected based on the theoretical concepts. However, few quantitative conclusions can be drawn from this model without the experimental data.

The model was calibrated using available experimental data as ~~a~~ reference. It was done to represent the studied engine in a satisfactory way. The average error along the speed range was of only 1.53%. The determination of the ignition delay time limit was estimated using the engine performance curves. As the compression ratio of the engine is limited for the use of gasoline, this limit should probably be overestimated. A better analysis could be made if the data were available from a cycle where knocking occurs, but with low intensities, in order to establish a more realistic limit.

In the latest analyzes developed in this work, interesting results were found. Keeping the original displacement and varying the compression ratio, an average increase of 5.36% in torque and power was found. When also the displacement was varied, in order to maintain the performance parameters informed by the automaker, it was possible to obtain an average reduction of up to 5.6% in the net fuel consumption.

Despite these good results, other knocking models could be incorporated in order to try to reproduce these results or define other modification strategies. Models that calculate the minimum octane number required to prevent knocking would be more appropriate for this analysis, since this would avoid the inconvenient need for data from a cycle with low knocking intensity to define the existence or not of this phenomenon.

The results of this paper were compared with the analyzed engine fueled with E100. These same results could be compared with the data available to the engine fueled with E27, however the performance data available show that the experiments with E27 lead to a lower performance than with E100. Once the paper search for possible improvements on the thermal machine, the comparison with the best experimental performance (fueled with E100) seems to be more realistic, and only this comparison was presented.

6. ACKNOWLEDGEMENTS

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7. REFERENCES

Caton, J. A., 2016. *An introduction to thermodynamic cycle simulation for internal combustion engines*. John Wiley & Sons, West Sussex.

Ferguson, C. R. and Kirkpatrick, A. T., 2016. *Internal Combustion Engines Applied Thermosciences*. John Wiley & Sons, New York, 3rd edition.

Ferrari, G., 2008. *Motori a Combustione Interna*. Il Capitello, Turin, 4th edition.

Hamilton, L. J., Rostedt, M. G., Caton, P. A. and Cowart, J. S., 2008. "Pre-ignition characteristics of ethanol ans E85 in a spark ignition engine". *SAE International Journal of Fuel and Lubricants*, Vol. 1, pp. 145-154.

Herynkova, H., 2009. *Impact analysis of difusion of electric vehicles in Denmark*. M.Sc. thesis, Lund University, Lund, Suécia.

Heywood, J. B., 1988. *Internal Combustion Engine Fundamentals*. McGraq-Hill Book Company, New York.

ITA, 2020. "Instituto Tecnológico de Aeronáutica", Instituto Tecnológico de Aeronáutica. 14 May 2020 <ita.br>.

Kintner-Meyer, M., Schneider and K. and Pratt, R., 2007. "Impact assessmet of plug-in hybrids on electric utilities and power grids" Pacific Northwest National Laboratory. 12 Apr. 2020 <https://www.ferc.gov/about/com-mem/5-24-07-technical-analy-wellinghoff.pdf>.

Mattos, A. P., 2018. *Estudo do desempenho de um motor turbo-alimentado a etanol empregando EGR para redução de emissões de NOx e controle de detonação.* Ph.D. thesis, Universidade Estadual de Campinas, Campinas, Brasil.

Millo, F., 2013. *Parametri geometrico : operative caratteristici*. Internal Publication – Politecnico di Torino, Turin.

Minstério do Meio Ambiente, 2020. "Clima e Biocombustíveis" Ministério do Meio Ambiente. 14 May 2020 <https://www.mma.gov.br/perguntasfrequentes.html?catid=13>.

Montoia, V., 2019. "O FIAT 147, primeiro carro a álcool brasileiro, era fabricado há 40 anos" Auto Esporte. 13 Apr. 2020 <https://revistaautoesporte.globo.com/Noticias/noticia/2019/07/o-fiat-147-primeiro-carro-alcool-brasileiro-era-fabricado-ha-40-anos.html>.

Moses, E., Yarin, A. L. and Bar-Yoseph, P., 1995. "On knocking prediction in spark ignition engines". *Combust Flame*, Vol. 101, pp. 239-261.

Ribeiro, R., 2018. "Cinco incentivos da Europa para carros elétricos que o Brasil pode adotar" UOL. 12 Apr. 2020 <https://www.uol.com.br/carros/noticias/redacao/2018/12/16/cinco-incentivos-da-europa-para-carros-eletricos-que-o-brasil-pode-adotar.htm>.

Tonon, D. S., 2018. *Simulação Computacional de um Motor de Combustão Interna do Tipo Alternativo Utilizando Etanol*. M.Sc. Thesis, Instituto Tecnológico de Aeronáutica, São José dos Campos, Brasil.

Tougri, I., Colaço, M. J., Leiroz, A. J. K. and Melo, T. C. C., 2016. "Knocking prediction in internal combustion engines via thermodynamic modeling: preliminary results and comparison with experimental data". *Journal of the Brazilian Society of Mechanical Sciences and Engineering*, Vol. 39, No. 1, pp. 321-327.

Wilkins, J., 1997. "Will electric vehicles really reduce pollution?", Ohio State University. 12 Apr. 2020 <https://www.asc.ohio-state.edu/wilkins.5/writing/Samples/policy/voytishlong.html>.

Yates, A. D. B. and Viljoen, C. L., 2008. "An improved empirical model for describing auto-ignition". *SAE Technical Paper*, 2008-01-1629.

Yates, A. D. B., Bell, A. and Swarts, A., 2010, "Insights relating to the auto-ignition characteristics of alcohol fuels". *Fuel*, Vol. 89, pp. 83-93.

Zarante, P. H. B. and Sodré, J. R., 2018. "Comparison of aldehyde emissions simulation with FTIR measurements in the exhaust of a spark ignition engine fueled by ethanol". *Heat ans Mass Transfer*, Vol. 54, pp. 2079-2087.

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