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# ANALYSIS OF THE THERMAL EFFICIENCY OF A DIESEL ENGINE OPERATING IN DIESEL AND ETHANOL BIOFUEL MODE

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***Abstract.*** *This study evaluates the thermal efficiency of a single-cycle diesel engine coupled to a three-phase electric power generator, with constant load. The behavior of the engine operating in dual-fuel mode, S-10 diesel oil and ethanol was analyzed using the ethanol fumigation technique by the air intake manifold. The tests consisted of changing the proportions of fuel mass supplied to the engine, by controlling the opening time of the ethanol injection nozzle and the accelerator that controls the diesel fuel injection pump, keeping the engine rotation constant. The data to carry out the analyzes were collected from load cells installed in the fuel tanks and from a pressure transducer installed inside the combustion chamber, from which it was possible to generate the actual cycle of the engine through the pressure versus diagram. volume. The analysis of the thermal efficiency of the engine for each test performed, showed a small increase for low mass proportions of ethanol, but a reduction of thermal efficiency according to the increase in the proportion of ethanol, due to the delay in the start of combustion and the reduction intake temperature, both factors influenced by ethanol fumigation. Another factor that was also observed was the increase in specific fuel consumption, due to the difference in calorific value between diesel oil and ethanol.*

***Keywords:*** *Thermal efficiency, ethanol fumigation, Diesel cycle.*

## 1. INTRODUCTION

Internal combustion engines are machines capable of transforming the energy made available by burning fuels into work. However, during this transformation process many losses occur, which can be, according to Martins (2013), mechanical losses, losses by heat transfer and losses due to the enthalpy of the exhaust gases. The objective of this research is to evaluate the thermal efficiency of a diesel cycle internal combustion engine operating in biofuel mode, S-10 diesel oil and ethanol, using the fumigation technique.

Initially, the data regarding the mass consumption of both fuels will be analyzed, in order to obtain the specific fuel consumption. This parameter, according to Obert (1971), is widely used to show how efficient an engine is turning fuel into work. The specific consumption is obtained through the ratio between the mass of fuel consumed by the engine and the power generated on its axis.

Then, the thermodynamic cycle of the engine will be analyzed. The cycle of an engine is capable of representing the physical and chemical transformations that occur with the active fluid while passing through the engine. Although the theoretical cycle does not correspond to the actual cycle of an engine, the first is a good reference for the thermodynamic study of engines. Through the theoretical cycle it is possible to analyze the operating conditions and compare different engines. Theoretical cycles are ideal cycles, where the active fluid is composed of air and behaves like a perfect gas (GIACOSA, 1980).

The actual cycle of an engine is traced through a pressure versus volume diagram, where the values are obtained experimentally by means of equipment capable of monitoring the pressure and the volume inside the cylinder (BRUNETTI, 2012). Figure 1 represents the actual cycle of the engine under study, generated during the control test operating with diesel oil.

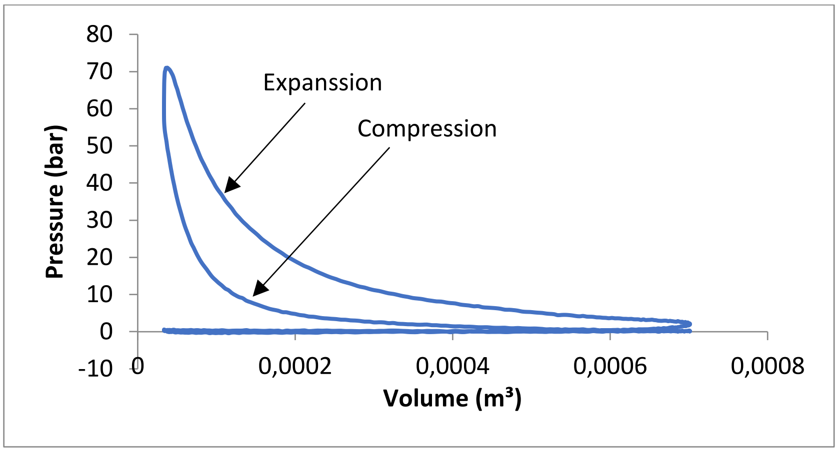


Figure 1. Actual cycle operating with diesel oil
Source: generated by the author

Figure 1 shows the compression and expansion times represented in the real pxV diagram, the intake and exhaust times are represented by overlapping lines due to the wide range of the sensor and the low pressure value measured in these steps. Once the actual cycle is determined by the pressure versus volume diagram, it is possible to calculate the actual work generated at each time in the cycle, through the cycle area. The indicated cycle power is determined by working the cycle per unit of time and can be obtained by Eq. (1) for 4-stroke engines (BRUNETTI, 2012).

$$P_i = W_c * \frac{n}{2} \tag{1}$$

Where:
$P_i$ = indicated power;
$W_c$ = cycle work;
$n$ = axis rotation.

The thermal efficiency of the engine will be obtained from the data of mass fuel consumption and fuel calorific value, these data will be related to the work generated by the cycle expansion time. In order to determine the most suitable fuel replacement condition in terms of efficiency.

Among the factors that comprise the thermal performance of the engine is the combustion efficiency and the adiabatic efficiency. The combustion efficiency deals with combustible compounds, such as CO, H2, HC and particles expelled by the exhaust due to incomplete combustion. These losses are accounted for by the combustion efficiency and it is the ratio between the energy released by the combustion and the energy contained in the fuel. The adiabatic efficiency, on the other hand, analyzes the part of the heat released by combustion that is lost by the heat transfer to the walls of the cylinder and the combustion chamber (MARTINS, 2013).

## 2. METHODOLOGY

The tests were performed on an internal combustion engine with a diesel cycle, with constant load and rotation. Initially, the control test was performed only with diesel oil, then the tests were repeated in bi-fuel mode, diesel oil and ethanol in different proportions of fuel substitution, through the ethanol fumigation by the engine air intake.

### 2.1 Characteristics of the experiment

A four-stroke single cycle diesel combustion engine, Agrale, model M93 ID with air cooling system with turbine incorporated into the steering wheel was used, the compression ratio was maintained at 21: 1. Coupled to the flywheel by a system of pulleys and belts, a three-phase Kohlbach electric generator, model 132LA, was installed and a set of electrical resistors connected to the generator, responsible for applying a constant load to the system of 7.22 kW .

To perform the ethanol injection, the fumigation technique was adopted. This technique consists of injecting fuel together with the air intake, providing the engine with an air-fuel mixture at the cycle intake time. The injector nozzle used was the IWP064, installed perpendicular to the air flow.

The control of ethanol injection is performed by the programmable electronic injection of the Fueltech brand, model FT300, which has the ability to adjust the opening time of the nozzle according to the programmed value, thus enabling the increase or reduction of the quantity of ethanol. injected into the engine. Its operation requires the signal of a hall-type rotation sensor installed close to the flywheel, which is activated by a magnet placed on the flywheel. Each time the magnet activates the hall sensor, it sends a signal to the electronic injection, indicating the moment when the nozzle should be opened, releasing the ethanol injection. As the handwheel rotates twice each cycle, there are two nozzle openings per cycle.

To monitor fuel consumption, a device was developed containing two reservoirs, one for diesel oil and the other for ethanol. These reservoirs are suspended in the structure of the device by two load cells model SPL, with a capacity of 5 kg and uncertainty of ± 0.5 g on the value read. The signals generated by the load cell go through an HX711 A / D converter and are sent to an Arduino microcontroller, model UNO R3. With this system it is possible to instantly monitor the fuel mass in each reservoir.

Fuel pumps were installed inside the reservoirs, one for diesel oil and another for ethanol, the latter being able to keep the ethanol supply line pressurized at 5 bar. The load cell calibration procedure was similar to the scale calibration procedure, where standard weight blocks are used.

## 2.2 Data acquisition to plot the pressure curve

The pressure monitoring in the combustion chamber during the tests was performed by a pressure transducer model EWCT-7-312M from the company KULITE SEMICONDUCTOR PRODUCTS INC. which has an uncertainty of 0.1% over the full scale of 300 bar. To install the pressure transducer, it was necessary to remove the engine head so that it was possible to drill the hole that serves as the housing for the transducer and connects the instrument to the combustion chamber. Figure 2 shows the drilling hole for the pressure transducer inside the engine head, next to the orifice of the diesel injector nozzle.

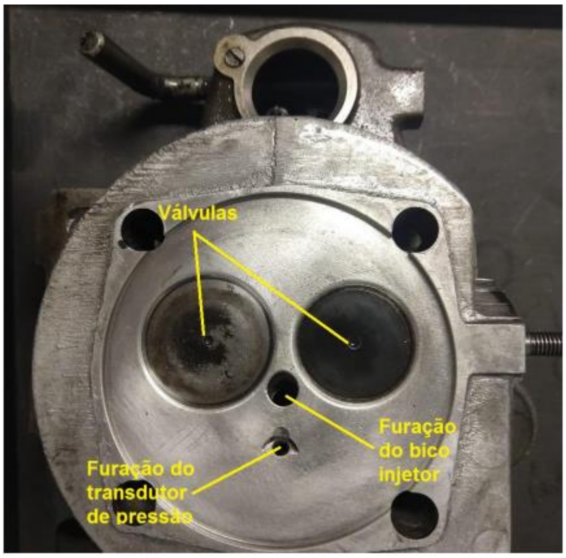


Figure 2. Position of the pressure transducer.
Source: registered by the author

To guarantee the reliability of the results obtained during the tests, the pressure transducer was calibrated on a manometer calibration bench, where the pressure on the calibration bench was gradually increased from 0 to 120 bar, in 4 bar intervals and the voltage signal emitted by the transducer was registered, so it was possible to compare the voltage values emitted by the transducer in relation to the pressure indicated by the manometers. Figure 3 shows the curve with the results generated from the stress versus pressure values. As can be seen, the trend line generated a mathematical model capable of explaining the behavior of the pressure transducer.

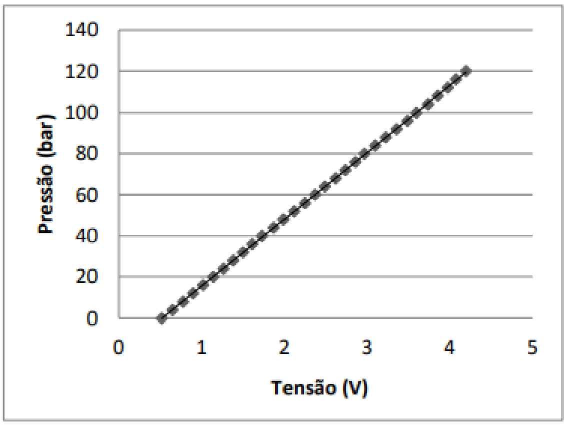


Figure 3. Pressure x voltage curve of the transducer.
Source: prepared by the author.

In conjunction with the pressure transducer, an angular position sensor was installed concentric to the flywheel, this equipment is capable of reading the crankshaft angle. With the pressure records inside the cylinder and the angle of the crankshaft it is possible to generate the pressure curve versus angle of the crankshaft that will be used later to analyze the behavior of the engine. The position sensor used was an incremental encoder type, this sensor converts the rotating movements of the crankshaft into square-wave electrical pulses, generating an exact number of electrical pulses perfectly distributed along the 360 degrees of the rotation axis. The angular position sensor has been calibrated so that the zero angle of the encoder coincides with the piston PMS.

With the data collected by the pressure transducer and the angular position sensor it is possible to generate the pressure versus crankshaft angle curves (pxα), the work produced by the cycle must be evaluated from the area above the pxα curves and can be obtained through the Eq. (2).

$$W_{liq} = \int_{0°}^{720°} p dV \tag{2}$$

Being:
Wliq: liquid work;
p: pressure;
V: volume.

The way to obtain the value of the internal volume of the cylinder is through Eq. (3) which relates the displacement of the piston according to the angle of the crankshaft.

$$V = V_{cc}\left[1 + 0{,}5(r_v - 1)\left(\frac{L}{R} + 1 - \cos\alpha - \sqrt{\left(\frac{L}{R}\right)^2 - \sin^2\alpha}\right)\right] \tag{3}$$

Being:
V: volume;
L: connecting rod length;
R: crankshaft radius;
Vcc: combustion chamber volume;
rv: compression rate;
α: crankshaft angle.

## 2.3 Procedures and tests

The tests were carried out at the Unisinos engine laboratory and consist of changing the proportions of fuel supplied to the engine, keeping the engine speed and load constant. Initially the engine will run for 15 minutes without load, then the load is applied and it is waited 30 minutes for the engine to reach the working temperature and for it to be possible to perform the control test using only S-10 diesel oil 7% biodiesel. After this stage, the tests with ethanol fumigation begin. Figure 4 shows the tests carried out with their respective proportions according to the mass of fuel consumed.

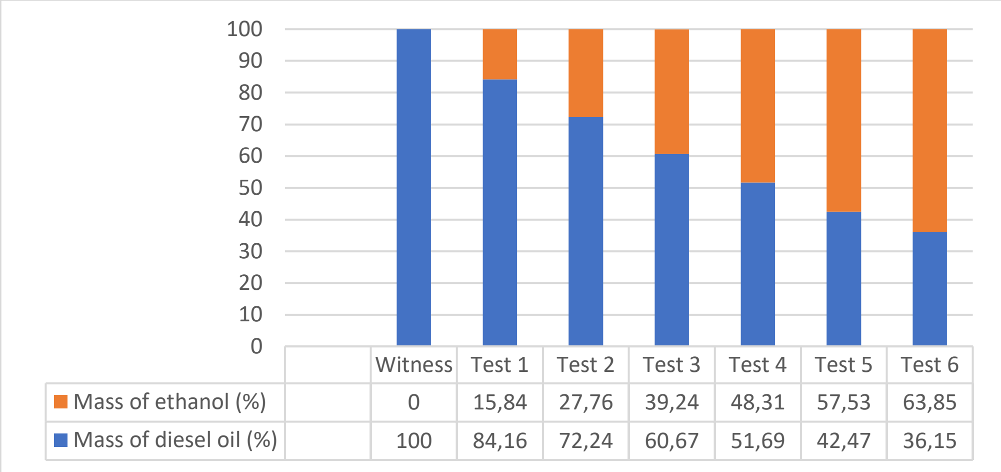


Figure 4. Tests performed.
Source: prepared by the author.

Each test lasted 3 minutes and according to the proportion of ethanol it is necessary to reduce the proportion of diesel oil, through the accelerator lever, keeping the engine speed constant in all tests at 2560 RPM. After carrying out the tests, the engine was left operating only with diesel oil for 15 minutes with load and another 15 minutes without load applied for cooling.

## 3. RESULTS

In this step, the results obtained with the experiment will be presented.

### 3.1 Fuel consumption

With the information collected by the measurements of the load cells installed in the fuel tanks, it was possible to generate the data presented in Table 1 that represent the respective fuel mass consumption in each test performed, the information presented corresponds to the data collected in the interval of 120 seconds for all tests. This limitation was performed to generate the standardization of the collected data.

Table 1. Fuel consumption.

| Teste | Diesel oil consumption (g) | Ethanol consumption (g) | Total consumption (g) | Specific consumption (g/kWh) |
|---|---|---|---|---|
| Witness | 92,25 | 0,00 | 92,25 | 199,44 |
| Test 1 | 79,14 | 14,90 | 94,04 | 205,56 |
| Test 2 | 69,47 | 26,69 | 96,16 | 207,36 |
| Test 3 | 61,02 | 39,41 | 100,43 | 225,00 |
| Test 4 | 54,96 | 51,37 | 106,33 | 246,60 |
| Test 5 | 46,36 | 62,79 | 109,15 | 257,76 |
| Test 6 | 42,48 | 75,04 | 117,52 | 267,84 |

As can be seen in Table 1, as the replacement of diesel oil with ethanol occurs, specific fuel consumption increases. This fact is explained by the difference between the calorific value of ethanol and diesel oil, 27710 kJ/kg and 43000 kJ/kg respectively, that is, a greater amount of ethanol is needed to produce the same amount of energy. According to the tests carried out by Guedes (2017), the specific fuel consumption for fixed injection rates increases, as the proportion of ethanol rises in the mixture.

Knowing that the data presented regarding fuel consumption represent a consumption interval of 120 seconds and that the engine speed has remained constant at 2560 RPM, we can calculate the fuel consumption per cycle, dividing the fuel consumption by the number of cycles in 120 seconds. We can also obtain the energy available by burning the fuel in each cycle if we multiply the calorific value of the fuel by the fuel consumption per cycle. Figure 5 shows the values, in Joule, of energy made available by the burning of fuels in each cycle.

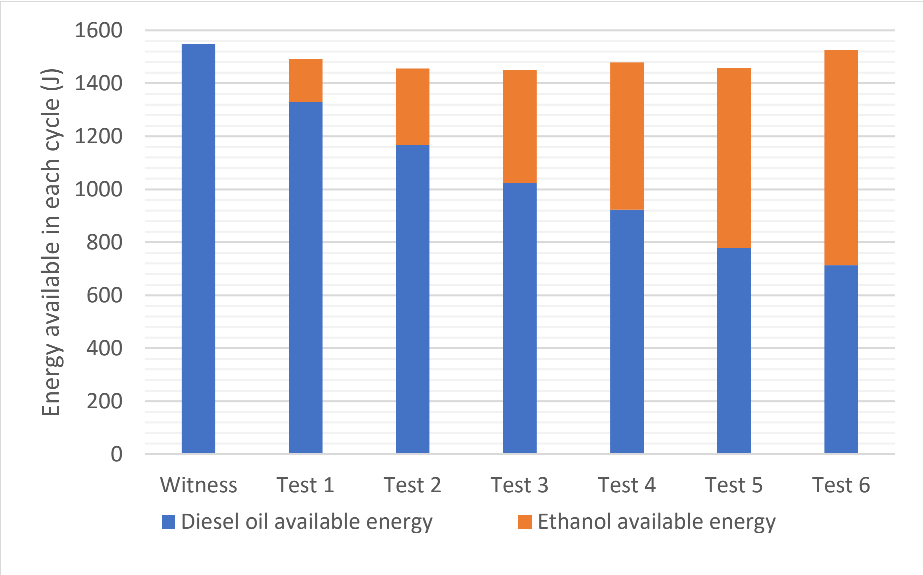


Figure 5. Available energy per cycle.
Source: prepared by the author.

In Fig. 5 it is possible to verify that as the substitution of diesel oil for ethanol occurs in the tests, the energy made available by each fuel also changes, however the total energy that represents the sum of both fuels in each test shows a variation too small. This fact can be explained due to the need to keep the engine speed and load applied constant, regardless of fuel substitution.

### 3.2 Work produced by the cycle

In the thermodynamic cycle of a four-stroke engine, only the expansion time generates positive work, the net work produced by the cycle will be the difference between the work generated in the expansion time and the work consumed in the rest of the times. Table 2 shows the work at each stage of the cycle during the tests performed.

Table 2. Mechanical work of the cycle.

| Test | Wadm (J) | Wcom (J) | Wexp (J) | Wexh (J) | Wliq (J) |
|---|---|---|---|---|---|
| Witness | -4,0 | -339,9 | 1015,6 | -21,2 | 650,6 |
| Test 1 | -4,7 | -340,7 | 1017,3 | -28,8 | 643,0 |
| Test 2 | -4,9 | -325,7 | 1004,9 | -22,5 | 651,8 |
| Test 3 | -5,1 | -324,1 | 974,4 | -17,5 | 627,7 |
| Test 4 | -4,5 | -327,0 | 964,5 | -26,5 | 606,5 |
| Test 5 | -4,7 | -327,3 | 949,5 | -22,2 | 595,3 |
| Test 6 | -4,2 | -314,9 | 961,7 | -25,2 | 617,3 |

The values shown in Tab. 2 are directly related to the values read by the pressure transducer. In the measurements made in the intake and exhaust times, the pressure value is much less than the maximum value of the sensor, therefore, the values can have great uncertainties.

### 3.3 Engine efficiency

To assess the thermal efficiency of the engine, a balance will be made between the energy made available by burning the fuel in each cycle and the net work produced by the cycle, whereas the isentropic expansion efficiency will be analyzed from an energy balance between the energy of the engine. fuel and energy produced in the cycle expansion work. Figure 6 shows the comparison between these data.

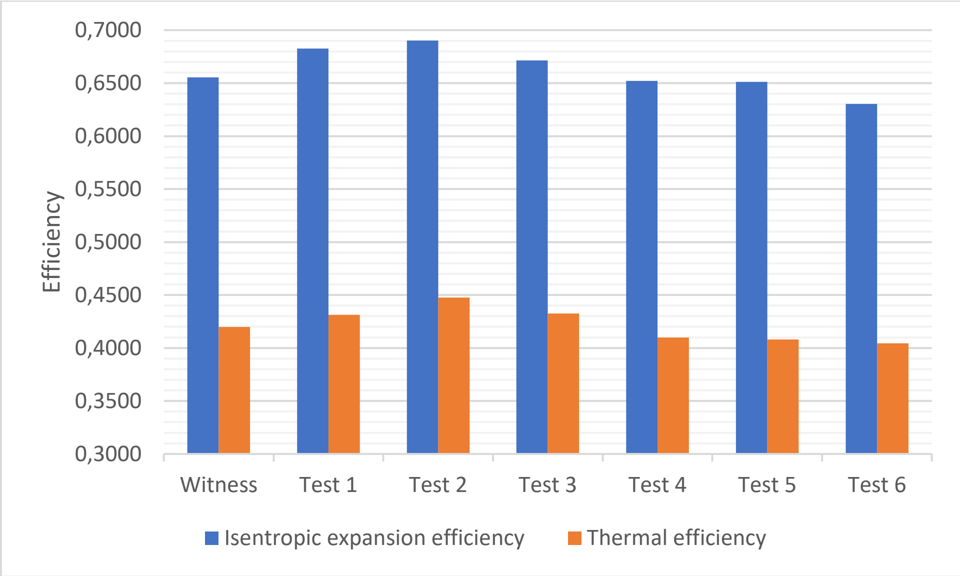


Figure 6. Efficiencies.
Source: prepared by the author.

According to Fig. 6, it is possible to observe that the engine under study showed the highest efficiency in Test 2, where the proportion of fuel mass is 72.24% diesel oil and 27.76% ethanol, however if we observe the work expansion shown in Tab. 2, it can be noted that both the Witness test and Test 1 have higher expansion work values than that of Test 2. This fact can be explained because the start of combustion exactly coincides with the PMS in the Test 2, which makes better use of the piston descent path in addition to avoiding back pressure on the piston if combustion starts before the PMS.

Another fact that can also be seen in Fig. 6 is the reduction of the thermal and isentropic efficiency of the engine from Test 3, a consequence that may be related to the delay in the start of combustion generated by the increase in the proportion of ethanol. Since the results obtained by Guedes (2017), also demonstrated the tendency of ethanol to delay the start of combustion, being necessary to adjust the fuel injection advance.

The start of combustion can be seen in Figure 7, that through the pressure variation rate compare the pressure increase rates as a function of the crankshaft angle.

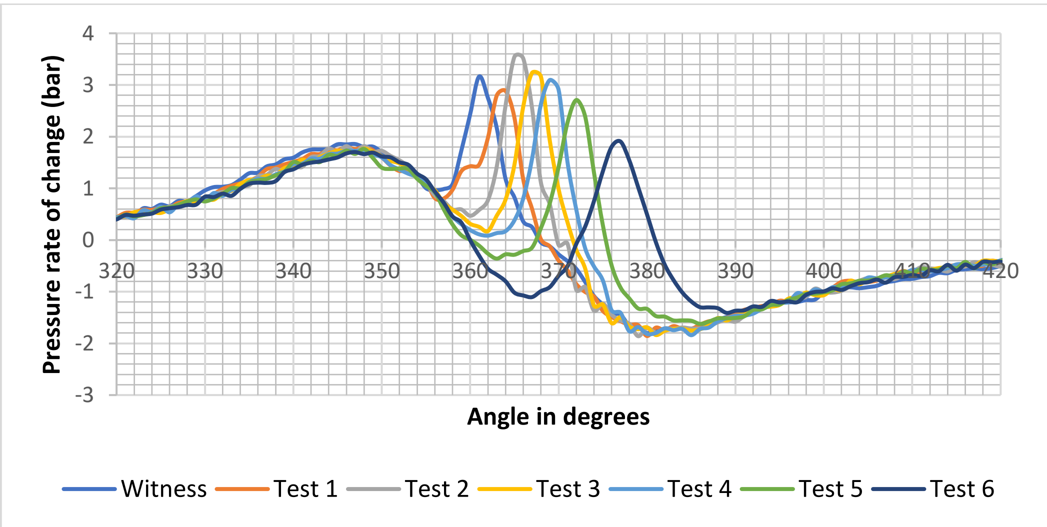


Figure 7. Pressure variation rate.
Source: prepared by the author.

When analyzing the curves represented in Fig. 7, it is evident that the increase in the proportion of ethanol in the mixture caused the delay in the start of combustion. Therefore, the combustion efficiency may be affected by the delay in the start of combustion and by the cooling of the combustible air mixture allowed by the collector, due to the latent heat of vaporization of ethanol.

## 4. CONCLUSIONS

After the execution of the experimental tests performed in the laboratory and the analysis of the information collected during the experiments, it was possible to conclude that the main modification in the cycle with the ethanol fumigation was the delay in the start of combustion and the cooling of the fuel mixture. With small amounts of ethanol, efficiency improved because the delay in the start of combustion allowed a better use of the energy released in combustion.

The graph presented by Fig. 7 clearly shows the combustion delay caused by the ethanol fumigation, it is believed that the drop in the intake temperature and the reduction in pressure at the end of the compression, influenced the combustion delay. Another important factor that may have contributed to the combustion delay is the fact that the auto-ignition temperature of ethanol is higher than the auto-ignition temperature of diesel oil, thus requiring higher values of pressure and temperature to ignite.

When analyzing the net work produced by the cycle and the thermal and isentropic efficiency of each test, it is noted that the largest net work produced and the highest efficiency coincide in test 2, where ethanol fumigation occurs and combustion starts at PMS, having in view that in the test it shows where the engine operates only with diesel oil the combustion starts with 2 degrees of advance. However, it is also possible to observe that the increase in net work generated by the cycle does not mean an increase in terms of efficiency, as can be seen in test 6, in which there was an increase in net work, but the thermal and isentropic efficiency continued to decrease. This drop in efficiency is due to the greater increase in fuel consumption compared to the increase in net work produced.

However, it is believed that to obtain better results in terms of engine performance and efficiency, it is necessary to advance the ignition point, as the proportion of ethanol increases in the mixture, causing combustion to start at the PMS.

## 5. REFERENCES

BRUNETTI, Franco. Motores de combustão interna. São Paulo, SP: Blucher, c2012. 2 v. ISBN 9788521207085 (v. 1).

BRUNETTI, Franco. Motores de combustão interna. São Paulo, SP: Blucher, c2013. 2 v. ISBN 9788521207092 (v. 2).

GIACOSA, Dante. Motores endotérmicos. 3. ed. Madrid: Editorial Dossat S.A, 1980. ISBN 84-237-0382-7.

GUEDES, Andrew David Mendes. Estudo Experimental sobre o Impacto do Etanol em Misturas Diesel-Biodiesel-Etanol nos Motores de Ignição por Compressão. 2017. 149 f. Dissertação (Mestrado) - Curso de Engenharia Mecânica, Departamento de Engenharia Mecânica, Pontifícia Universidade Católica do Rio de Janeiro, Rio de Janeiro, 2017.

HEYWOOD, John B. Internal Combustion Engine Fundamentals. New York: McGraw-Hill, 1980. ISBN 978-0-07-028637-5.

MARTINS, Jorge. Motores de Combustão Interna. 4. ed. Porto: Publindústria, 2013. ISBN 978-989-723-033-2.

OBERT, Edward F. Motores de combustão interna. 2. ed. Porto Alegre: Editora Globo S.A, 1971.

PULKRABEK, Willard W. Engineering fundamentals of the Internal Combustion Engine. 2. ed. Upper Saddle River: Pearson Prentice-Hall, 2004. ISBN 0-13-140570- 5.

ROSA, Josimar Souza. Estudo de um motor diesel monocilíndrico bicombustível. 2014. 102 f. Dissertação (Mestrado) - Curso de Engenharia Mecânica, Unisinos, São Leopoldo, 2014.

TAYLOR, Charles Fayette. Análise dos motores de combustão interna. 2. ed. São Paulo: Edgard Blucher Ltda, 1995. 2 v. Tradutor: Mauro Ormeu Cardoso Amorelli.

TURNS, Stephen R. Introdução a combustão: conceitos e aplicações. 3. ed. Porto Alegre: AMGH, 2013. INBN 978-85-8055-274-4.

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