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EVALUATING ENGINE COMBUSTION PARAMETERS FROM CYLINDER PRESSURE MEASUREMENT

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Abstract. *In this study the parameters from cylinder pressure measurement were evaluated from a four cylinder, naturally aspirated diesel power generator engine fuelled by blends of 93% diesel oil + 7 % biodiesel (B5). Cylinder pressure data was recorded using a data acquisition system at every 0.1° and was done at constant speed (1800rpm). The measures includes the acquisition of instantaneous cylinder pressure, determination of the top dead center and the measure of instantaneous crank angle from a magnetic speed sensor mounted opposite to a phonic wheel installed in the engine crankshaft. The pressure data was measured at different loads and used to calculate the heat release rate using the first law of thermodynamic, determine the maximum pressure, the ignition delay, the indicated mean effective pressure and the power generator mechanical efficiency.*

Keywords: *in-cylinder pressure, diesel engine, combustion*

1. INTRODUCTION

The combustion process in diesel engines is complex because it depends on details as the fuel characteristics, the combustion chamber geometry, the fuel injection system and operation conditions (Heywood, 1988). During the conversion of chemical energy into mechanical work process, for producing reciprocating motion of the piston during compression and expansion of in-cylinder gas, the pressure in the combustion chamber affects directly the power and engine performance (Zhao and Ladommatos, 2001). Thus, the study of in-cylinder pressure in synchronous to the piston position or to the crank angle is important. The objective of this work is to obtain performance parameters of a diesel engine from the gas pressure curve inside of one of the engine cylinders.

The record of the pressure curve is an important source of information for the analysis of chemical, physical and mechanical processes occurring inside the combustion chamber of internal combustion engines. It provides information such as, the engine indicated mechanical power, peak pressure, heat release rate, ignition time, combustion duration, compression parameters, combustion problems, exhaust gases recirculation, emissions evaluation of particulate matter (PM) and NOx, among others (Antonopoulos and Hountalas, 2012; Chung et al., 2013). For this, the in-cylinder pressure diagram should be recorded and processed by pressure sensors and transducers with high strength, accuracy, reliability and acquisition frequency. Pressure sensors are extremely useful in research that changes the engine structure or operation condition.

Maurya, Pal and Agarwal (2013) performed the diagnosis of combustion in a direct injection, naturally aspirated, water-cooled, four-cylinder diesel engine powered with gasoline. One cylinder was modified to operate in homogeneous charge compression ignition (HCCI) mode, while the other cylinders operated in the standard mode. During the experimental investigation, the engine operated in different conditions of speed, load and air-fuel ratio. The in-cylinder pressure signals were captured using a piezoelectric pressure transducer (Kistler model 6013C) mounted on the cylinder head adapted for HCCI operation. The pressure signals data were filtered and used to calculate, cycle to cycle, the average of the pressure variations, the pressure standard deviations, the pressure growth rate and the heat release rate.

Bodisco and Brown (2013) investigated the inter-cycle variability of in-cylinder pressure in an ethanol fumigated common rail diesel engine. Specifically, the authors investigated the effects of ethanol concentration on the maximum pressure rise rate, peak pressure magnitude and moment, the ignition delay and the inter-cycle variability. A piezoelectric pressure transducer (Kistler model 6053CC60) was mounted, to measure the in-cylinder pressure, with a simultaneous four channels data acquisition module, to eliminate the phase shift between channels and to perform measurements in the same time reference. The authors varied the ethanol concentrations and torque conditions. The study allowed to observe significant changes in ignition delay with the use of high ethanol concentrations.

Tutak (2014) evaluated the effects of E85 (85% ethanol and 15% gasoline) atomization in the intake manifold of a compression ignition direct injection diesel engine. The in-cylinder pressure analysis showed reduction in the pressure during the intake stroke, with the use of E85, due to the cooling effect caused by the fuel. The increase in E85 injection caused an increase in ignition delay and in the peak of heat release rate. The author also showed that with 20% of E85, the heat release curve trend was similar to the obtained using pure diesel oil.

2. THEORETICAL ANALYSIS

The indicate work per cycle is defined as the work done by the gas on the piston (Heywood, 1988). The engine p-V diagram can be obtained from the in-cylinder pressure and volume data during the cycles. The indicated work is calculated by integrating the curve, as shown by Eq. (1):

$$W_i = \oint p.dV \tag{1}$$

Where W_i is the indicated work per cycle (kJ); p is the in-cylinder pressure (kPa); and V is the cylinder volume (m³).

The indicated power per cycle is resultant from the work done by the gas on the piston, and corresponds to the sum of the effective power, the friction power and the power used to the engine intake and exhaust strokes. It can be calculated by:

$$P_i = \frac{W_i.\omega}{n_r} \tag{2}$$

Where P_i is the indicated power per cycle (kW) and n_r is the number of revolutions of the crankshaft for a cylinder working cycle. For a four stroke engine, n_r = 2.

The indicated mean effective pressure is given by the ratio between the indicated work per cycle and the volume displaced per cycle, given by:

$$IMEP = \frac{W_i}{V_d} \tag{3}$$

Where IMEP is the indicated mean effective pressure (kPa) and V_d is the volume displaced by the piston per cycle (m³).

From the indicated power per cycle, it is possible to determine the engine mechanical efficiency (Heywood, 1988):

$$\eta_m = 100\%.\frac{P_b}{P_i} \tag{4}$$

Where η_m is the engine mechanical efficiency (%) and P_b is the engine output power(kW).

Heywood (1988) defines the heat release rate as the rate at which the chemical energy of the fuel is released by the combustion process. This parameter can be determined from the engine in-cylinder pressure in function of the crank angle, calculating the energy required to obtain the pressure measured experimentally. The diesel engine combustion is a complex process and it complicates the establishment of a universal mathematical model to calculate the heat release rate. Simplified model are used to compare the engine performance at different operation conditions. Heywood (1988) presents a zero dimensional model applying the first law of thermodynamics for an open system. The equation that determines the apparent net heat release rate is given by:

$$\frac{dQ_l}{d\theta} = \left(\frac{\gamma}{\gamma - 1}\right).p.\frac{dV}{d\theta} + \frac{1}{\gamma - 1}.V.\frac{dP}{d\theta} \tag{5}$$

Where $dQ_l/d\theta$ is the apparent net heat release rate (kJ/ºCA); γ is the ratio of specific heats (*cp/cv*); *p.dV/dθ* is the work rate developed by the gas on the piston (kJ/ºCA).

The ignition delay, in diesel engines, is the interval between the start of fuel injection and the start of combustion, usually measured by the crank angle. It was considered the start of combustion as the maximum value of the second derivative of the pressure curve in relation to the crank angle, based on the energy release rate in the form of heat is directly proportional to the rate of pressure change in the combustion (Zhu et al., 2001). In the engine used in this work, the fuel injection angle is 23ºCA BTDC (before top dead center). The difference between this angle and the angle determined as start of combustion was considered as the ignition delay. The duration of combustion was determined from the interval between the start of combustion and the angle that occurs 95% of the total cumulative heat released (Zhu et al., 2011; Zhang et al., 2013).

3. EXPERIMENTAL METHODOLOGY

The experiments were carried out in a production, naturally-aspirated, four-stroke, four-cylinder diesel engine. Table 1 presents the engine main characteristics.

Table 1. Diesel Power Generator Specification

Equipment	PARAMETER	VALUE
Engine	Number of cylinders	4
	Bore × Stroke	0.102 m × 0.120 m
	Total displacement	3.922×10^{-3} m^3
	Compression ratio	17.0:1
	Rated power @ 1800 rev/min	44 kW
Electric generator	Number of poles	4
	Voltage	220 V
	Number of phases	3
	Nominal power	55 kW
	Frequency	60 Hz

The pressure inside the first engine cylinder was measured by a quartz piezoelectric sensor, model 6061B, manufactured by Kistler Instrument Corporation, with measuring range between 0 and 250 bar and sensitivity of -25.6 pC/bar. This sensor has a dynamic nature, providing an output signal only if stimulated, i.e., when there is a pressure variation. This feature affects how the sensor is calibrated, as its output signal is processed. The measurement apparatus must include a water cooling system. The cooling system confers stability to the sensor, enabling its use at high temperatures ambient, such as inside the engine cylinder. For the conversion of the sensor electric charge variation to voltage linear variation, it was used a charge amplifier, model 5037B. Figure 1 shows the Kistler pressure sensor and the charge amplifier.

Figure 1. Kistler pressure sensor model 6061B and charge amplifier model 5037B

As mentioned, these sensors measure the dynamic pressure of the cylinder, i.e., the pressure variation inside the cylinder. To determine the various parameters from these data is necessary to determine the absolute pressure of the cylinder; therefore, the curve should be adjusted to represent this pressure (Rogers, 2010). There are several techniques to perform this adjustment, varying in complexity, processing time and accuracy. In this work it was used the technique of fixing a point as reference value. This technique consists of moving the curve obtained by adding a known pressure value at a certain angle of the crankshaft, and is suitable for engines without throttle valve in the intake manifold and naturally aspirated. Generally, it is used as reference the intake manifold air pressure, which can be obtained by an absolute pressure transducer, which was made in this work.

The data acquisition rate was set at 100,000 Hz, equivalent to an angular resolution of 0,108º CA. Data from the pressure transducer are stored in worksheets for each load test. A trigger wheel with a magnetic sensor was used to synchronize the pressure data with the engine cycle and was acquired simultaneously with the pressure sensor. The pressure curves were positioned in function of the crank angle and was calculated the pressure average using 30 cycles. Figure 2 shows the pressure sensor position in the engine head.

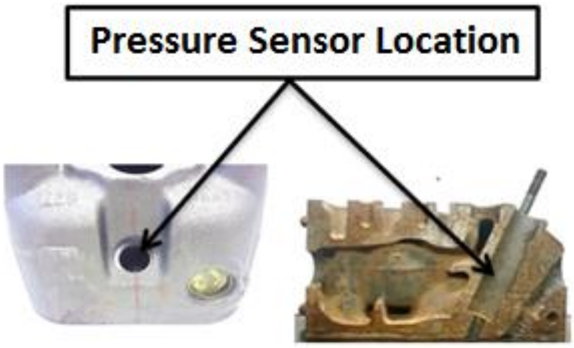

Figure 2. Position of the pressure sensor in the engine head

The control of the tests was carried out from a supervisory developed in LabVIEW platform. Electrical and electronic circuits make the interface between the sensors and the electrical load bank and a computer. In addition to the in-cylinder pressure data, other several parameters were monitored during the tests as: temperature at different points, the mass flow of air and fuel, atmospheric conditions, exhaust gases composition and electrical characteristics of the generated energy. Two data acquisition modules provided by National Instruments were used, USB-NI-DAQ-6229 and USB-NI-DAQ-6211. Figure 3 shows the experimental apparatus diagram.

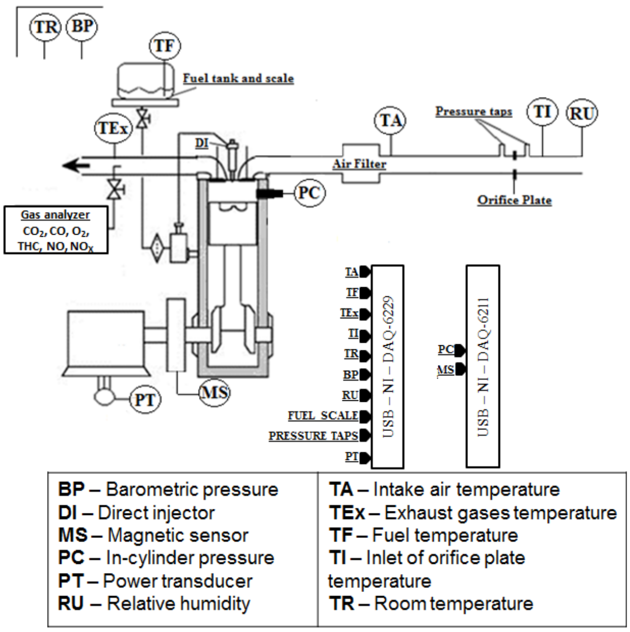

Figure 3. Diagram of the experimental apparatus

The experimental procedure follow the definitions and procedures from the standard ABNT NBR 6396: 1976. The tests were carried out with variation of the applied load power of 0 kW, 10 kW, 20 kW, 30 kW, and 37.5 kW. The fuel consists of a mixture of 93% diesel oil and 7% biodiesel (B7).

4. RESULTS

Pressure curves were positioned as a function of crank angle and the means were calculated from the pressure transducer and inductive sensor data, enabling the calculation of the following presented parameters. Figure 4 shows the in-cylinder pressure as a function of crank angle and the applied load. In ignition compression engine, the peak pressure is dependent on the amount of fuel burned at the beginning of combustion, i.e., the premixed combustion phase (Sinha and Agarwal, 2007). The in-cylinder pressure increases as the applied load increases, since greater quantity of fuel is injected and greater amount of heat is released during the combustion. The peak pressure moves away from TDC with load increasing, behavior also reported by Zhu et al. (2011). The peak pressure was 2645 kPa for 0 kW, 2966 kPa for 10 kW, 3327 kPa for 20 kW, 3556 kPa for 30 kW and 3667 kPa for 37.5 kW.

Figure 5 shows the net heat release rate during combustion (Eq. (5)). The results showed tendency to increase the peak heat release rate for low to medium loads and to maintain or reduce for high loads, as also shown by Zhu et al. (2011). The maximum heat release rate values were 23.8 J/ºCA (0 kW), 39.1 J/ºCA (10 kW), 50.7 J/ºCA (20 kW), 51.0 J/ºCA (30 kW) and 49.1 J/ºCA (37,5kW).

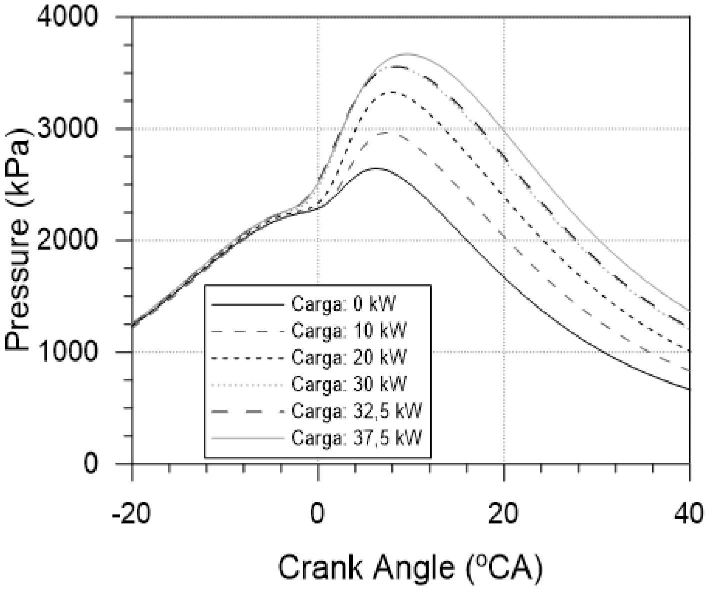

Figure 4. In-cylinder pressure

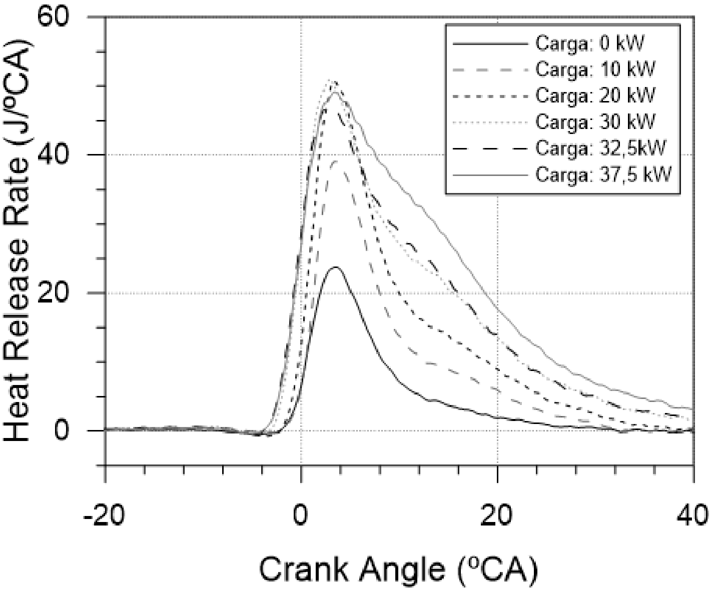

Figure 5. Net apparent heat release rate inside the cylinder

It was observed that, through the pressure curves, with increasing load, there is a tendency to reduction in ignition delay. The ignition delay was determined from the second derivative of the pressure curves. Figure 6 shows the second derivative obtained for the load of 37.5 KW. The maximum peak was regarded as the start of combustion.

Figure 9 shows the ignition delay variation in function of the engine load. It was found that there is an increase in ignition delay until the load of 20 kW, and from this load a reduction of the ignition delay. The ignition delay values were

23.4ºCA (0 kW), 24.1ºCA (10 kW), 24.3ºCA (20 kW), 22.9ºCA (30 kW) and 21.9ºCA (37.5 kW). The ignition delay trend can explain the trend of increase the peak heat release rate between the low and medium loads, and reduction or maintaining for high loads, since larger ignition delays cause a greater amount fuel accumulated before the start of combustion, inducing more fuel to be burned during the premixed combustion phase (Pidol et al., 2012). Figure 10 shows the duration of combustion in function of applied load. There is an increase in combustion duration with the load increase, explained due to the longer duration of fuel injection and formation of the air/fuel mixture to supply the load demand (Zhang et al., 2013).

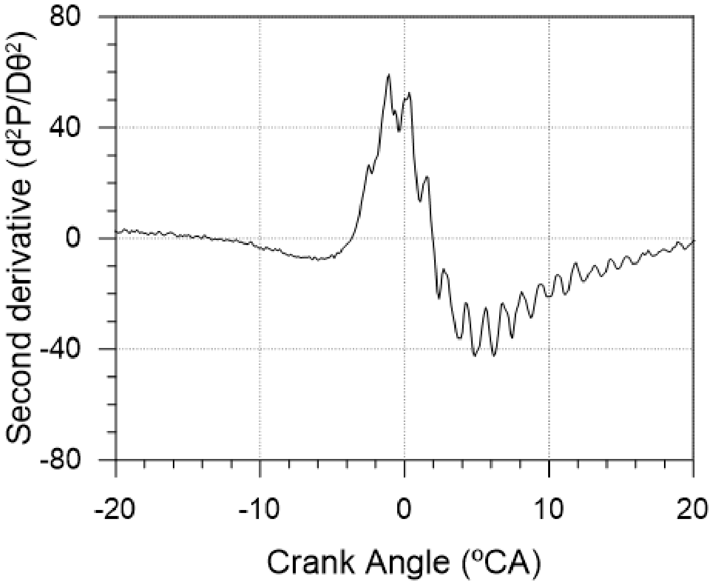

Figure 6. Second derivative of pressure in function of crank angle for the load of 37.5 KW

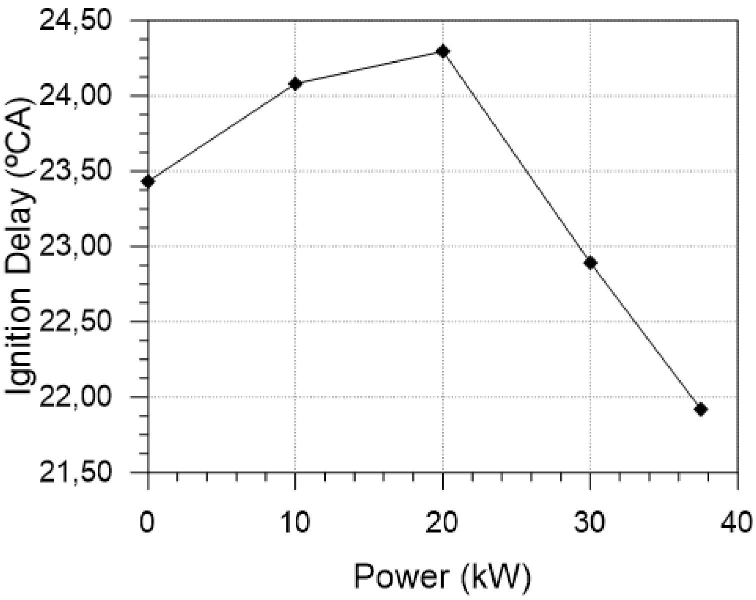

Figure 7. Ignition delay in function of engine load

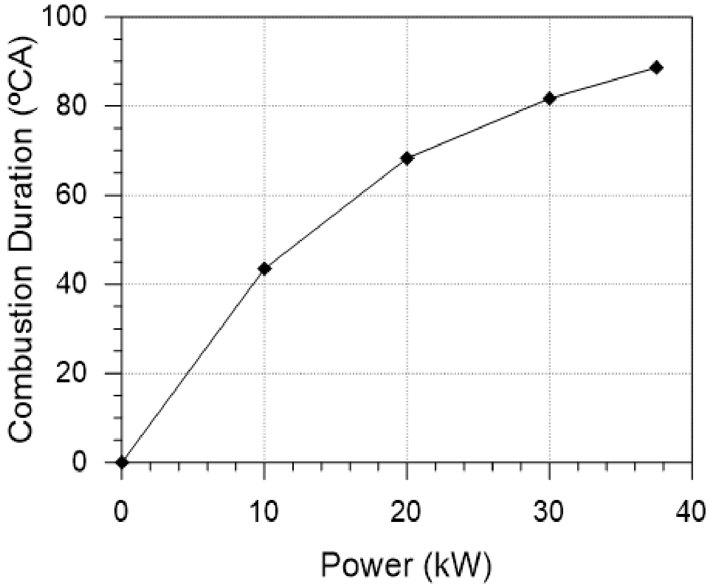

Figure 8. Duration of combustion in function of the engine load

Figure 11 shows the p-V diagram for the engine loads. The diagram have typical format of four-stroke internal combustion engines and, from Eq. (1), the indicated work per cycle was determined, as well as the indicated power (Eq. (2)) and IMEP (Eq. (3)). With increase in engine load, pressure levels increased; consequently there was an increase in the area of the diagrams, i.e., increase in the work done during the cycle. The calculated IMEP values indicate an increase of this parameter with increase in engine applied load.

The indicated power was determined to obtain the power generator mechanical efficiency for each applied load (Eq. (4)). Figure 12 shows the results, indicating an increase in mechanical efficiency with increase in the engine load. The no-load engine operation (0 kW), resulted in null mechanical efficiency and with the operation in maximum load, 37.5 kW, resulted in a mechanical efficiency of 88%. Part of the power produced by the engine is dissipated due to friction of the moving parts of the engine, as pistons and bearings (Heywood, 1988). This power is considered constant in great engine speed range. Thereby, the engine mechanical efficiency is greater as the power demand increases, within the engine design power range.

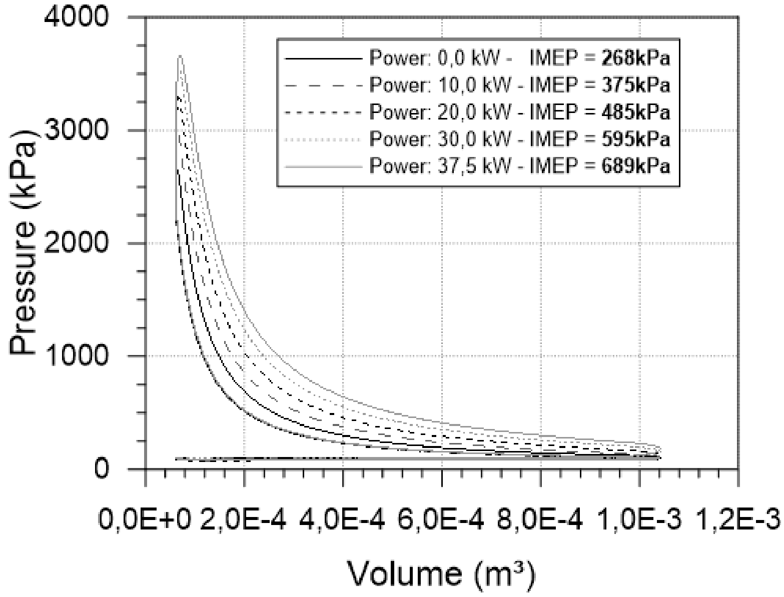

Figure 9. p-V diagram and IMEP

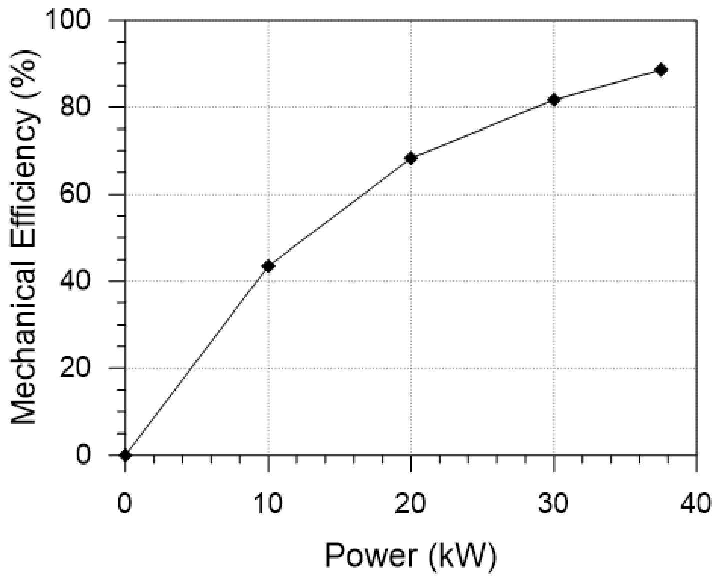

Figure 10. Mechanical efficiency of the power generator engine

5. CONCLUSIONS

This study investigated parameters of combustion and engine performance, from the experimental in-cylinder pressure curve. The main results showed:

- Peak pressure increase inside the cylinder with increase of the engine load;
- The high loads presented lower ignition delay in relation to low and medium loads, attributed to higher pressures and temperatures reached with increased fuel demand;
- The higher fuel consumption with the increase in engine load caused increase in combustion duration and the amount of heat released;
- The p-V diagrams allowed the determination of the IMEP and the power generator mechanical efficiency, showing increase in efficiency with the increase in engine load.

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