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IMPACTS ON THE PARAMETERS OF WIEBE FUNCTION WITH LOAD VARIATION IN COMPRESSION IGNITION ENGINES

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Abstract. *The present work deals with the impacts influenced by the variation of the load in the unknown parameters of the Wiebe function applied in a Cummins diesel generator set, 4 cylinders, displaced 3.9 liters and consuming commercial fuel oil. Experimental tests were performed to measure the parameters of effective pressure indicated as a function of the crankshaft, fuel consumption, air consumption, electrical power and temperatures. In order to evaluate the combustion process, we used mathematical methods to calculate the rate of release of heat, released heat, burned fuel mass fraction and determination of the unknown parameters of the Wiebe function that describes the combustion process in reciprocating engines. combustion. The results obtained show changes in the start times and duration of the combustion and in the shape parameter of the fraction of experimental fuel mass fraction with the change of the applied load regime in the engine. The different amounts of air-fuel masses consumed by the engine to compensate for the increase in load anticipated the start of the flame resulting in a shorter duration of the combustion process with the engine running at full load.*

Keywords: *Combustion, diesel comercial, Wiebe function, generator set, load variation.*

1. INTRODUCTION

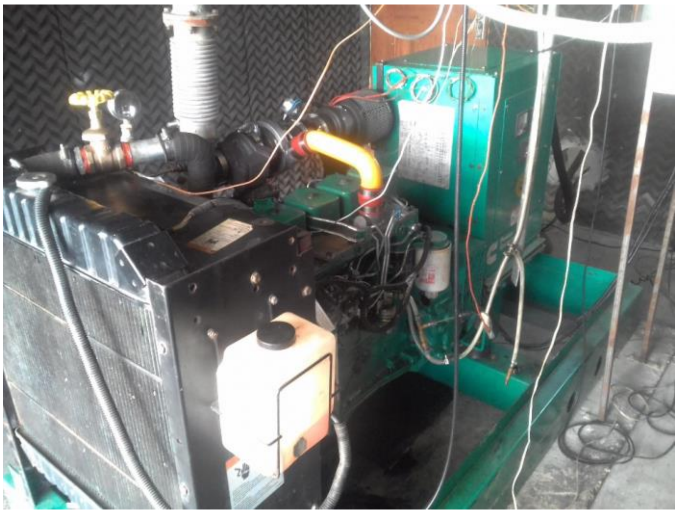
Generator sets are designed to work in a constant rotation range, respecting the standard frequency of electric power of 60 Hz, however, motor rotation tends to vary depending on the fluctuations of the electric consumption demand for different hours throughout the day. To maintain rotation within the generation frequency range, the engine should increase the fuel consumption rate as the required load increases, or vice versa, to lower demands (Da Conceição, 2015). This control is carried out by the amount of fuel injected into the combustion chamber as a function of the applied load, involving different air-fuel masses directly impacting the combustion process, such as: ignition delay, start of combustion, duration of combustion, heat and effective pressure indicated inside the cylinder (Fergunson, 2011 and Da Conceição, 2015).

The indicated effective pressure curve is an effective parameter for the study and analysis of internal combustion engines, provides important information about the process that burns fuel inside the cylinder (Oliveira, 2016 and Rajasekar, 2014), allowing to characterize the behavior of the combustion process under different operating conditions and / or modifications of adjustable parameters so that the engine consumes fuel more efficiently and within the standards required by regulatory bodies.

In this context, the work proposes an analysis of the combustion process in a Cummins generator set, of compression ignition consuming commercial fuel oil and the changes assigned in the parameters of the Wiebe function for different load regimes applied in the generator in operation.

2. EXPERIMENTAL PROCEDURE

The generator set of 42 kW of power, 4 cylinders in line, 3.9 liters, operates with commercial diesel oil in a rotation range of approximately 1800 rpm, Fig. 1 presents a general picture of the object of study.



Figue 1. Cummins generator set overview.

The resistive loads were applied in plots of 40, 50, 70, 90 and 100% of the total power capacity. The fuel consumption was determined by a positive flow meter supplied by the Oval, model LSF41 with ± 1% accuracy of the reading. The electrical magnitudes were measured using a SAGA 4500 energy analyzer manufactured by Landis + Gyr, recording values of power, voltage, current and frequency of each generator phase with accuracy of 1% of the reading.

The temperatures were recorded using K-type thermocouples at localized points to determine the permanent operating regime of the motor, recognized by the stabilization of the temperature registers to start the data collection, as well as used to calculate the mass consumption of fuel. A MAF (Mass Airflow Sensors) sensor was installed on the turbocharger inlet line, this equipment measured the mass quantity of air admitted per unit time used to calculate the volumetric efficiency of the engine.

In order to measure the pressure inside the cylinder as a function of the crank angle, an AVL-supplied equipment was used consisting of a piezoelectric pressure sensor, model GU21D, installed in the motor head by an adapted glove with direct access to the combustion chamber through a bore, an Encoder 365C model rotation sensor positioned on the motor shaft to map the angular position of the crank crank system, all integrated into a single AVL Indimicro 602 data acquisition and processing system.

In synchrony with the rotation sensor, the piezoelectric sensor, under the action of a load, generates electricity proportional to the compressive force produced by the piston in the mass of air during the time of compression and expansion of the gases (Brunetti, 2012 and Oliveira, 2016). The rotation sensor encodes the position of the angle of the crank rod system, sending signals to the Indimicro 602 processor, responsible for monitoring the combustion process in real time (Oliveira, 2016). The equipment also has two softwares to calculate from the pressure curve and imputed motor construction data, parameters of heat release rate, released heat, maxima and minima, inflection points of the pressure curves and the rate of change of the pressure and position of the crankshaft (Oliveira, 2016). The pressure data collected for analysis of the combustion process in this work is the result of the average of 500 power cycles obtained with the engine running.

3. MATHEMATICAL FORMULATION

The formulations of the mathematical model used in this work are based on the principles of the first Law of Thermodynamics adopting the same simplifications by Heywood (1988) and works developed with engines in the study of combustion. The temperatures and pressures inside the cylinder are considered uniform during the combustion process, the phenomena of the vaporization process of fuel to form the air-fuel mixture after the injection are neglected, as well as the volumes between the plunger, rings and wall of the cylinder and the leakage between ring seals and cylinder walls (Heywood, 1988).

The calculation of the heat release rate, formulation Eq. (1), relates the rates of change of the indicated effective pressure and the combustion chamber volume in relation to the rate of change of the position of the crank rod system described by Heywood (1988).

$$\frac{Q(\theta)}{d\theta} = \left(\frac{\gamma}{\gamma - 1}\right) \times p(\theta)\frac{dV(\theta)}{d\theta} + \left(\frac{1}{\gamma - 1}\right) \times V(\theta)\frac{dp(\theta)}{d\theta} \tag{1}$$

$p(\theta)$ and $V(\theta)$ correspond to the pressure and volume of the cylinder in the θ position of the crankshaft, γ is the ratio of specific heats c_p e c_v given by Eq. (2).

$$\gamma = \frac{c_p}{c_v} \tag{2}$$

Considering that the air-fuel mixture behaves like an ideal gas, the specific heats c_p and c_v are listed as follows (Brunetti, 2012).

$$R = c_p - c_v \tag{3}$$

R assumes the value of the constant for an ideal gas of 287,12 [J/kg K].

The rate of change in volume ($dV(\theta)/d\theta$) is obtained by deriving Eq. (4). This equation represents the volume of the cylinder as a function of the crankshaft.

$$V(\theta) = V_c + \frac{V_c}{2} \times (r_c - 1) \times \left[R_m + 1 - \cos\theta - \left(R_m^2 - \sin^2\theta\right)^{1/2}\right] \tag{4}$$

V_c is the dead volume of the combustion chamber when the piston is in the upper dead center position.
The compression ratio r_c is given by Eq. (5) and the variable R_m represents the ratio of the length of the connecting rod to the crank radius.

$$r_c = \frac{V_d}{V_c} \tag{5}$$

The curve of the released heat is obtained by the integral of the rate of release of heat Eq. (6) (Heywood, 1988).

$$Q(\theta) = \int_{\theta_{start}}^{\theta_{end}} \frac{Q(\theta)}{d\theta} d\theta \tag{6}$$

In the combustion process, the mass of fuel is gradually consumed by the flame resulting in the release of heat until all the fuel is consumed (Turns, 2013), establishes a direct relation between the heat released and the total heat delivered by the fuel Q_{fuel} (Heywoob, 1988). From this analysis it is possible to calculate the mass fraction of fuel burned MFB which represents the percentage value of the amount of fuel consumed by the flame during the combustion process formulated by the Eq. (7) (Rocha, 2016).

$$MFB = \frac{Q(\theta)}{Q_{fuel}} = \frac{Q(\theta)}{m_{fuel} \times PCI_{fuel}} \tag{7}$$

Where m_{fuel} is the mass of fuel consumed per power cycle and the PCI_{fuel} represents the lower calorific value of the fuel.

3.1 Wiebe function

The empirical equation of Wiebe Eq. (8) which describes the mass fraction of burned fuel is usually used to characterize the combustion process in internal combustion engines, it is a relation of the time of the beginning of the combustion θ_0, duration of combustion $\Delta\theta$, the shape parameter "m" which characterizes the rate at which the fuel is consumed by the flame and the combustion efficiency factor "a". The variable θ is the angular position at every instant of the system crank crank in movement (Rocha, 2016; Bueno, 2011; Fergunson, 2011; Kumar *et al*., 2016; Sun, 2017; Cooney, C. *et al*., 2008).

$$x(\theta) = 1 - \exp\left[-a \times \left(\frac{\theta - \theta_0}{\Delta\theta}\right)^{m+1}\right] \tag{8}$$

The mass fraction of burned fuel is a measure of the ratio of the cumulative amount of heat released and the total energy delivered by the mass of fuel converging to a unit value ($x(\theta) = 1$) at the end of the combustion process (Fergunson, 2016). In the literature the end of the combustion is usually defined by an arbitrary value that represents a quantity in percentage of the fuel that burned (Fergunson, 2016). The values of 90%, 99% and 99.9% of the burned fuel mass fraction are usually considered, corresponding to the respective values 2,302, 4,605 and 6,908 assigned as an efficiency factor "a" of Wiebe (Rocha, 2016; Fergunson, 2011; Kumar *et al*., 2016; Cooney, C. *et al*., 2008).

4. METHODOLOGY

The objective of this work was to analyze the combustion process of a stationary Diesel cycle engine operating at different load regimes and to predict the main influences on the parameters of the Wiebe function when the engine is subjected to load variation. The combustion process of Diesel cycle engines is quite complex due to the amount of physical and chemical phenomena that occurred during the formation and duration of the fuel burning (Fergunson, 2011 and Luna, 2014). Therefore, in this work, simplifications were adopted in the mathematical models to determine the parameters of the Wiebe function from the pressure curve, measured experimentally inside the cylinder.

The pressure inside the cylinder can be measured directly on the internal combustion engines by a pressure gauge used to calculate the combustion parameters (Oliveira, 2016 and Rajasekar, 2014). The released heat rate is determined from the First Law of Thermodynamics using cylinder pressure data and geometric parameters that vary depending on the crank rod system (Chung, 2013 and Rajasekar, 2014) described in section 3.

In this work the start of the combustion was determined by the analysis of the effective pressure curve indicated applying the method of the first derivative ($dp/d\theta$). With this method, it is possible to locate the combustion on the spot (Fergunson, 2011 and Oliveira, 2016). When the fuel is injected into the cylinder part of the heat of the mass of air that has been compressed is transferred to the liquid fuel that evaporates forming the flammable mixture initiating the combustion (Ferguson, 2011 and Rocha, 2016). All these effects are reproduced in the behavior of the pressure inside the cylinder (Oliveira, 2016 and Sousa, 2016). Thus, the onset of combustion is recognized as the maximum positive value of the first derivative after the start of fuel injection (Fergunson, 2011 and Oliveira, 2016).

Some constructive characteristics of the compression ignition combustion engines are intentionally constructed so that the combustion process occurs in turbulent mode such as piston head cavities and variable geometries of the intake port (Fergunson, 2011). Turbulence is also a characteristic of the flame inside the cylinder causing perturbations in the pressure curve (Rocha, 2016). In this work, no form of filter was used that could mask the pressure behavior inside the cylinder, using the raw values of the pressure reading to calculate the heat release rate determined by Eq. (1), considering the variable value $\gamma = 1{,}37$ constant throughout the gas compression and expansion process.

The end of combustion was considered when 90% of MFB is consumed. In the calculation, only the effective heat delivered to the crank system resulting from the pressure measurement produced by the combustion was quantified, referred to as the apparent liquid heat rate (Loganathan, S. *et al*., 2018), thus, the Eq. (7) was modified in the form of Eq. (9). The variable $Q_{\max}$ is equivalent to the maximum accumulated net value of the heat release by the combustion in the form of pressure, also obtained in the form of the integral of Eq. (6).

$$MBF = \frac{Q(\theta)}{Q_{\max}} \tag{9}$$

In this work, the efficiency factor of the Wiebe function was defined as the value equivalent to 90% of the fraction of fuel burned considered for the end of the combustion, leaving only the shape parameter of the Wiebe function as an unknown factor. For this, the Levenberg-Marquardt method of minimization was used to determine the best value that represents the shape parameter of the curve MFB. The Levenberg-Marquardt method is an optimization of the Gauss-Newton method for solution of nonlinear systems by least squares (Gardenghi, 2011), adjusts the model equation to the data obtained experimentally (Ranganathan, 2004).

5. RESULTS

The results obtained with the generator set operating at different load regimes are presented below in the form of graphs of the indicated effective pressure curve, heat release rate, burned fuel mass fraction and Wiebe function parameters for each operating regime the engine.

The pressure curves inside the cylinder are shown in Fig. 2, their shape moves continuously in the combustion regions when the load requirements are higher, this behavior is a consequence of the larger amount of fuel consumed by the motor needed to compensate for the increase of charge. The pressure inside the cylinder depends on the amount of fuel mixed with the air during the combustion process, the higher the amount of fuel injected, the higher the pressure values reached (Rajasekar, 2014). At first, only the fuel that was mixed with the air during the ignition delay is consumed instantaneously by the premixed flame, producing high rates of pressure increase in the cylinder reaching the peak values (Ferguson, 2011 and Rajasekar, 2014). In the first instance, only the fuel which has been mixed with the air during the ignition delay is consumed instantaneously by the premixed flame, producing high rates of pressure increase in the cylinder reaching peak values, the remainder of the fuel which did not burn during pre-mixed combustion, is consumed at a slower rate during diffuse and / or controlled combustion (Heywood, 1988; Fergunson, 2011 and Rajasekar, 2014). Referring to the graph of Fig. 2, the increase in pressure peak is a result of the greater amount of fuel consumed as the load value increases.

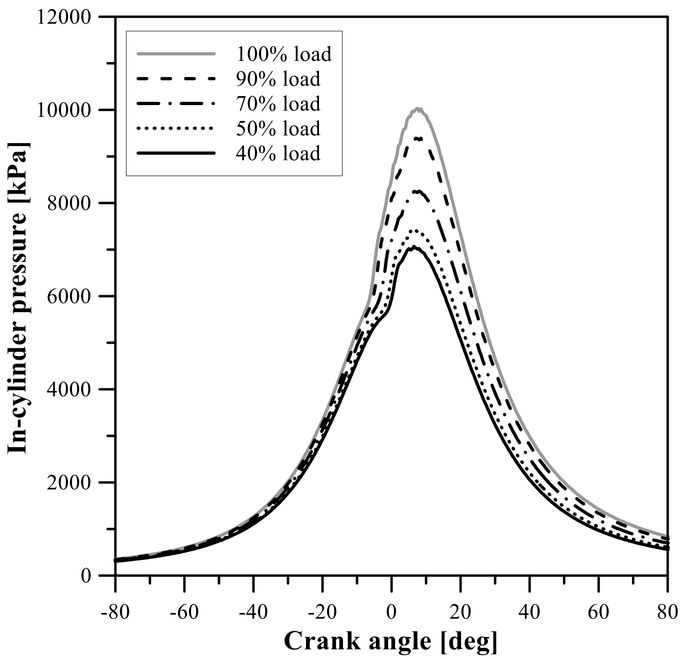


Figure 2. Pressure curves as a function of the angle of the crankshaft.

The amount of energy released during combustion is proportional to the amount of fuel that is consumed to maintain the load applied to the motor shaft. The higher pressure rates are attributed to increased engine load, especially when operating at 100% power. A larger amount of fuel injected into the cylinder faster the flammable air-fuel mixture will be formed in anticipation of the onset of combustion (Fergunson, 2011).

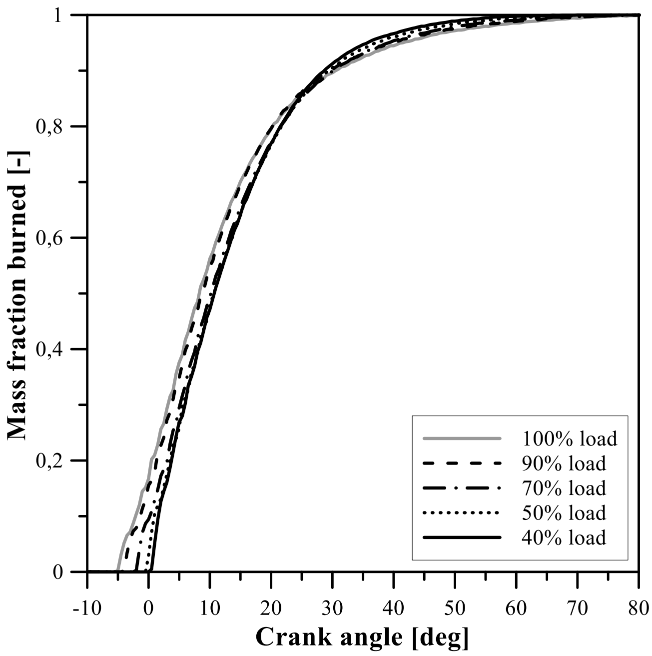


Figure 3. Fuel mass fraction burned for different loads.

The results of the burned fuel mass fraction are shown in Fig. 3. In the graph it is possible to observe the anticipation of the start of combustion, resulting from a lower ignition delay when the engine is subjected to loads that demand higher fuel consumption.

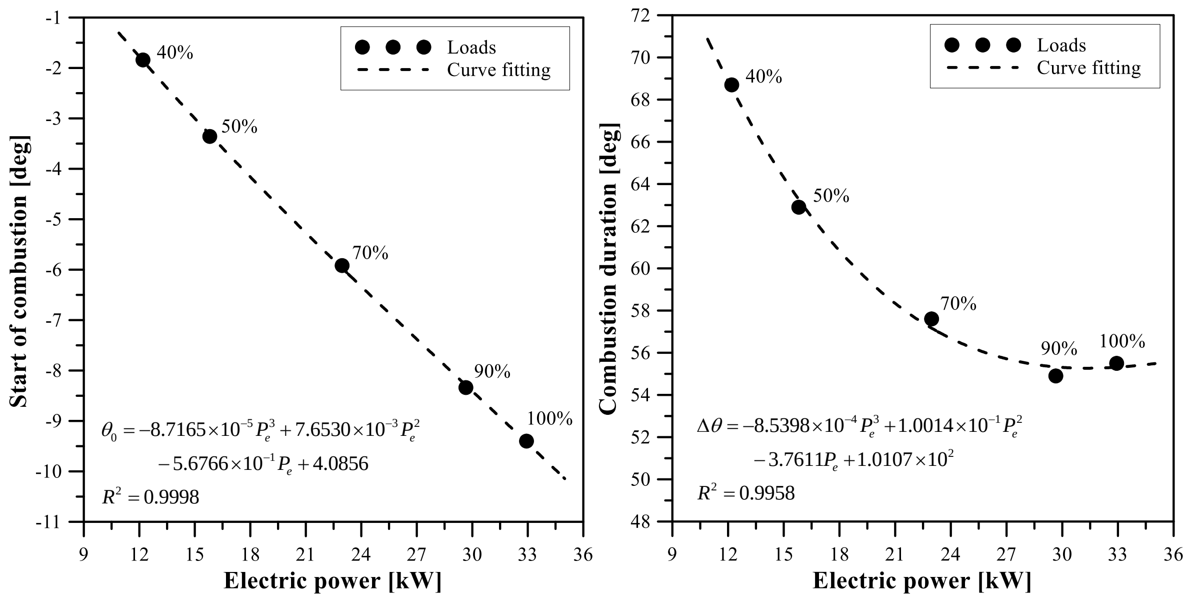


Figure 4. Curves combustion times as a function of the power applied to the generator.

In Fig. 4 the effects produced by the power variation in the combustion times of the Wiebe function are shown. The start of the combustion was gradually anticipated with the increase of the load, also resulting in a shorter duration of combustion. The higher amount of fuel mass injected into the cylinder contributed to form a flammable air-fuel mixture in a shorter time interval (Fergunson, 2011), thus the combustion process starts faster for higher loads where fuel consumption has been bigger. As a consequence of the higher amount of flammable mixture consumed during the premixed flame, the energy levels reached at this stage are higher by increasing the blending rate of the fuel that has not yet burned in contact with the high temperatures, subsequently consumed in the diffuse combustion phase , the higher the rate of burning, the better the burning rate, the lower the burn-up time, as the burning rate is controlled by the mixing rate (Heywood, 1988 and Fergunson, 2011).

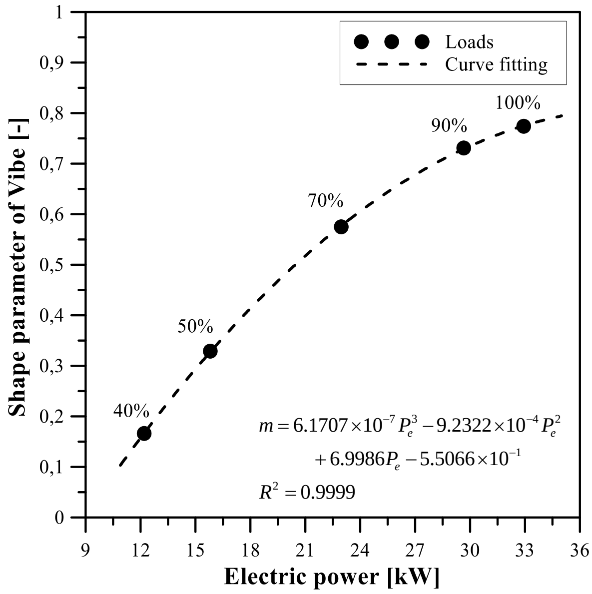


Figure 5. Behavior of the parameter "m" depending on the operating power of the generator.

The effects on the fuel burn rate produced by the load variation in the generator are shown in the graph of Fig. 5. The shape parameter of the Wiebe function describes the speed that the fuel is burned during the combustion process (Rocha, 2016 and Sun, 2017). In compression ignition engines the released heat rate is more intense at the start of combustion and slower at the end (Fergunson, 2011). The results of Fig. 6 show that the fuel was consumed faster by the flame as the load portion in the generator was increased, corresponding to the gradual reduction of the duration of the combustion.

6. CONCLUSION

In this work the main influences on the behavior of the parameters of the Wiebe function for a Diesel cycle engine operating at constant speed and under different load regimes were evaluated. The results showed changes in the start times and duration of the combustion and in the shape parameter of the burned fuel mass fraction curve for each load regime obtained in experimental tests.

The onset of combustion gradually anticipated the increase in generator load. The load variation also had an impact on the duration of the combustion. More eluted loads resulted in shorter combustion times, resulting from higher rates of heat release due to higher fuel burn rates.

7. ACKNOWLEDGEMENTS

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8. REFERENCES

Bueno, J. P. V. M. *Análise do desempenho de motores diesel utilizando óleo combustível pesado e combustível destilado marítimo*. Universidade Federal do Rio de Janeiro. Brazil, 2011.

Brunetti, F. *Motores de combustão interna*. São Paulo: Blucher, v1. 2012.

Chung, Jaesung *et al*. Real-time combustion parameter estimation algorithm for light-duty diesel engines using in-cylinder pressure measurement. *Applied Thermal Engineering*, v. 60, n. 1-2, p. 33-43, 2013.

Cooney, C. *et al*. Wiebe function parameter determination for mass fraction burn calculation in an ethanol-gasoline fuelled SI engine. *Journal of KONES*, v. 15, p. 567-574, 2008.

Da Conceição, A. O. *Recuperação e comissionamento de um grupo-gerador a diesel modelo YANMAR-4TNV88-GGE de 20 kVA*. Trabalho de conclusão de curso, Faculdade de Engenharia Mecânica, Universidade Federal do Pará, 2015.

Ferguson, C. R.; Kirkpatrick, A. T. *Internal combustion engines : applied thermodynamics*. 2nd. New York: John Wiley & Sons, 2001. xiii, 369 p.

Gardenghi, John Lenon C.; SANTOS, *Sandra Augusta. Sistemas não lineares via regiao de confiança: o algoritmo de Levenberg-Marquardt*. Relatório de Pesquisa, 2011.

Heywood, John B. *Internal Combustion Engine Fundamentals*. USA: McGraw-Hill, Inc., 1988.

Kumar, B. Rajesh *et al*. A comparative analysis on combustion and emissions of some next generation higher-alcohol/diesel blends in a direct-injection diesel engine. *Energy Conversion and Management*, v. 119, p. 246-256, 2016.

Loganathan, S. *et al*. Heat release rate and performance simulation of DME fuelled diesel engine using oxygenate correction factor and load correction factor in double Wiebe function. *Energy*, v. 150, p. 77-91, 2018.

Luna, André Luiz Carvalho. *Memorial de Instalação de Banco de Provas para Combustíveis Pesados*. Projeto de Graduação – Curso de Engenharia Mecânica, Universidade Federal do Rio de Janeiro, Rio de Janeiro, 2014.

Oliveira, R. A.. *Determinação da Pressão Média Efetiva Indicada de Motores Diesel*. Dissertação de mestrado, Universidade Federal do Pará, 2016.

Rajasekar, E.; Selvi, S. Review of combustion characteristics of CI engines fueled with biodiesel. *Renewable and Sustainable Energy Reviews*, v. 35, p. 390-399, 2014.

Ranganathan, Ananth. The levenberg-marquardt algorithm. *Tutoral on LM algorithm*, v. 11, n. 1, p. 101-110, 2004.

Rocha, Hendrick Maxil Zárate. *Efeitos da utilização de hidrogênio em geradores a diesel operando com diferentes misturas diesel-óleo vegetal*. 2016. Instituto Alberto Luiz Coimbra de Pós-Graduação e Pesquisa de Engenharia. Tese (Doutorado) – Universidade Federal do Rio de Janeiro, 2016.

Sun, Yongrui et al. Development and validation of a marine sequential turbocharging diesel engine combustion model based on double Wiebe function and partial least squares method. *Energy Conversion and Management,* v. 151, p. 481-495, 2017.

Turns, Stephen R.. *Introdução à Combustão Conceitos e Aplicações*. 3 ed. Porto Alegre: AMGH Editora Ltda. 2013.

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