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STUDIES OF THE INFLUENCE IN THE PERFORMANCE PARAMETERS IN SPARK IGNITION ENGINE OF TWO HEAT TRANSFER MODELS IN A PHENOMENOLOGICAL MODEL USING BIOMETHANE, BIOGAS AND ETHANOL

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Abstract. *Internal combustion engines represented a significant technological advance for humanity and significantly improved production and transportation. However, the unrestrained use of fossil fuels causes great damage to the planet due to the emission of greenhouse gases (GHGs), causing global warming. According to a study by the World Meteorological Organization (WMO), there is a 50% chance that the increase in global average temperature will exceed 1.5℃ in the five years starting from 2022. Through this problem, several alternative energy sources have been developed to replace conventional fuels, such as ethanol, biogas and biomethane, discussed in this work. Despite the burning of ethanol and biomethane emitting CO_2, there are compensations in the use of both from an environmental point of view. Ethanol is renewable because it is obtained from the plantation of vegetable raw material and is considered cleaner because, in the plant's life cycle, there is the absorption of CO_2. Biomethane and biogas are renewable because it is obtained from decomposing much organic matter in household waste, manure, and sewage sludge. It is considered less polluting because it favors better treatment of these materials. This work aims to evaluate two models of heat transfer through computer simulation. They are Hohenberg and Woschni models, applied in the phenomenological model in a engine using ethanol, biomethane, and biogas as fuels. The simulation was implemented in MATLAB® by González, 2010, and the phenomenological model used is the Ferguson model. Therefore, it is expected to analyze the impact of models on performance parameters by comparing them with experimental data from Berlini, 2017, using biogas and ethanol as fuels, and determining the best method. After that, to simulate the efficiency data using the best heat transfer model for biomethane. After the simulation it was possible to observe that the Hohenberg model presented the lowest errors compared to the Woschni model. From the use of only the Hohenberg model, ethanol was shown to be the fuel with the highest general values of maximum pressure, indicated average pressure and thermal efficiency, followed by biomethane, and finally biogas with the lowest values. Thus, enabling justifications for greater use of these renewable fuels through existing models, but not applied to the situations described in this work.*

Keywords: *heat transfer models, phenomenological model, spark ignition engine simulation, renewable fuels, biofuels*

1. INTRODUCTION

Global warming is a phenomenon in which an increase in the Earth's average temperature occurs due to human activities that emit greenhouse gases into the atmosphere. The main gases responsible for global warming are carbon dioxide (CO_2), methane (CH_4) and nitrous oxide (N_2O). According to the World Meteorological Organization (WMO, 2021), the decade from 2011 to 2020 was the warmest on record, with global average temperatures about 1.1 degrees Celsius above pre-industrial levels.

Pollutant emissions from fossil fuel use are one of the main causes of global warming and air pollution, and carbon dioxide emissions are the most significant and contribute directly to the increase in the greenhouse effect. According to the Global Carbon Project (Globo G1 and RFI, 2022), global CO_2 emissions related to the burning of fossil fuels and industrial processes reached a record high in 2019, totaling approximately 36.4 billion tons. Faced with this challenge, many alternative fuels are being studied and used in ways that reduce these emissions. In this work, three very promising fuels are studied: biomethane, biogas and ethanol.

Biomethane is a type of renewable natural gas produced from biological sources such as organic waste and can be used as a direct substitute for conventional natural gas, both for residential and commercial purposes and for use in vehicles. The biomethane production process involves the anaerobic decomposition of organic matter such as agricultural waste, food waste, sewage, animal manure and other biomass. This decomposition takes place under controlled conditions, such as in biodigesters or landfills, where organic matter is broken down by bacteria in an oxygen-free environment, producing biogas.

To produce biomethane, it is necessary to purify the biogas by removing impurities such as CO_2 and other compounds. This process is called "upgrading" and produces a chemical composition similar to that of conventional natural gas, but with traces of CO_2 that is usually less than 10%. Biomethane is considered a sustainable alternative for several reasons such as coming from waste that is renewable and continuously generated, reducing methane emissions and allowing the valorization of organic residues.

Ethanol is an organic chemical compound that belongs to the class of alcohols. Its production is mainly obtained from the fermentation of sugar or starch sources such as sugar cane, corn, sugar beet and cereals. Ethanol has environmental advantages, as burning it releases less carbon dioxide and other pollutants compared to fossil fuels. In addition, it can be produced from renewable sources, helping to reduce dependence on non-renewable fuels.

In engineering, to model a process is to use assumptions and equations to obtain data to characterize a process. According to Heywood (1988), for engines, there are two general types of models: thermodynamic and fluid-dynamic, depending on the predominance of the types of equations used. The phenomenological or zero-dimensional model is part of the group of thermodynamic models and consists of the addition of information beyond the balance equations for each phenomenon.

The objective of this work is to analyze the influence of the Woschni and Hohenberg models (compared in the work of Gallo, 1988), using biogas and ethanol in the Ferguson phenomenological model, on the efficiency parameters in an internal combustion engine, and finally applying the use of biomethane with 90% CH_4 and 10% CO_2.

2. METHODOLOGY

Figure 1 shows briefly how the work was developed.

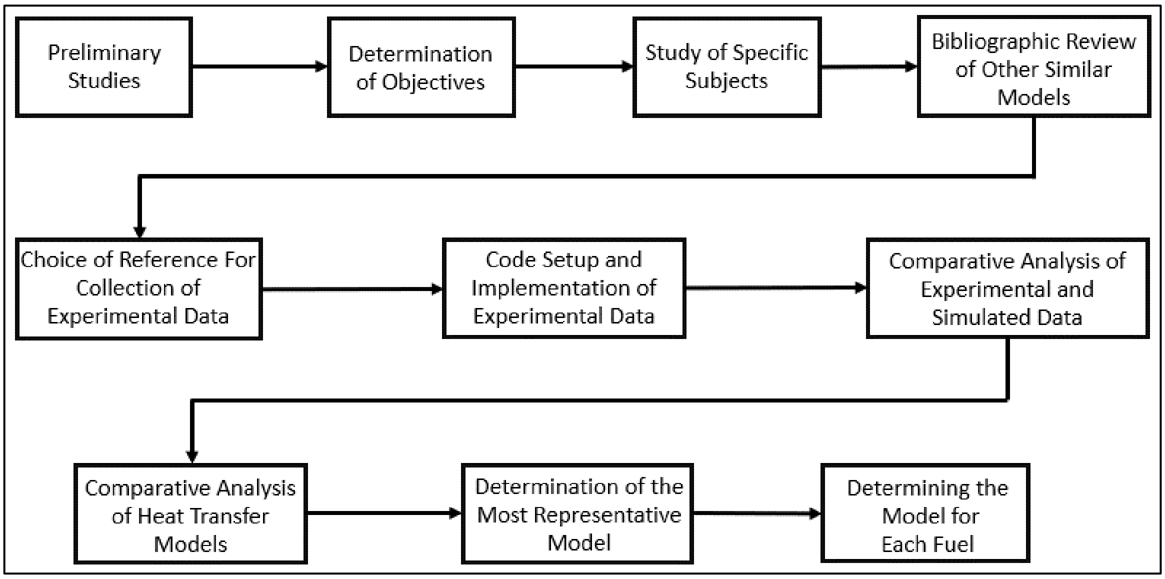

Figure 1. Flowchart of the work methodology.

The preliminary studies were focused on: the understanding of Ferguson's phenomenological model, based on the first law of thermodynamics; the study of the combustion reaction of ethanol and biogas; and the importance and use of renewable fuels addressed in the work. Ferguson's phenomenological model was initially implemented in FORTRAN®, however the code used was adapted for MATLAB®, and is based on the application of six differential equations (Eq. (1) to (6)), each represents the change of a variable as a function of the crankshaft angle [θ].

$$\frac{dP}{d\theta} = (A + B + C)/(D + E)\,, \tag{1}$$

$$\frac{dT_b}{d\theta} = \frac{-h\left(\frac{\pi b^2}{2}+\frac{4V}{b}\right)x^{\left(\frac{1}{2}\right)}(T_b-T_w)}{\omega m c_{Pb} x} + \frac{v_b}{c_{Pb}}\frac{\partial \ln v_b}{\partial \ln T_b}\left(\frac{A+B+C}{D+E}\right) + \frac{h_u-h_b}{x c_{Pb}}\left[\frac{dx}{d\theta} - (x-x^2)\frac{C}{\omega}\right], \tag{2}$$

$$\frac{dT_u}{d\theta} = \frac{-h\left(\frac{\pi b^2}{2}+\frac{4V}{b}\right)\left(1-x^{\left(\frac{1}{2}\right)}\right)(T_u-T_w)}{\omega m c_{Pu}(1-x)} + \frac{v_u}{c_{Pu}}\frac{\partial \ln v_u}{\partial \ln T_u}\left(\frac{A+B+C}{D+E}\right), \tag{3}$$

$$\frac{dW}{d\theta} = P\frac{dV}{d\theta}, \tag{4}$$

$$\frac{dQ_l}{d\theta} = \frac{h}{\omega}\left(\frac{\pi b^2}{2} + \frac{4V}{b}\right)\left[x^{\left(\frac{1}{2}\right)}(T_b - T_w) + \left(1 - x^{\left(\frac{1}{2}\right)}\right)(T_u - T_w)\right], \tag{5}$$

$$\frac{dH_l}{d\theta} = \frac{Cm}{\omega}\left[(x - x^2)h_u + x^2 h_b\right], \tag{6}$$

where $\boldsymbol{P}$ is the pressure, $\boldsymbol{T_b}$ is the temperature of the burned gases, $\boldsymbol{T_u}$ is the temperature of the unburned gases, $\boldsymbol{W}$ is the work, $\boldsymbol{Q_l}$ is the heat lost and $\boldsymbol{H_l}$ the lost enthalpy. The other symbols are described below:

$\boldsymbol{b}$ – cylinder bore;
$\boldsymbol{V}$ – cylinder volume;
$\boldsymbol{Cp_b}$ – specific heat at constant pressure of the burned gases;
$\boldsymbol{Cp_u}$ – specific heat at constant pressure of the unburned gases;
$\boldsymbol{h_b}$ – specific enthalpy of the burned gases;
$\boldsymbol{h_u}$ – specific enthalpy of the unburned gases;
$\boldsymbol{x}$ – mass fraction of burned gases;
$\boldsymbol{m}$ – total mass in the cylinder;
$\boldsymbol{\omega}$ – engine angular frequency.

$\boldsymbol{A}$, $\boldsymbol{B}$, $\boldsymbol{C}$, $\boldsymbol{D}$, and $\boldsymbol{E}$ are variables established to simplify the formulations and are defined by the following equations (Eq. (7) to (11)):

$$A = \frac{1}{m}\left(\frac{dV}{d\theta} + \frac{VC}{\omega}\right), \tag{7}$$

$$B = h\frac{\left(\frac{\pi b^2}{2}+\frac{4V}{b}\right)}{\omega m}\left[\frac{v_b}{c_{Pb}}\frac{\partial \ln v_b}{\partial \ln T_b}x^{\left(\frac{1}{2}\right)}\frac{T_b-T_w}{T_b} + \frac{v_u}{c_{Pu}}\frac{\partial \ln v_u}{\partial \ln T_u}\left(1 - x^{\left(\frac{1}{2}\right)}\right)\frac{T_b-T_w}{T_b}\right], \tag{8}$$

$$C = (-v_b - v_u)\frac{dx}{d\theta} - v_b\frac{v_b}{c_{Pb}}\frac{\partial \ln v_b}{\partial \ln T_b}\frac{h_u-h_b}{c_{Pb}T_b}\left[\frac{dx}{d\theta} - \frac{C(x-x^2)}{\omega}\right], \tag{9}$$

$$D = x\left[\frac{v_b^2}{c_{Pb}T_b}\left(\frac{\partial \ln v_b}{\partial \ln T_b}\right)^2 + \frac{v_b}{P}\frac{\partial \ln v_b}{\partial \ln P}\right], \tag{10}$$

$$E = (1-x)\left[\frac{v_u^2}{c_{Pu}T_u}\left(\frac{\partial \ln v_u}{\partial \ln T_u}\right)^2 + \frac{v_b}{P}\frac{\partial \ln v_u}{\partial \ln P}\right], \tag{11}$$

Ferguson, 2001, used the zero-dimensional model, and to determine the mass fraction burned as a function of crank angle the cosine function was used (Eq. (12)).

$$x_b\,(\theta) = 0.5\left[\frac{1-\cos(\pi(\theta-\theta_S))}{\Delta\theta_b}\right], \tag{12}$$

where $\boldsymbol{\theta_s}$ is the angular position of the crankshaft at the moment of the spark and $\Delta\boldsymbol{\theta_b}$ represents the angle equivalent to the duration of combustion.

As for the fuel data, the thermodynamic properties of ethanol are already implemented in the code taken from the thermodynamic tables of Heywood, 1988. The thermodynamic properties of biogas were calculated based on its chemical composition with 65% CH_4 and 35% CO_2.

The heat transfer analysis aims to obtain the amount of heat rejected to the cooling system. Therefore, heat transfer occurs mainly by means of convection and radiation. A quasi-permanent regime and constant wall temperature are also considered. The heat transfer models are composed of empirical equations whose main objective is to determine the film coefficient $\boldsymbol{h}$ required for the Eq. 5 lost heat variation. This work will use two heat transfer models: from Woschni (1967) and from Hohenberg (1979).

Woschni (1967) based on dimensional analysis, explicitly accounting for the influence of engine size, pressure and instantaneous temperature proposed Eq. (13). This model also considers that the characteristic velocity depends on the increase of combustion turbulence:

$$h = C_1 D^{-0,2} P^{0,8} T^{-0,53}\left[C_2 V_P + C_3\frac{V_S T_1}{(P_1 V_1)(P-P_O)}\right]^{0,8}\left[\frac{\mathrm{W}}{\mathrm{m^2K}}\right], \tag{13}$$

where $\boldsymbol{T_1}$ [K], $\boldsymbol{V_1}$[m/s] and $\boldsymbol{P_1}$[Pa] are some known conditions, $\boldsymbol{V_S}$[$\mathrm{m^3}$] is the cylinder capacity, $\boldsymbol{V_p}$[m/s] is the average speed of the piston and $\boldsymbol{P_O}$[Pa] is the pressure when there is no combustion. For the constants $\boldsymbol{C_1}$, $\boldsymbol{C_2}$ and $\boldsymbol{C_3}$ the values 110, 2.28 e 3.24×10^3 were assumed respectively, according to the author's experimental analysis.

Hohenberg (1979) modified the expression of Woschni, 1967 by discarding the combustion-dependent characteristic velocity term and worked with the instantaneous volume with characteristic dimension, therefore proposing Eq. (14).

$$h = C_1 V^{-0,06} P^{0,8} T^{-0,4} \left(V_p + C_2\right)^{0,8} \cdot \left[\frac{\mathrm{W}}{\mathrm{m^2K}}\right], \tag{14}$$

where the values of 130 and 1.4 were adopted for $\boldsymbol{C_1}$ and $\boldsymbol{C_2}$ respectively, according to the author's experimental analysis.

The experimental data were taken from the work of Berlini, 2017. Table 1 shows the engine specifications, and the volumetric ratio was modified according to the geometric analysis made in the work.

Table 1. Engine Specifications (Berlini, 2017).

Tipo de Specification	Specification
Model	AVL 5495
Number of cylinders	1
Bore (mm)	82
Stroke (mm)	86
Connecting rod length (mm)	144
Modified volumetric ratio	13.6:1
Displaced volume (cm^3)	454.16

The volumetric efficiencies assumed were 70% for 1800 rpm and 74% for 3600 rpm, as they are values close to those simulated by GONZÁLES (2010). The efficiency parameters considered were maximum pressure ($Pmax$), indicated pressure ($IMEP$) and thermal efficiency. Table 2 shows the relevant input data for the simulation.

Table 2. Experimental input properties.

Properties	Ethanol 1800 rpm	Ethanol 3600 rpm	Biogas 1800 rpm	Biogas 3600 rpm
Duration of combustion, [°] (1)	19.6	23.5	29.2	33.6
Start of combustion, [°] (1)	-9.5	-11.7	-17.8	-23.7
Inlet pressure, bar (1)	0.96	0.96	0.97	0.97
Inlet temperature, K (1)	306.0	307.0	301.0	301.0
Equivalence ratio (1)	1.1	1.0	1.0	1.0
Volumetric efficiency	0.70	0.74	0.70	0.74

(1) data from Berlini, 2017

3. RESULTS AND DISCUSSIONS

Tables 3 and 4 show the results of the simulations and the comparison with the experimental data.

Table 3. Simulation result for 1800 rpm.

Parameters	Woschni					
	Ethanol			Biogas		
	Experimental (1)	Simulated	Error (%)	Experimental (1)	Simulated	Error (%)
Pmax (bar)	57.40	67.08	16.86	56.40	60.40	7.09
IMEP (bar)	8.86	7.26	18.06	7.65	6.00	21.57
Thermal Efficiency	48.40	40.89	15.52	50.10	34.37	31.40
	Hohenberg					
	Ethanol			Biogas		
	Experimental (1)	Simulated	Error (%)	Experimental (1)	Simulated	Error (%)
Pmax (bar)	57.40	67.68	17.91	56.40	61.16	8.44
IMEP (bar)	8.86	7.63	13.88	7.65	6.34	17.12
Thermal Efficiency	48.40	43.00	11.16	50.10	36.32	27.50

(1) data from Berlini, 2017

As can be noted, the Hohenberg (1979) model obtained results closer to the experimental values ensuring lower error compared to the Woschni (1967) model. It can also be observed that the model had greater error when calculating the thermal efficiency of biogas, with errors greater than 20%. This is probably due to the lack of some parameters among the experimental data, such as the volumetric efficiency and the cylinder wall temperature.

Table 4. Simulation result for 3600 rpm.

Parameters	**Woschni**					
	Ethanol			**Biogas**		
	Experimental (1)	Simulated	Error (%)	Experimental (1)	Simulated	Error (%)
Pmax (bar)	57.00	68.74	20.60	59.60	67.73	13.64
IMEP (bar)	9.61	8.11	15.61	8.10	6.75	16.67
Thermal Efficiency	47.70	43.77	8.24	49.40	36.57	25.97
	Hohenberg					
	Ethanol			**Biogas**		
	Experimental (1)	Simulated	Error (%)	Experimental (1)	Simulated	Error (%)
Pmax (bar)	57.00	68.62	20.39	59.60	67.92	13.96
IMEP (bar)	9.61	8.20	14.67	8.10	6.82	15.80
Thermal Efficiency	47.70	44.23	7.27	49.40	36.91	25.28

(1) data from Berlini, 2017

It is also observed that the model shows higher values of maximum pressure (Figures 2 and 3), and indicated pressure for ethanol compared to biogas. It is also important to mention that the cosine law model is not the best combustion model, and the Ferguson model is for the closed phase of the engine (compression, combustion and expansion). In addition, this model had to assume values for volumetric efficiency.

Given so many variables, the results show trends in the parameters in question. One point to note is that it is unlikely that an ICE engine operating with biogas or ethanol will achieve high thermal efficiencies as presented in the experiments of Berlini (2017), and the modeling results are more consistent with the usual ranges of engines operating with these fuels.

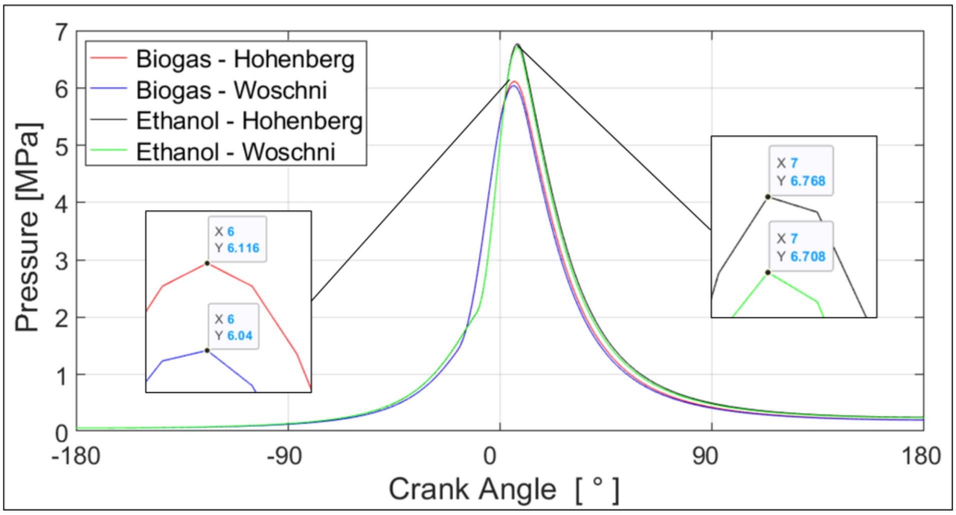

Figure 2. Pressure *versus* Crank angle for 1800 rpm using Ethanol and Biogas.

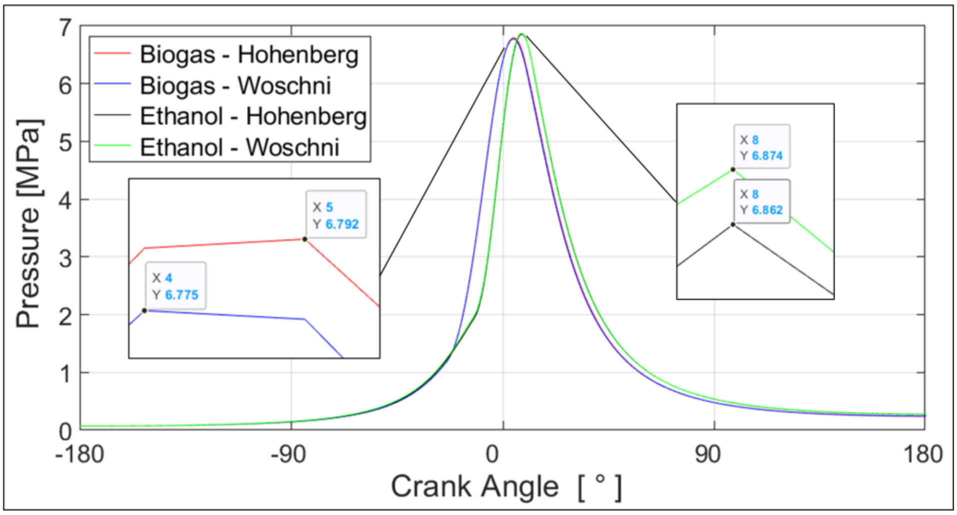

Figure 3. Pressure *versus* Crank angle for 3600 rpm using Ethanol and Biogas.

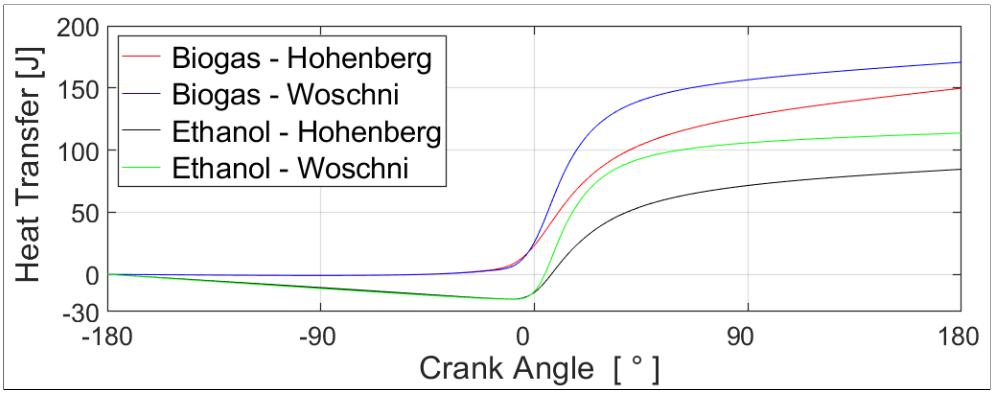

Figure 4. Heat Transfer *versus* Crank angle for 1800 rpm using Ethanol and Biogas.

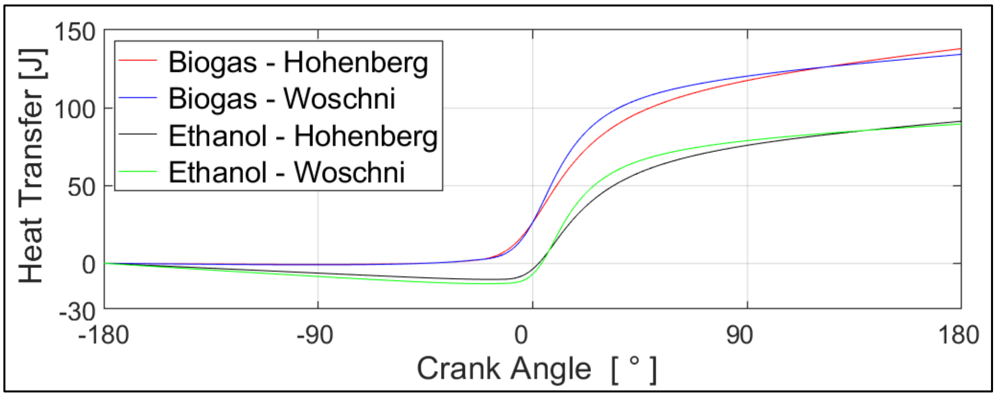

Figure 5. Heat Transfer *versus* Crank angle for 3600 rpm using Ethanol and Biogas.

When analyzing the heat transfer (Figures 4 and 5) for the rotations 1800 rpm and 3600 rpm, there was a convergence of the values in the graphs using the Woschni and Hohenberg model, demonstrating that both models better describe this parameter the higher the rotation. Based on the conclusion that the best heat transfer model was the Hohenberg model, the simulation with biomethane was performed only with this model. The results are in Table 4 and Figures 6 and 7.

Table 4. Result of performance parameters for biomethane using the Hohenberg model.

Parameter	**1800 rpm**	**3600 rpm**
Pmax [bar]	63.54	70.54
IMEP [bar]	6.64	7.14
Thermal efficiency [%]	38.89	39.54

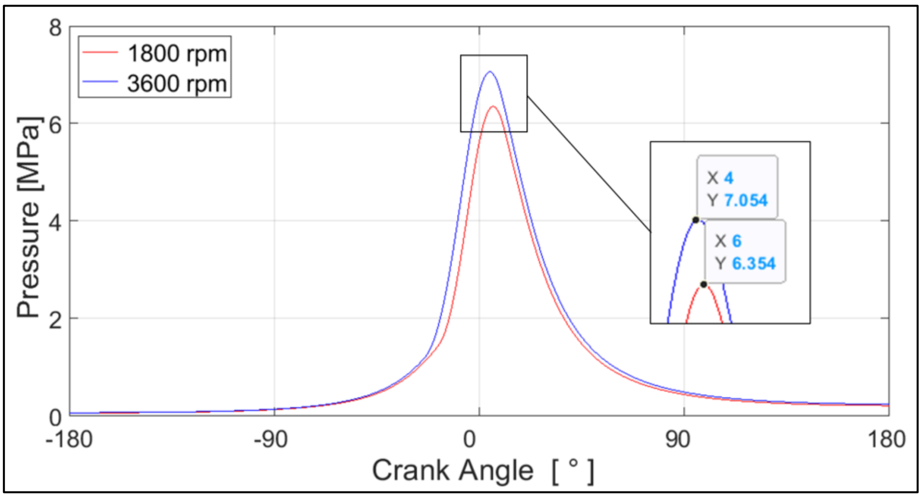

Figure 6. Pressure *versus* Crank angle for biomethane at 1800 rpm and 3600 rpm.

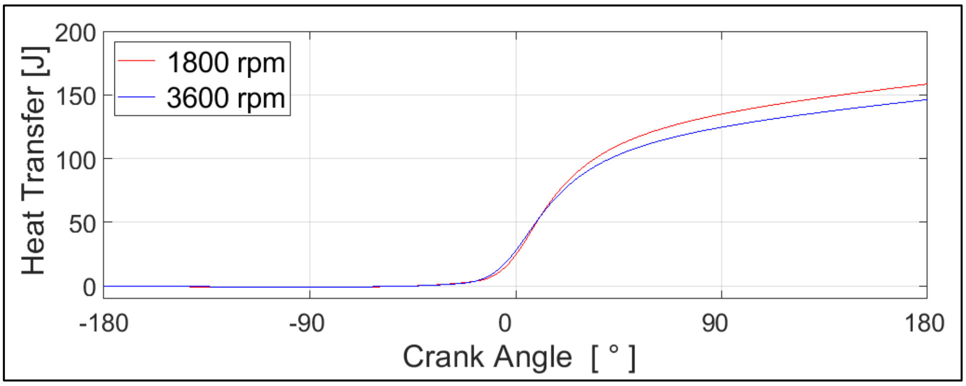

Figure 7. Heat Transfer *versus* Crank angle for biomethane at 1800 rpm and 3600 rpm.

Based on the results found for biomethane, the values found are pertinent, regarding the trend, as biomethane presents a half amount of methane, it was expected that the thermal efficiency would be higher as well as the peak pressure and IMEP. It is also expected that the peak pressure and IMEP for high revolutions are higher (Figure 6). When the engine speed is lower there is more time for a greater heat exchange and this can be observed in Figure 7, where there is a higher value for 1800 rpm (Figure 7).

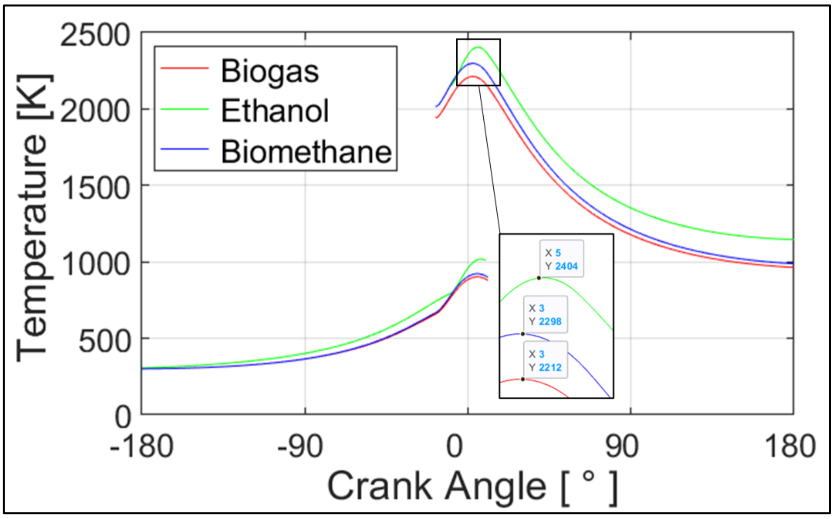

Figure 8. Temperature *versus* Crank angle at 1800 rpm.

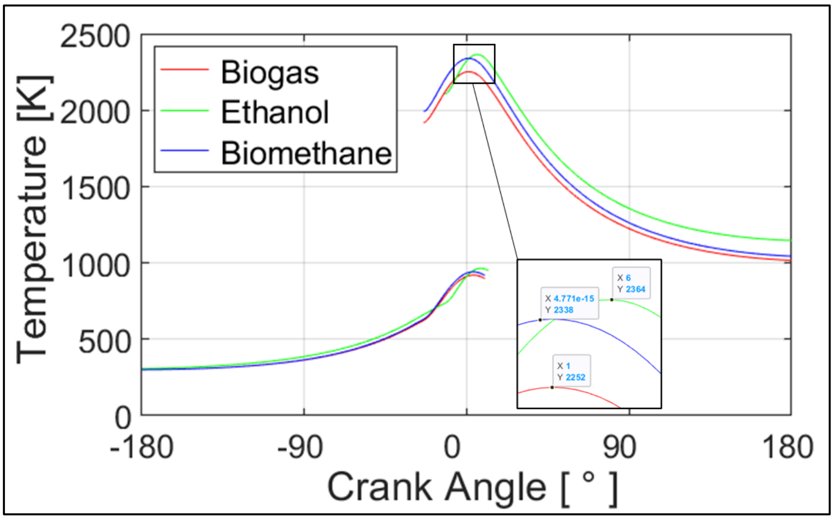

Figure 9. Temperature *versus* Crank angle at 3600 rpm.

Regarding temperature, the curve with temperatures mostly below 1000 K refers to unburned gases and the curve with temperatures mostly above 1000 K refers to burned gases. It is possible to observe in Figures 8 and 9 higher values for ethanol, then for biomethane, and finally the lowest temperatures for biogas. It is worth mentioning that for the two rotations there are no major variations, but unlike biogas and biomethane, ethanol increases its peak temperature with increasing rotation.

4. CONCLUSIONS

The computer simulation, even though it does not generate exact results, due to the lack of experimental data and the use of a non-complete model, is able to show a general picture, and depending on its complexity, capable of generating data very close to the real ones. In this work it was observed that for the pressure parameters the data were considerably close to the experimental ones even with the lack of certain input variables. It was also possible to verify the superiority in the analyzed performance parameters of ethanol in relation to biogas, and biomethane at 1800 rpm, but the maximum pressure of biomethane at 3600 rpm even exceeds that of ethanol under these same conditions. It is worth mentioning that it was possible to verify the superiority of biomethane in all parameters compared to biogas. Finally, it was found that the Hohenberg model proved to be better for determining these parameters in the model used because it presented the smallest errors.

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