OPERATION CYCLE ANALYSIS OF A CFR CETANE ENGINE

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Abstract. *This paper presents the test results of a ASTM CFR diesel engine with variable compression ratio, measuring power, efficiency, fuel consumption, maximum combustion pressure, combustion profile, ignition delay, indicated medium pressure, indicated pressure versus crankshaft angle graphic. The experimental results are compared to those obtained in a process simulation model and also in a air fuel cycle. Several conclusion and recommendation are presented.*

Keywords*. CFR Cetane Engine, Operation Cycle, Simulation*

1. Introduction

Variable strokes engines are gaining attention by researchers and automobile manufactures for their fuel economy advantage. In fixed stroke engines load variation is balanced out by throttling the intake fuel-air mixture. This approach leaves the pumping and frictional loses unchanged since the stroke remained constant. However, in variable stroke engine these looses are reduced since a short stroke is used for low engine load and longer strokes are used for higher engines loads.

The standard ASTM variable stroke diesel engine is called ASTM CFR engine. It is used basically for measuring the knock characteristic of diesel fuel, measured by its cetane number. A good diesel engine fuel should have short ignition lag and will ignite more readily and the cetane number should be sufficiently high to avoid knock. So, the engine has already a pressure sensor inside the combustion chamber that, when associated to the piston position in the cylinder, will provide a proper measure of the several variables of the operation cycle of the engine.

A full displacement analysis is performed where the piston displacement and connecting rod angle are related to the crank angle for several stroke length (compression ratio) arrangements. A simulation model for the engine power cycle is developed in order to compare to the engine experimental results and also to a air-fuel cycle. Different fuels are also presented in the operation cycle of the engine.

Different power and efficiency characteristics are computed and presented for different stroke lengths and the corresponding compression ratio.

2. The operation cycle analysis

The cycle analysis of the CFR engine is performed by obtaining experimentally a graphic representation of the cycle in function of cylinder pressure and volume (V) inside the cylinder. For improving experimental results, we can also present the variable V in function of crankshaft angle.

Figure (1) shows the geometry of the ASTM CFR cetane engine.

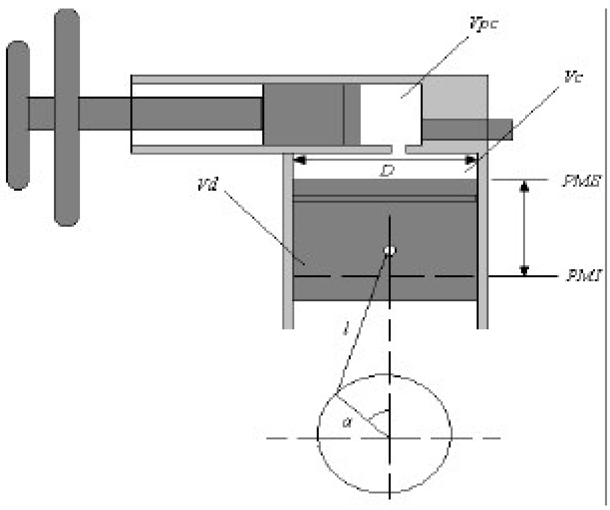


Figure 1. Geometry of ASTM CFR cetane engine.

Simbology:

Vpc – pre-combustion chamber volume;
Vc – clearance volume;
Vd – displacement or swept volume;
l – connecting rod length;
D – bore;
a - crankshaft radius;
BM – *l*/a
θ - crankshaft angle;

In Fig. (1), we can define the displacement (Vd) of the engine by the following expression:

$$V_d = \frac{p.D^2}{4}.l \tag{1}$$

And also, the compression ratio (rc) of the engine will be:

$$rc = \frac{Vd + Vc}{Vc} \tag{2}$$

Or:

$$rc = \frac{Vd + (Vc + Vpc)}{(Vc + Vpc)} \tag{3}$$

The instantaneous volume in function of angle crankshaft, will be:

$$V = \left[1 + \frac{1}{2}.(rc - 1).\left[BM + 1 - \cos(q) - \left(BM^2 - \mathrm{sen}(q)^2\right)^{\frac{1}{2}}\right]\right].(Vc + Vpc) \tag{4}$$

Taking into account those relations, the know P-V diagram can be transformed to a P-θ diagram showed in Fig. (2), which represents the operation cycle inside the CFR diesel engine.

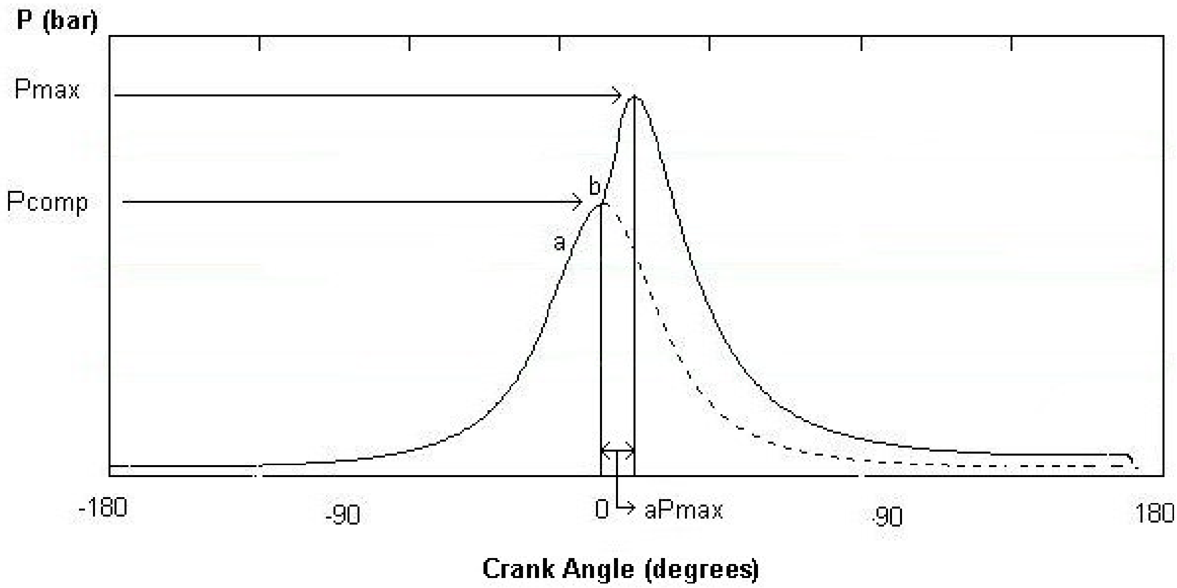


Figure 2. Pressure Vs crankshaft angle (P-θ) diagram.

In the Fig. (2), we have:
Pmax – maximum combustion pressure;
Pcomp – maximum compression pressure;
aPmax – angle of Pmax (related to top dead center taken as reference 0^o);
a – b – ignition delay (period between the injection of fuel and the beginning of combustion);

The engine cycle represented by the Fig. (2) can be obtained experimentally. We can also develop a simulation model of the cycle and the results of this model can be compared with the experimental results obtained. This procedure will be described in this paper.

3. Apparatus and basic procedure

An ASTM CFR engine (Walkesha manufactory), with one (1) cylinder, was selected for the study of operational cycle. The engine was tested with standard diesel fuel and also with palm oil as a alternative fuel, with the same characteristic described in Pimentel et all (1999, 2000).

When tested with palm oil, the fuel is pre-heated by an electric heater to the admission temperature of 85 ^oC in engine admission (Pimentel et all, 1999, 2000) in order to decrease its viscosity to a value near to the diesel oil at ambient temperature.

An electric motor is coupled to the CFR engine in order to start the engine and also provides load variation at constant speed (rpm).

The following measurements instruments are provided:

1- Volumetric type flow meter, to measure fuel consumption by the Burette method (ASTM D613) (Soares et all, 2000);
2- Lubricating oil temperature thermocouple;
3- Admission air temperature thermocouple linked to a electric resistance in order to maintain a constant value of the air temperature;
4- Engine water cooling temperature;
5- Exhaust gas thermocouple;
6- Palm oil admission air temperature thermocouple that is also linked to a electric heater in order to maintain a constant value of the oil admission temperature;
7- Admission air pressure sensor;
8- Cylinder pressure transducer;
9- Crankshaft angle measurement device;
10- RPM measurement device;
11- Exhaust gases analyzer;
12- Data acquisition system;

For obtaining the curve of pressure (P) versus crankshaft angle (θ)experimentally, we used the AVL 364 angle encoder, described in Fig. (3).

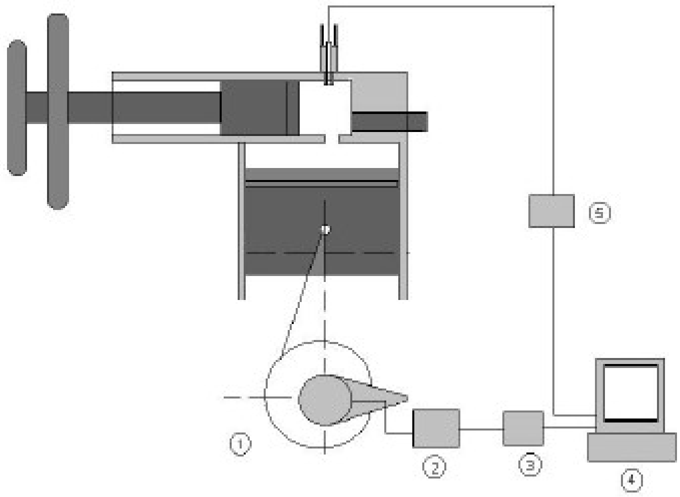


Figure 3. System measurer for the curve pressure vs crankshaft angle (P-θ).

In the Fig. (3), we have:

1- Crankshaft angle sensor;
2- Signal converter;
3- Amplifier;
4- Acquisition data module and display;
5- Amplifier;
6- Pressure transducer;

The tests were performed for three different compressions ratio of the engine. The standard operational conditions are:

- RPM: 900 ± 9;

- Injection advance: 13^o before TDC;
- Injection pressure: 10.3 ± 0.34 Mpa;
- Fuel flow: 13 ± 0.2 ml/min;
- Lubrication oil pressure: 0.17 a 0.20 Mpa;
- Lubrication oil temperature: 57 ± 8 C;
- Admission air temperature: 66 ± 0.5 C;
- Cooling water temperature on the injector: 38 ± 3 C;
- Cooling water temperature in the exit of the engine: 100 ± 2 C;

4. Thermodynamic simulation model

The model adopted in the simulation can be categorized as thermodynamic whether the equations, which give the model, its predominant structure based on energy conservation (Ricardo and Hempson, 1988). It considers the operation cycle transformations where there is more significant work variation as compression, combustion and expansion.

It is also considered as zero-dimensional, because the independent variable is only time (or crankshaft angle). The analysis of the pressure inside the cylinder as function of crankshaft angle during compression and expansion provides the necessary input data to model the combustion process of the engine. The zero-dimensional models analyze the combustion process through the heat release rate and with the data of the P-θ curves obtained experimentally in the engine (Ricardo and Hempson, 1988,Lakshminarayanan et all, 1986, Lavoie et all, 1980, Watson et all, 1980).

In the model adopted, the working fluid is an ideal gas and the composition and temperature are uniform in its volume.

4.1. Compression

The model uses the pressure and temperature relations for a polytropic transformation, taking into account the vaporization of the fuel in the process. So we will have the Eq. (5):

$$PV^{kr} = \text{constant} \quad (5)$$

Where:
P = pressure;
V = volume;
kr = Polytropic coefficient (1.32 to 1.4 for diesel engines (Gallo, 1991));

4.2. Ignition delay

The model used the empirical formulations from Haedenberg and Hase (1979) to predict ignition delay, that showed up to be the most adequate to the CFR indirect injection engine. However, the experimental data obtained from the engine, required that the expression should be corrected by a term in function of the compression ratio.

Then we have, that the ignition delay in function of θ, will be [r(θ)], as showed in the Eq. (6).

$$t(q) = (0.36 + 0.22.sp).\left[\exp\left[Ea.\left[\left(\frac{1}{R.T}\right) - \left(\frac{1}{17190}\right)\right]\right].\left[\frac{21.2}{(p-12.4)}\right]^{0.63}.\left(\frac{rc}{12.36}\right)^{0.85}\right]\right] \quad (6)$$

Where:
rc = Compression ratio;
kr = Polytropic coefficient;

4.3 - Combustion

Cylinder pressure versus crankshaft angle data over the compression and expansion strokes of the engine operating cycle can be used to obtain quantitative information on the progress of combustion (Ganesan, 1995). Suitable methods of analysis that yield the rate of release of the fuel’s chemical energy (heat release) or rate of fuel burning through the diesel engine combustion process are adopted (Heywood, 1988).

They start with the first law of thermodynamics for an open cycle, which is quasi-static (uniform in pressure and temperature) that became:

$$\frac{dQ}{dt} - p\frac{dV}{dt} + \sum m_i.h_i = \frac{dU}{dt} \quad (7)$$

Where:

dQ/dt is the heat transfer rate across the system boundary into the system;
p(dV/dt) is the rate of work transfer done by the system boundary displacement;
m_i is the mass flow rate into the system across the system boundary at location i;
h_i is the enthalpy of flux *I* entering or leaving the system;
U is the energy of the material contained inside the system boundary.
However, several problems make the direct application of the Eq. (7) to diesel combustion difficult:

1- Fuel is injected into the cylinder at last part of the compression process. The liquid added to the cylinder mixes with heated air and vaporizes and produce a fuel/air ratio distribution, which is nonuniform and varies with time. When the mixture reaches the auto-ignition temperature, the combustion begins in several different part of the combustion chamber at different times. The process in not quasi-static;
2- The composition of the burned gases is nonuniform;
3- The accuracy of available correlations for predicting heat transfer in diesel engines in not well defined;
4- Volumes among the piston, rings and cylinders wall constitute a few percent of the clearance volume. The gas in these regions is cooled to close to the wall temperature, increasing its density and, therefore, the relative importance of those spaces, increasing heat transfer.

We have seen that in zero-dimensional model the heat transfer has to be predicted. Wiebe (1970) presented a semi-empirical equation for heat transfer rate for engines in the case that mixture begins outside the cylinder. This not the real process in diesel engines, where the combustion happens in two stages. Watson et all (1980) presented a zero-dimensional model which the heat transfer rate is expressed by two added components, one relative to the combustion of the initial mixture and the other due to diffusion. They are functions of the operational variables and construction characteristic of the diesel engine.

The Wiebe function, developed by Watson et all (1980), in its differential form, can be expressed by the Eq. (8), Eq. (9) and Eq. (10):

$$\frac{dQ}{dq} = c_{desc}\frac{dQ_{desc}}{dq} + \left(1 - c_{desc}\right)\frac{dQ_{cont}}{dq} \tag{8}$$

$$\frac{dQ_{desc}}{dq} = 6.908\frac{m_{desc}+1}{\Delta q_{desc}}\left(\frac{q-q_c}{\Delta q_{desc}}\right)^{m_{desc}} \exp\left\{-6.908\left(\frac{q-q_c}{\Delta q_{desc}}\right)^{m_{desc}+1}\right\} \tag{9}$$

$$\frac{dQ_{cont}}{dq} = 6.908\frac{m_{desc}+1}{\Delta q_{cont}}\left(\frac{q-q_c}{\Delta q_{cont}}\right)^{m_{cont}} \exp\left\{-6.908\left(\frac{q-q_c}{\Delta q_{cont}}\right)^{m_{cont}+1}\right\} \tag{10}$$

Where:
c_{desc} = fraction of the fuel burned in the rapid combustion period;
Dq_{desc} = duration, in crank angle, of the rapid combustion period;
Dq_{cont} = duration, in crank angle, of the controlled combustion period;
m_{desc} = form factor of the rapid combustion;
m_{cont} = form factor of the controlled combustion;

4.4. Expansion

The model adopted the relations of pressure and temperature for a polytropic process, give by Eq. (11).

$$PV^{ke} = \text{constant} \tag{11}$$

Where:
ke = Polytropic coefficient, that varies between 1.18 to 1.23 for diesel engines (Ricardo and Hempson, 1988);

5. The fuel-air cycle

The fuel-air cycle studied in this proper is presented in Gallo (1981), that uses several parameters obtained in experimental results performed in a universe of engines. The method presents the calculation of the significant points of the operational cycle and also the indicated mean effective pressure (Pam i), indicated power and efficiency. It uses exhaust gases emissions data and operational and constructive parameters of the engine.

6. Experimental and model correlation results

A study was conducted with the basic aim to compare the pressure versus crank angle curve and also the operational cycle parameters of the experimental results, the simulation model and fuel-air cycle.

The geometric parameters of the engine where the same input data and the engine was tested with diesel oil and palm oil, under the same conditions and the compression ratio was fixed to the values of *11.92*, *12.36* and *13.96*. Table (1) presents the input data and Tab. (2) presents the combustion evolution according to Wiebe model.

Table 1. Input Data

Geometry of the Engine	**Data**		
Cylinder Diameter, Bore (mm)	83		
Stroke (mm)	114		
Compression ratio	11,91:1	12,36:1	13,86:1
l/a relation:	4,44		
Pre-chamber volume (m^3)	$4.291.10^{-5}$		
Piston speed (mm/s)	3.4		
Operational Parameters of the Engine	**Data**		
RPM	900		
Air admission temperature (^oC)	66		
Air admission pressure (kPa)	0.8		
Polytropic coefficient	1.36		
Fuel Data	**Diesel**		**Palm Oil**
Low heating value (kcal/kg)	10.150		8.733
Cetane number	47		45
Perfect gas Constant, R (kgf.m/kg.K)	29.3		
Fuel Flow (ml)	13		13.8
Specific gravity (kg/m^3)	832		864

Table 2. Wiebe model combustion evolution.

Diesel	Compression Ratio (CR) 11.92	Compression Ratio (CR) 12.36	Compression Ratio (CR) 13.96
Dq_{desc}	12	13	10
m_{desc}	2.4	2.6	2.7
Dq_{cont}	81	7.9	85
m_{cont}	0.95	1.2	0.9
χ_{desc}	0.62	0.74	0.8

Palm Oil	Compression Ratio (CR) 11.92	Compression Ratio (CR) 12.36	Compression Ratio (CR) 13.96
Dq_{desc}	16	18	12
m_{desc}	3.5	3.5	4
Dq_{cont}	69	69	62
m_{cont}	0.7	0.95	1
χ_{desc}	0.6	0.61	0.62

The Figs. (4) to (15) presents the correlation curves between the simulation model with palm oil and with diesel oil as well as correlation between experimental and simulation model results for the engine running with diesel oil and palm oil.

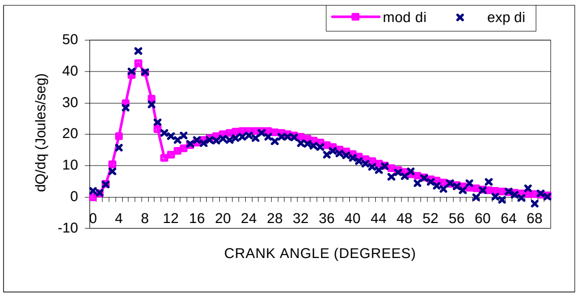


Figure 4. Evolution of diesel combustion – CR: 12.36.

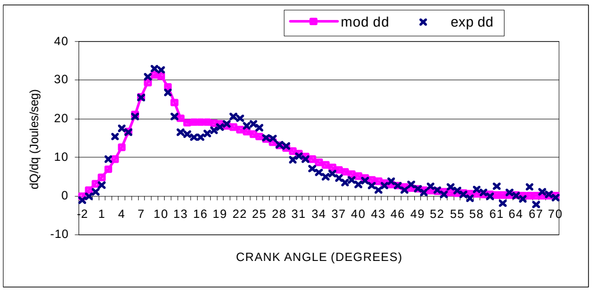
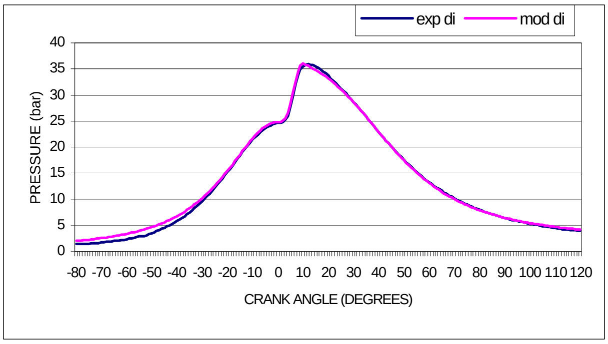


Figure 5. Evolution of palm oil combustion – CR: 12.36.



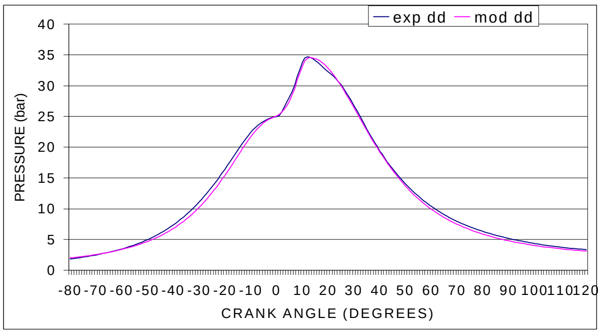
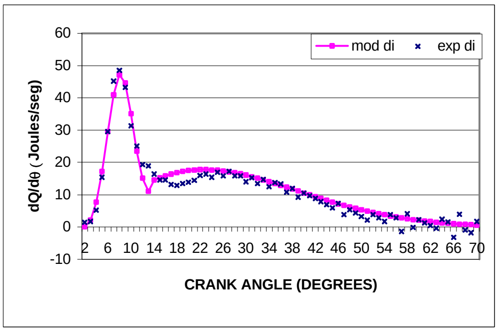


Figure 6. Pressure Vs Crank angle (diesel) – CR: 12.36.

Figure 7. Pressure Vs Crank angle (palm oil)– CR: 12.36.



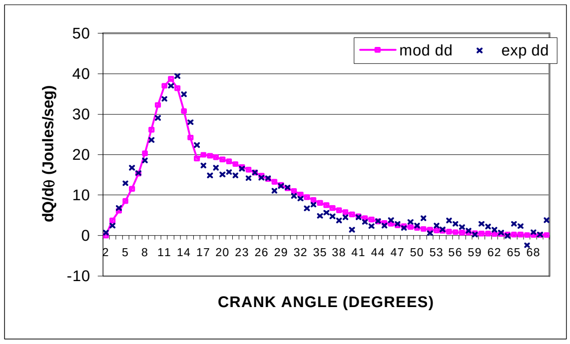
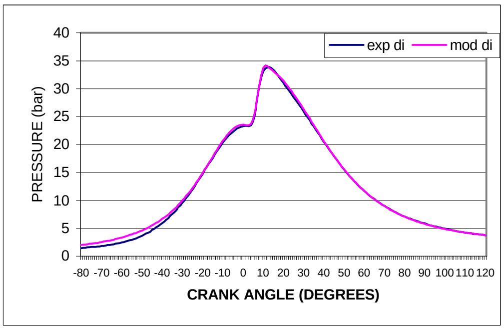


Figure 8. Evolution of diesel combustion – CR: 11.91.

Figure 9. Evolution of palm oil combustion – CR: 11.91.



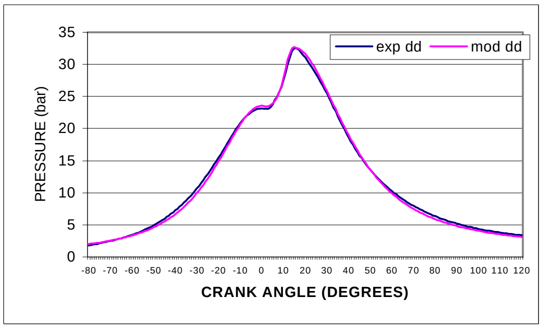
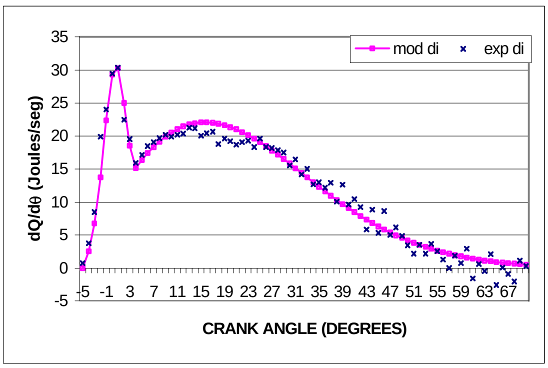


Figure 10. Pressure Vs Crank angle (diesel) – CR: 11.91.

Figure 11. Pressure Vs Crank angle (palm oil) – CR: 11.91.



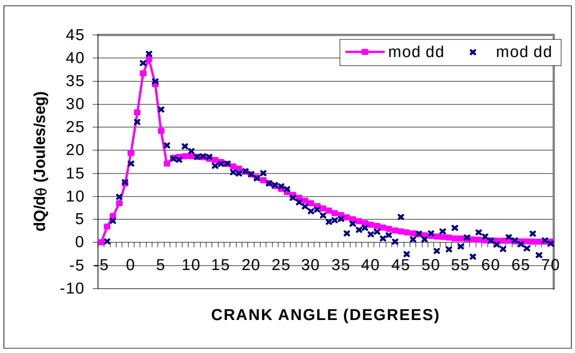


Figure 12. Evolution of diesel combustion – CR: 13.86.

Figure 13. Evolution of palm oil combustion – CR: 13.86.

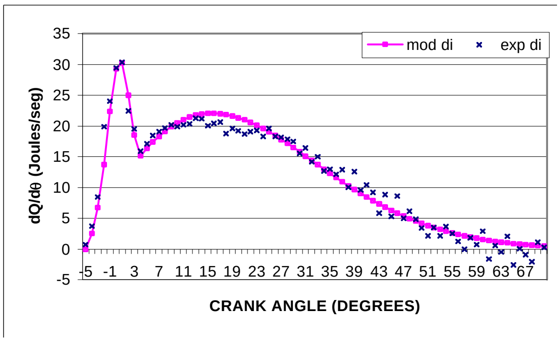


Figure 14. Evolution of pressure (diesel) – CR: 13.86.

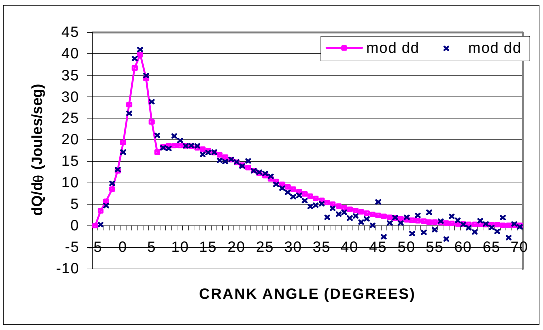


Figure 15. Evolution of pressure (palm oil) – CR: 13.86.

Table (3) presents the same results of the Figs. (4) to (15), with the values of several parameters for comparison.

Table 3. Experimental and simulation model results

Compression Ratio 12.36	**Experimental Results - Diesel**	**Simulation Model - Diesel**	**Air-Fuel Cycle**	**Experimental Results Palm Oil**	**Simulation Model Palm Oil**
Max. Combustion pressure (bar)	35.86	35.81	-	34.62	34.86
Angle of max combustion pressure (degrees)	372	371	-	372	372
Ignition delay (degrees)	13	13.7	-	15	14.7
Power (kW)	2.83	2.71	3.02	2.55	2.39
Specific fuel consumption (g/kWh)	229.26	239.19	213.2	286.95	305.02
Compression Ratio 11.91	**Experimental Results - Diesel**	**Simulation Model - Diesel**	**Air-Fuel Cycle**	**Experimental Results Palm Oil**	**Simulation Model Palm Oil**
Max. Combustion pressure (bar)	34.29	34.19	-	32.53	32.49
Angle of max combustion pressure (degrees)	373	371	-	376	375
Ignition delay (degrees)	17.25	15.7	-	18	16.5
Power (kW)	2.464	2.35	3.22	2.43	2.26
Specific fuel consumption (g/kWh)	263.41	275.77	201.3	301.02	319.41
Compression Ratio 13.96	**Experimental Results - Diesel**	**Simulation Model - Diesel**	**Air-Fuel Cycle**	**Experimental Results Palm Oil**	**Simulation Model Palm Oil**
Max. Combustion pressure (bar)	42.82	43.12	-	44.88	44.96
Angle of max combustion pressure (degrees)	371	371	-	370	370
Ignition delay (degrees)	10	10.6	-	10	11.3
Power (kW)	3.133	3	3.07	2.624	2.52
Specific fuel consumption (g/kWh)	207.10	216.01	211.13	278.93	290.42

7. Analysis of the results – conclusions

We can notice that acceptable correlations of results are presented in Tab. (3) and in the Figs. (4) to (15). The ignition delay values presented better correlation for high compression ratio.

The injection time observed experimentally was introduced as a input data for the thermodynamic model calculator, which were then independent of the ignition delay calculated.

In Tab. (3) we can notice that the duration of the rapid combustion period for the engine running with palm oil is 36% higher than the value presented with diesel oil, due to the higher ignition delay of the palm oil.

8. Acknowledgements

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