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ABCM Associação Brasileira de Engenharia e Ciências Mecânicas

18th Brazilian Congress of Thermal Sciences and Engineering
November 16–20, 2020 (Online)

ENC-2020-0206
NUMERICAL ANALYSIS OF A SINGLE-CYLINDER COMPRESSION IGNITION ENGINE FUELED WITH DIESEL AND STRAIGHT SOYBEAN OIL-DIESEL BLENDS

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Abstract. *This work reports a numerical approach and validation against experimental data available in the literature of performance parameters and heat release rate of a compression-ignition internal combustion engine fueled with fossil diesel oil and straight soybean oil-diesel blends. Four fuel blends named 100D(25°C), 50S50D(25°C), 50S50D(85°C) and 80S20D(85°C) regarding the soybean and diesel volumetric percentage and fuel temperature were simulated for 1800 rpm, 2100 rpm and 2200 rpm engine speeds and brake mean effective pressure conditions as in tests on dynamometric bench. The AVL-BOOST™ package was the computational tool used in this work. Performance parameters as torque, power, mean effective pressure (MEP), specific fuel consumption (sfc), volumetric efficiency, thermal efficiency and fuel mass injected, as well as the heat release rate, are then presented. The developed AVL-BOOST™ engine model involves all the main engine components in terms of intake and exhaust collectors, including pipe lines, plenums, cylinder, intake and exhaust valves geometrical parameters and operation. The numerical results of this work show good agreement with experimental data, with experimental to numerical errors (average values) of around* $\pm 0.4\%$ *for Diesel oil,* $\pm 1.2\%$ *for Soybean oil / Diesel oil - 50% / 50% (25°C),* $\pm 2.1\%$ *for Soybean oil / Diesel oil - 50% / 50% (85°C) and* $\pm 1.7\%$ *for Soybean oil / Diesel oil - 80% / 20% (85°C). Absolute errors are then listed for all parameter and conditions.*

Keywords: *Numerical simulation, Compression-ignition internal combustion engines, AVL-Boost™ , Straight soybean oil, Diesel oil.*

1. INTRODUCTION

Researches are carried out daily on internal combustion engines (ICE) with the intention of improving their operations when the biofuel is used as the main fuel or blended with fossil fuels. For the compression-ignition internal combustion engine (CI-ICE), the most used biofuels are the biodiesels and the vegetable oils. Several review works can be found in the technical literature, characterizing the different feedstocks, biofuel production process, the vantages and disadvantages, classification by groups as edible, non-edible, animal fats or oils from other sources. (Mishra and Goswami, 2018; Ma and Hanna, 1999; Aransiola *et al.*, 2014; Singh *et al.*, 2020; Chozhavendhan *et al.*, 2020). In this way, it is also possible to find different ICE numerical and experimental studies on dynamometric benches, where performance parameters are obtained at different operating conditions and testing different kinds of biofuel blends. Volpato *et al.* (2012) compare performance parameters in a four-stroke, four cylinders, compression-ignition internal combustion engine using biodiesel from olive oil (B100) and diesel oil. An improvement in torque with the use of biodiesel 8.43 % greater than diesel oil was observed, also, olive biodiesel showed lower specific and energy consumption compared to diesel oil (12.8 % and 30 % less, respectively). Other performance parameters were obtained and in-cylinder pressure curves were not collected. Ranjith *et al.* (2020) focused their work on the emissions from Neem oil in CI-ICE. Global values as torque, CO, HC and smoke emissions were then reported and the blend of 20 % Neem oil + 80 % diesel showed the best performance characteristics for the engine range tested. Hoang (2019) assessed the relationship between the spray parameters (spray penetration, cone angle and preheating temperature) of straight coconut oil compared to fossil diesel. The influence of the spray parameters on the breakup mechanism, the brake thermal efficiency, heat release rate, and emission characteristics also were investigated. As main results, straight coconut oil preheated to 105 °C achieved similar physical properties and spray parameters compared to diesel fuel, but emission parameters of carbon monoxide (CO) and unburnt hydrocarbon (HC) were higher 23.10 % and 23.36 %, respectively, brake thermal efficiency and emissions of CO_2, NO_X and smoke index were lower 3.36 %, 12.0 %, 8.86 % and 48.48 %, respectively, than those of fossil diesel fuel. Elkelawy *et al.* (2019) performed an experimental study using sunflower and soybean oils blends in CI-ICE. Both biofuels have similar physicochemical properties. After the transesterification and purification process, the biodiesel produced from sunflower and soybean oils mixture presented some physicochemical properties as kinematic viscosity, flash and fire points and cetane number, considerably different (and higher) when compared to fossil diesel. Along the experiments, global performed parameters were then obtained and in-cylinder pressure curves were then collected for different engine loads. As main results, they concluded that biodiesel produced from sunflower and soybean oils mixture could be blended with diesel up to 70 % for powering diesel engines without any modifications. Vellaiyan (2020) investigated the combustion, performance and emission of CI-ICE's using soybean biodiesel/water blends. The obtained biodiesel showed some physicochemical properties higher than the fossil diesel as observed in other works (Elkelawy *et al.* (2019); Mishra and Goswami (2018); Singh *et al.* (2020)). Global engine performance parameters were determined and in-cylinder pressure curves used for assessing the combustion process. An important observation here is that the cetane index of the soybean /water blend was dropping as water content was increasing, reaching the cetane index equal and lower than that of fossil diesel. They conclude that the biodiesel produced from soybean crude oil is a favorable alternative fuel for diesel and having the potential to promote greener emissions exception of NO_X. Working also with soybean biodiesel, Seraç *et al.* (2020) performed an evaluation of comparative combustion, performance, and emission in a CI engine. Global performance parameters were obtained and the combustion process characterized in terms of cylinder gas pressure, heat release rate, neat heat release, mass burning rate, average gas temperature and knock intensity. From the experimental results, they conclude that soybean biodiesel can be used instead of fossil diesel without any changes in the engine fuel system. Garzón (2017) performed an experimental and numerical work involving measurement and modeling of a CI-ICE operating with blends of straight soybean oil and diesel oil. Experiments were performed in a single-cylinder CI-ICE, displaced volume of 1200 cm^3, compression ratio of 17.3, nominal power of 14.7 kW/2200 rpm using a dynamometric bench for the blends 50 % and 80 % v/v of straight soybean oil in Brazilian commercial diesel oil S10. The heat release rate was determined from transient in-cylinder pressure measurement, using a zero-dimensional thermodynamic model programmed in MATLAB™ software. Concerning the simulation process of CI-ICE's, many references can be found in the technical literature. Since research works with full 3D field solutions (Henschel Jr. and Cancino, 2019; Sánchez *et al.*, 2016; Huang *et al.*, 2016) involving spray behavior and break-up, engine gas exchange and combustion process until zero-dimensional research works using commercial software, free software or in-house codes, attempting to solve simplified engine models and reporting global performance parameters (Iliev, 2014b; Reddy *et al.*, 2016; Iliev, 2014a; Seawright-de Campos *et al.*, 2015). The engine modeling is a useful tool for studying and optimizing different variables on the performance of a CI-ICE operating with alternative fuels. For that reason, in this work, the main purpose was to numerically reproduce the engine performance parameters and heat release rate using the measured/collected in-cylinder pressure data published by Garzón (2017). Numerical simulations were then performed for four fuels involving pure diesel oil and blends of straight soybean oil/diesel oil. Numerical results were validated against the experimental and calculated data from Garzón (2017). The computational tool used in this work was the AVL-BOOST™ package, module with zero/one-dimensional approach, being a simulation process more simplified than the full 3D simulation process on AVL-FIRE™ package. All the necessary input data for the numerical simulation on AVL-BOOST™ were collected from the experimental dynamometric bench at the Combustion and Thermal Systems Engineering Laboratory - LABCET/UFSC. Finally, this work confirmed the AVL-BOOST™ as a powerful computational tool for engine modeling.

2. THEORETICAL BACKGROUND

2.1 Compression-ignition internal combustion engines

The compression-ignition internal combustion engine, also known as diesel cycle engine, is part of the group of alternative internal combustion engines. They are used in vehicles that require high torque, normally applied in road transport vehicles, such as trucks and buses. The main characteristics of a CI-ICE is that combustion process starts after the fast injection of fuel into the cylinder, before the piston reaches top dead center (TDC) at the end of the compression stroke. The fuel self-ignition occurs because the favorable thermodynamics and fluid-dynamics conditions of fuel / air mixture in combination with the physical and chemical properties of the fuel used (Heywood, 1988; Brunetti, 2012). The fuel injection is an important process to reach the mixture autoignition. Along this process, the fuel undergoes several physical and chemical changes all of them correctly tuned for each engine operating condition. Consequently, the fuel physical properties are very significant, specifically the viscosity and the cetane index. (Heywood, 1988; Merker *et al.*, 2012). The fuel ignition quality is a determining factor for compression ignition engines, and it is indicated by the cetane number (CN). A compression ignition engine always operates with the maximum capacity of air volume, thus, what is controlled is the amount of fuel mass injected. The bigger the fuel mass injected, the bigger the engine torque. However, the fuel mass is restricted to poor fuel mixture for maximum combustion efficiency, avoiding unburned fuel (Martins, 2013). Additional information about CI-ICE operation and constructive aspects can be found in the technical literature Heywood (1988); Merker *et al.* (2012); Brunetti (2012).

2.2 Fuel injection, Cavitation phenomena and Combustion process in CI-ICE

The diesel fuel injection have been analyzed by several research groups around the word by using both, experimental and numerical approaches (Henschel Jr. and Cancino, 2019; Adão and Cancino, 2019; Merker *et al.*, 2012). Figure 1(a) shows the typical spray development in a SAC - type injection hole. The first break-up, named primary break-up, results in large droplets that form the dense spray near the nozzle. At high pressure injection, cavitation and turbulence could be the main break-up mechanisms (Baumgarten, 2006). As spray develops, the droplets with low kinetic energy are pushed to outer spray region forming a conical design. The further away from the initial cone, the more steam and less liquid are observed. In the dilute spray further downstream, the main factors of influence on further spray disintegration and evaporation are the boundary conditions imposed by chamber as gas temperature and density as well as gas flow (Baumgarten, 2006). Cavitation develops inside the nozzle holes because of the decrease of static pressure as a consequence of the strong acceleration of liquid, different from boiling, where there is an increase in temperature. Figure 1(b) shows the effect of the cavitation / erosion process in the spray symmetry.

In CI-ICE the combustion processes occurs along four typical stages and each one can be visualized in the heat release rate curve obtained from the in-cylinder pressure curve, Figure 1(c). The four stages are used in order to characterize the combustion process in the engine. The first one is related to the "diesel ignition delay time", indicating the time elapsed from injection to the firsts autoignition focus observed in the fuel. The second stage is related to the "premixed combustion phase", in which period the flame will generate thermochemical conditions for the next stage (the third one) called "non-premixed combustion phase" in which period usually occurs the main fuel conversion. At the end of the non-premixed combustion phase starts the four and last combustion phase called "late combustion phase" where at least all the remained residual fuel will burn. Note that all the four stages occurs in a short time, usually less than 70°crank angle, depending of the engine design and involving the piston passage by the TDC Heywood (1988). All the combustion stages are strongly influenced by the injection process as discussed briefly in the last section, and by the same way, the injection process is influenced by the physicochemical fuel properties than would have deep influence in the spray behavior. For biodiesel and vegetable oils, usually the viscosity and density are higher when compared to fossil oil, having big impact in the spray penetration (S), cone angle (θ) and primary and secondary break-up regions (Figure 1(a)). Affecting the spray behavior, the diesel ignition delay time (Figure 1(c)) will be affected. Additionally, the higher the viscosity the higher the injection pressure in the common rail, leading big chance to occur cavitation phenomena (Figure 1(b)). Note that, all the phenomenology involved in the spray behavior and combustion stages are influenced mainly by the fuel physicochemical properties and when an CI-ICE which was designed for fossil diesel oil is fueled with biofels all these aspects must be left into account in order to set-up the engine into the range of satisfactory operation conditions in therms of performance parameters and emissions. In order to overcome that issue, researchers try to tune the physicochemical properties of the proposed biofuel to the ones of fossil diesel oil. In this way several techniques as biofuel preheating in order to reduce the viscosity close to the one of fossil diesel oil (Garzón, 2017) and / or additives to tune the cetane index (Mishra and Goswami, 2018; Ma and Hanna, 1999; Chozhavendhan *et al.*, 2020) are used.

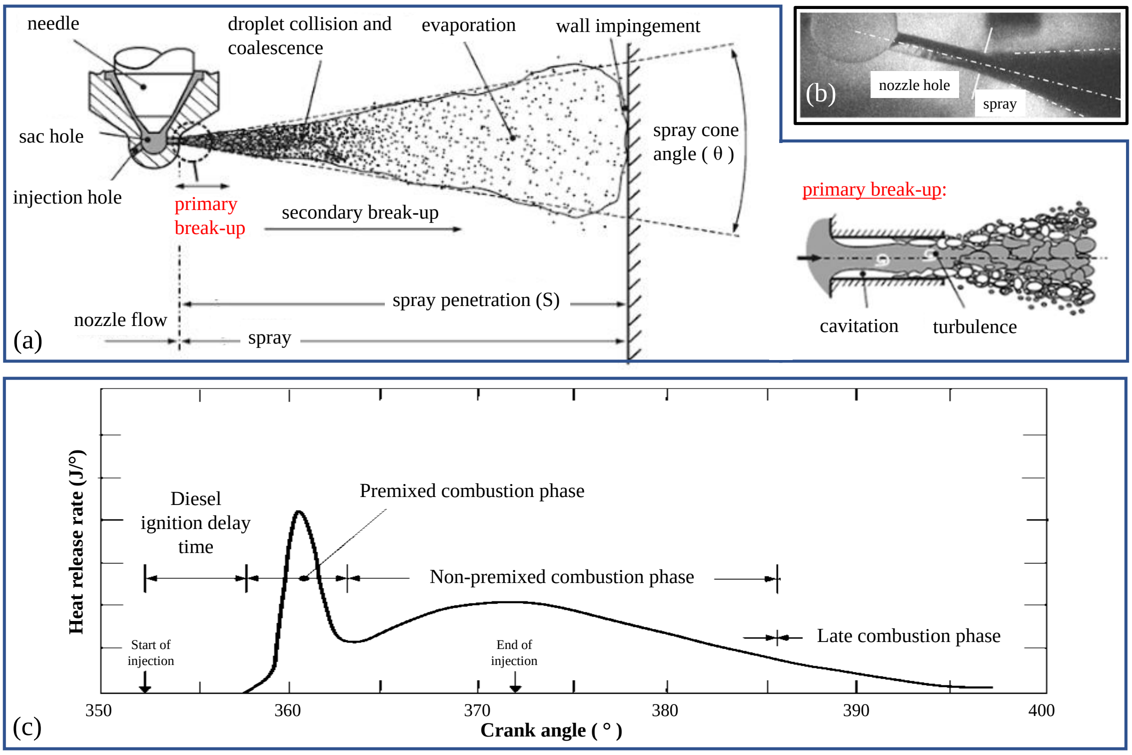

Figure 1. (a) Typical full-cone spray development in a SAC - type injection hole. (b) Effect of the cavitation / erosion process in the spray symmetry. Adapted from Baumgarten (2006). (c) Combustion phases in a CI-ICE, Adapted from Heywood (1988)

3. METHODOLOGY

In this work, the AVL-BOOST™ computational tool was used to numerically simulate the engine performance of a single-cylinder CI-ICE (Yanmar, model YT22E). The in-cylinder pressure curves from Garzón (2017) were then used as input in the model as well as the necessary geometric data for model set-up was obtained directly from the dynamometric test bench at the Combustion and Thermal Systems Engineering Laboratory - LABCET/CTC/UFSC. Numerical simulations were then performed at the Internal Combusiton Engines Laboratory - LABMCI/CTJ/UFSC and results are then compared to the experimental results from Garzón (2017).

3.1 In-cylinder pressure curves and experimental set-up from previous works

Figures 2 (a) and (b) show the experimental pressure curves recorded from all fuel blends experimentally tested by Garzón (2017) and numerically simulated in this work. Note that, four blends (two pre-heating blend temperature and different volumetric composition) tested at three engine speed, as shown in Table 1, result in twelve experimental conditions to be numerically simulated and compared to test bench results. Table 1 also presents some fuel properties as Low Heating Value (LHF) and Air to Fuel ratio at stoichiometric condition (A/F)s of the blend tested. It is possible to observe from experimental results, Figure 2(b) that as the fuel blend changes, the peak pressure and peak pressure crank angle changes.

Table 1. Fuel properties of all the blends simulated in this work - Experimental data from Garzón (2017)

Fuel blend	100D (25°C)	50S/50D (25°C)	50S/50D (85°C)	80S/20D (85°C)
Diesel oil (vol.%)	100	50	50	20
Soybean oil (vol.%)	0	50	50	80
LHV (kJ/kg)	42435	39566	39566	37968
(A/F)s	14.92	14.23	14.23	12.82

Figures 3 (a) and (b) show the measured valve lift (Garzón, 2017) and the valve flow coefficients determined numerically (Sánchez *et al.*, 2016; Sánchez, 2017), respectively. The intake valve open (IVO), intake valve close (IVC), exhaust valve open (EVO) and exhaust valve close (EVC) values as well as the injection point where kept constant for all experiments. Those data was then used as set-up parameters for AVL-BOOST™ in this work.

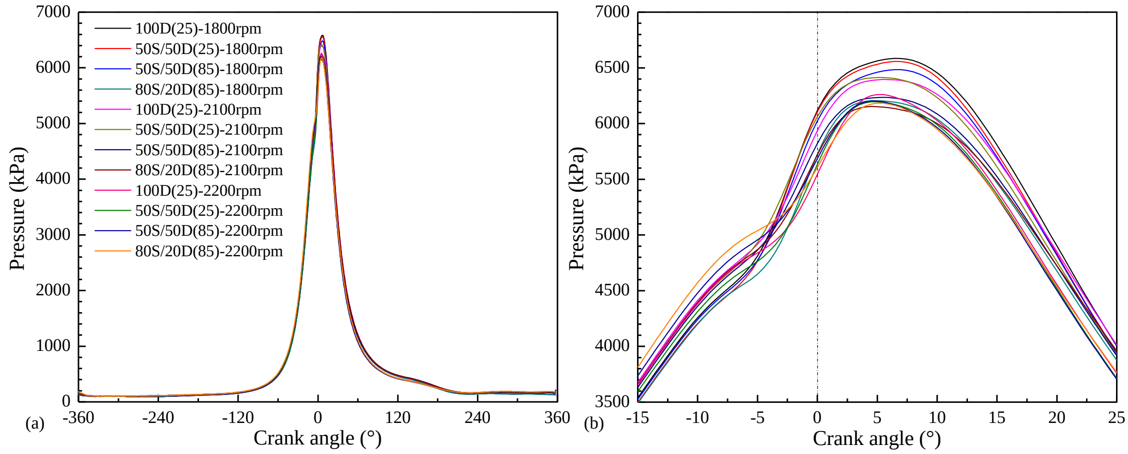

Figure 2. In-cylinder pressure curves for all fuel blends numerically tested in this work. (a) Along the four engine strokes, (b) Close the TDC region (Experimental data from Garzón (2017)

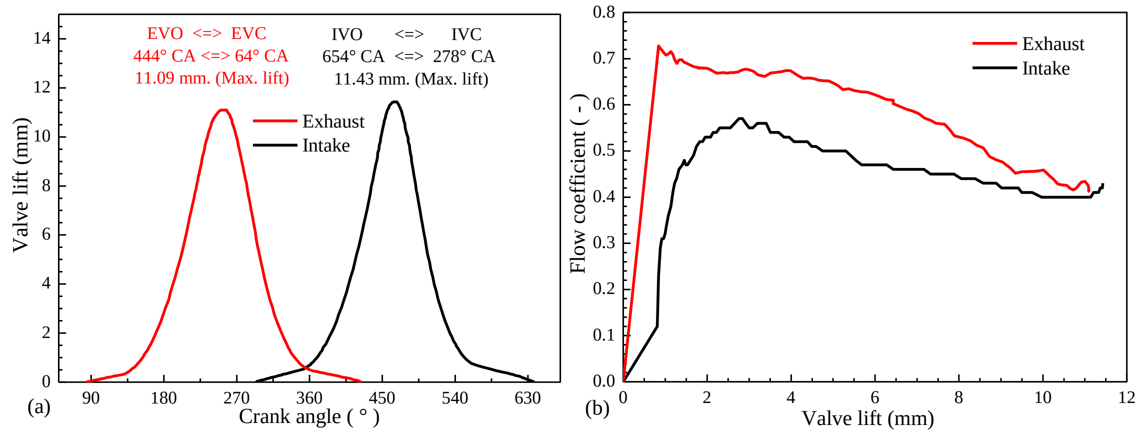

Figure 3. (a) Valve lift includding the IVO, IVC, EVO and EVC. (data from Garzón (2017)) (b) Valve flow coefficients @ 1800 rpm. (data from Sánchez *et al.* (2016); Sánchez (2017)

3.2 Numerical approach - The AVL-BOOST engine model used in this work

The numerical simulation requires detailed definition of type of fuel and its physico-chemical properties, engine geometries, engine rotation speeds, fuel mass injected, data related to friction of moving components, crankshaft geometry, combustion model, heat transfer model, gas exchange process through the intake and exhaust valves, among others. The engine model was originally developed by Wisnieski (2017) and then adapted / up-graded for the purposes of this research work. Complete description of the original model in terms of components and geometry dimensions can be found at Wisnieski (2017) and Garzón (2017). Figures 4 (a) and (b) shows the experimental dynamometric test bench and the AVL-BOOST™ engine model diagram. Unlike Wisnieski (2017), who simulated the engine for 100D(25°C), in the present work simulations were up-graded and extended to 100D(25°C), 50S/50D(25°C), 50S/50D(85°C), 80S/20D(85°C) fuel blends. In this work, set-up parameters as engine heat transfer models, valve lifts and valve flow coefficients were then up-graded. Other important modification in the AVL-BOOST™ engine model of this work is related to the use of the "BMEP control" option for the fuel mass injected in each simulation.

In the engine AVL-BOOST™ model, the option *non-default fuel combustion properties*, which allows the user to modify default fuel combustion properties (LHV and $(A/F)_S$) was activated and values from Table 1 were then used. For the valve port set-up is necessary to define the areas and temperatures for air / solid heat transfer (intake and exhaust valves), along the gas exchange process. Valve areas for heat tranfser were calculated using the engine CAD in SolidWorks™ , and temperatures were numerically estimated by Sánchez (2017). Geometrical dimensions such as the internal diameter of the valve seat, arm clearance and port scale factor were also introduced / up-dated. The valve lift data were then up-dated giving the exact angles where the valves open and close, and the maximum lift that can reach along intake and exhaust strokes. Valve flow coefficients were specified for the intake and exhaust valves as a function of valve lift. These coefficients were calculated with actual mass flow from multidimensional simulation (3D) (Sánchez, 2017) and the

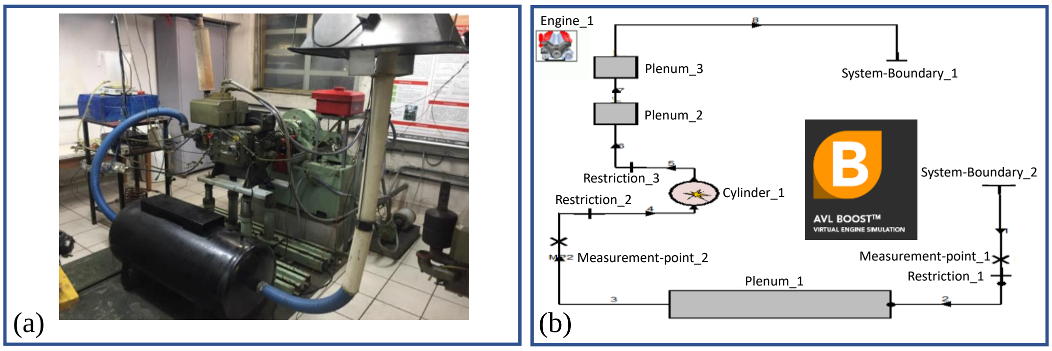

Figure 4. (a) Dynamometric test bench at Combustion and Thermal Systems Engineering Laboratory - LABCET/CTC/UFSC. (b) AVL-BOOST™ engine model of this work - Adapted from Wisnieski (2017)

isotropic mass flow from zero-dimensional modeling (0D) (Garzón, 2017). The cam command is an OHV (Over Head Valve) system with a rocker arm ratio of 1.532:1 and gear ratio of 1:2.

4. RESULTS AND DISCUSSIONS

In this section, the numerical results of heat release rate, performance parameters as power, torque, mean effective pressure, specific fuel consumption, fuel mass injected, volumetric and thermal efficiencies were compared to the values obtained by Garzón (2017).

4.1 Heat release rate

The heat release rate (HRR) for all the blends described in Table 1 for the three engine speeds numerically obtained in this work using AVL-BOOST™ are plotted in Figures 5, 6, 7 and 8 and compared to the heat release rate obtained using the in-house-code from Garzón (2017). The heat release rate means the heat generation rate along the combustion process, including the premixed and non-premixed phases (see Fig. 1(c)). Both methodologies use the in-cylinder pressure data $(p, dp/dt)$ to obtain the heat release rate from the energy conservation equation applied to the system of the engine cylinder while the valves remain closed. For this system, the only mass flow across the boundaries is the fuel (crevice flows are neglected) and its energy balance equation is shown in Equation 1.

$$\frac{dQ}{dt} - p\frac{dV}{dt} + \frac{dm_f}{dt}h_f = \frac{dU}{dt} \tag{1}$$

where dQ/dt represents the heat transfer rate through of system boundaries, pdV/dt is the work rate done by the system due to system boundary displacement, dm_f/dt is the mass fuel flow, h_f is the fuel enthalpy and, dU/dt is the rate of internal energy of the gas inside the system boundary. According to Heywood (1988), there are two methods normally used to obtain the heat release rate from in-cylinder pressure data using Equation 1. The first one, used in this work, considers U as the sensible internal energy and h_f as the sensible fuel enthalpy, and then the term dQ/dt becomes the difference between the heat released by the combustion and the heat transfer rate to the walls, obtaining the apparent heat release rate. The second one, used by Garzón (2017), assumes a complete burning instantaneously as the fuel enters the combustion chamber, and that the thermodynamics states of the working fluid are affected by the chemical energy release, then, the term dm_f/dt is considered as the apparent fuel mass burning rate and the heat release rate is determined multiplying by the lower heating value of the fuel. In this second approach, the gas properties in the cylinder are calculated as a function of p, T and ϕ (equivalence ratio). Note that the in-house-code from Garzón (2017) is numerically consistent with the AVL-BOOST™ approach. There are small differences in the HRR calculated by each methodology, the explanation could be related to the difference in the gas properties and the heat loss rate estimation to the walls. In both works, this term is estimated as the sum of convective and radiative heat transfer process each one with the respective transfer coefficients, using the Hohenberg model to determine the convective heat transfer coefficient. The in-house-code from Garzón (2017) does not take into account the pressure losses throughout pipe lines, plenum and other test bench components, conversely, the AVL-BOOST™ engine model used in this work takes into account the pressure losses. Pressure losses will affect the velocity and Reynolds number used for convective heat transfer coefficient determination, in this form, the heat loss term estimation must be different in both methods.

All the HRR curves are shown for crank angle range from 345°CA to 444°CA, corresponding the period when the intake and exhaust valves are closed. In an overall way, all the HRR curves AVL-BOOST™ fit in good agreement with the prediction of Garzón (2017), including the negative HRR values related to fuel phase change in all the cases. In the

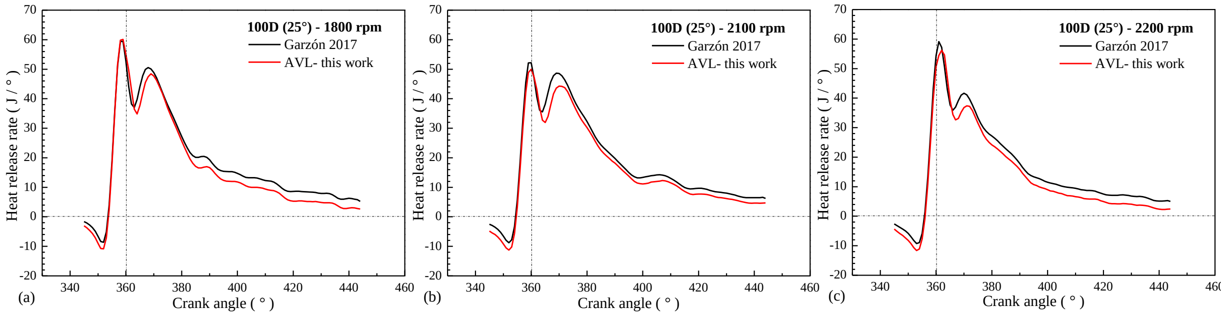

Figure 5. Heat release rate for diesel oil (100% - 25°C) for (a) 1800 rpm, (b) 2100 rpm and (c) 2200 rpm

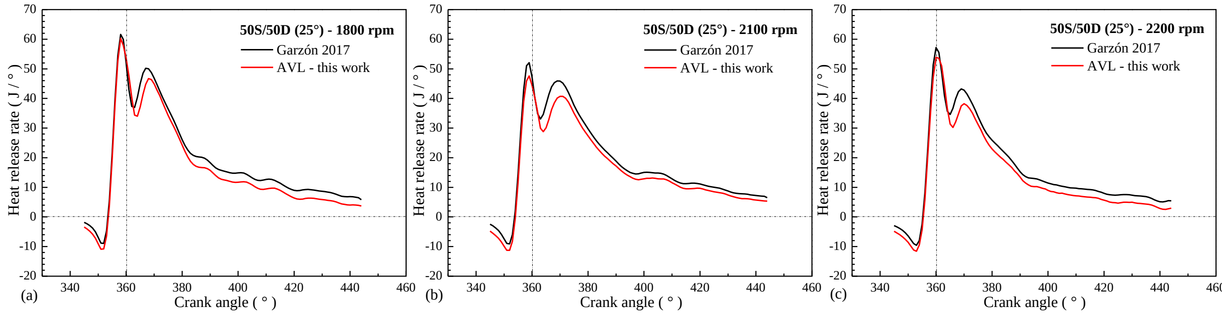

Figure 6. Heat release rate for the blend soybean oil / fossil diesel oil (50%/50% - 25°C) (a) 1800 rpm, (b) 2100 rpm and (c) 2200 rpm

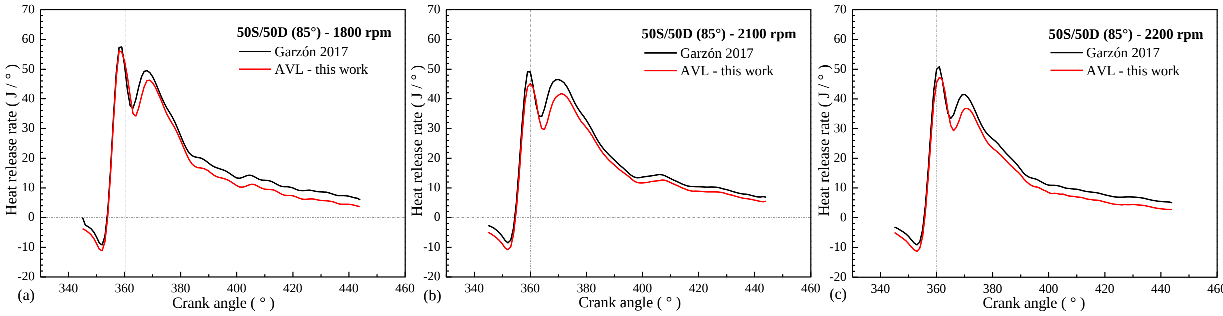

Figure 7. Heat release rate for the blend soybean oil / fossil diesel oil (50%/50% - 85°C) (a) 1800 rpm, (b) 2100 rpm and (c) 2200 rpm

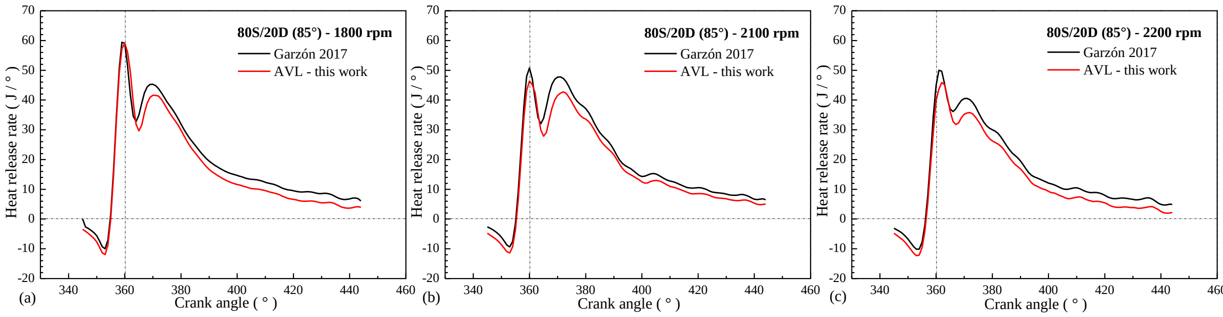

Figure 8. Heat release rate for the blend soybean oil / fossil diesel oil (80%/20% - 85°C) (a) 1800 rpm, (b) 2100 rpm and (c) 2200 rpm

same way, in all the HRR numerical predictions can be identified the four combustion stages in CI-ICE's delineated by Heywood (1988) and shown in Figure 1(c).

4.2 Performance parameters

Performance parameters results are presented as torque, power, mean effective pressure (MEP), specific fuel consumption (sfc), volumetric efficiency, thermal efficiency and fuel mass injected. Table 2 shows all the performance parameters and compares them to the experimental data obtained by Garzón (2017), including the error estimated for the results of this work. One observes negative errors (higher values) for specific fuel consumption and fuel mass injected, indicating of this form that, from the AVL-BOOST™ predictions, the numerically investigated CI-ICE is less efficient when compared to the experimental data. On this point, it is necessary to advise the issue related to the heat transfer rate estimation to the walls. In general, one notes coherence between the simulated results and the experimental data, showing in the predicted values a behavior according to the properties of the fuels and the test condition on the dynamometric bench (load, engine speed). The absolute error estimated for all performance parameters is in the interval of 1.8 % to 17.1 %.

Table 2. AVL-BOOST™ numerical predictions (this work) vrs Garzón (2017) experimental performance parameters for all the blends at three engine speeds

Diesel oil -100% (25°C)									
Parameter	**1800 rpm**			**2100 rpm**			**2200 rpm**		
	Exp. Garzón (2017)	Num. AVL This work	Error (%)	Exp. Garzón (2017)	Num. AVL This work	Error (%)	Exp . Garzón (2017)	Num. AVL This work	Error (%)
Torque (N.m)	66.19	60.00	9.3	61.57	55.99	9.1	50.40	47.30	6.1
Power (kW)	12.48	11.31	9.4	13.55	12.31	9.1	11.66	10.91	6.5
Mean Effec. Press. (kPa)	696.5	631.3	9.4	647.7	589.0	9.1	530.3	498.0	6.1
Spec. Fuel Cons. (g/kW.h)	294.6	335.5	-13.9	308.9	350.6	-13.5	271.0	317.4	-17.1
Volumetric Eff. (%)	0.868	0.833	4.0	0.859	1.000	-16.4	0.849	0.974	-14.8
Thermal Eff. (%)	0.288	0.253	12.2	0.275	0.282	-2.7	0.313	0.327	-4.4
Mass Fuel Inj. (mg/cycle)	68.09	70.27	-3.2	66.40	68.50	-3.2	47.68	52.45	-10.0
Soybean oil / Diesel oil - 50% / 50% (25°C)									
Parameter	**1800 rpm**			**2100 rpm**			**2200 rpm**		
	Exp. Garzón (2017)	Num. AVL This work	Error (%)	Exp. Garzón (2017)	Num. AVL This work	Error (%)	Exp . Garzón (2017)	Num. AVL This work	Error (%)
Torque (N.m)	65.07	59.44	8.65	60.21	55.01	8.63	50.40	46.60	7.53
Power (kW)	12.28	11.20	8.8	13.31	12.10	9.10	11.50	10.70	6.60
Mean Effec. Press. (kPa)	684.5	625.3	8.6	633.4	578.7	8.60	522.8	490,3	6.20
Spec. Fuel Cons. (g/kW.h)	311.5	351.9	-13.0	327.5	369.8	-12.9	291.4	322.3	-10.6
Volumetric Eff. (%)	0.782	0.852	-8.9	0.885	1.000	-13.0	0.855	0.979	-14.5
Thermal Eff. (%)	0.292	0.262	10.20	0.278	0.287	-3.20	0.312	0.345	-10.6
Mass Fuel Inj. (mg/cycle)	70.76	72.02	-1.8	68.82	71.02	-3.20	50.55	52.43	-3.70
Soybean oil / Diesel oil - 50% / 50% (85°C)									
Parameter	**1800 rpm**			**2100 rpm**			**2200 rpm**		
	Exp. Garzón (2017)	Num. AVL This work	Error (%)	Exp. Garzón (2017)	Num. AVL This work	Error (%)	Exp . Garzón (2017)	Num. AVL This work	Error (%)
Torque (N.m)	64.30	58.92	8.37	59.35	54.57	8.06	46.91	44.39	5.36
Power (kW)	12.10	11.10	8.20	13.10	12.0	8.40	10.85	10.23	5.70
Mean Effec. Press. (kPa)	676.8	619.8	8.40	624.5	574.0	8.10	493.3	467.1	5.30
Spec. Fuel Cons. (g/kW.h)	307.1	346.1	-12.7	320.1	359.4	-12.3	287.9	294.3	-2.2
Volumetric Eff. (%)	0.892	0.851	4.60	0.868	1.000	-15.2	0.871	0.977	-12.1
Thermal Eff. (%)	0.295	0.263	10.9	0.283	0.295	-4.40	0.315	0.378	-20.0
Mass Fuel Inj. (mg/cycle)	68.97	71.18	-3.2	66.33	68.45	-3.20	47.12	45.61	3.20
Soybean oil / Diesel oil - 80% / 20% (85°C)									
Parameter	**1800 rpm**			**2100 rpm**			**2200 rpm**		
	Exp. Garzón (2017)	Num. AVL This work	Error (%)	Exp. Garzón (2017)	Num. AVL This work	Error (%)	Exp . Garzón (2017)	Num. AVL This work	Error (%)
Torque (N.m)	61.35	57.30	6.60	58.65	55.80	4.85	47.12	45.30	3.86
Power (kW)	11.60	10.80	6.90	12.94	12.27	5.20	10.96	10.44	4.80
Mean Effec. Press. (kPa)	645.5	602.8	6.60	616.7	587.0	4.80	498.7	476.6	4.40
Spec. Fuel Cons. (g/kW.h)	320.3	365.0	-14.0	329.5	357.3	-8.4	298.7	322.5	-8.00
Volumetric Eff. (%)	0.921	0.851	7.60	0.894	1.000	-11.9	0.909	0.975	-7.30
Thermal Eff. (%)	0.286	0.260	9.20	0.287	0.310	-7.9	0.316	0.359	-13.7
Mass Fuel Inj. (mg/cycle)	70.74	73.00	-3.20	67.43	69.59	-3.2	49.41	50.99	-3.20

Figure 9 shows the numerical results obtained in this work compared to the experimental data of power and torque as a function of engine speed. From Figure 9 is possible to conclude that the AVL-BOOST™ predictions are lower when compared to the experimental data. The AVL-BOOST™ predictions involve a full zero/one-dimensional numerical approach, including other effects as pressure losses through pile lines and other devices will have an impact in the numerical results. Also, it can be observed that numerical predictions of this work have the same trend of the experimental data, observing the best power for engine speed of 2100 rpm and best torque for engine speed of 1800 rpm. For the AVL-BOOST™ predictions, just for the blend 80S/20D 85°C at the engine speed of 2100 rpm, an improvement in relation to the other blends was observed. That behavior was not observed in the experimental data from Garzón (2017).

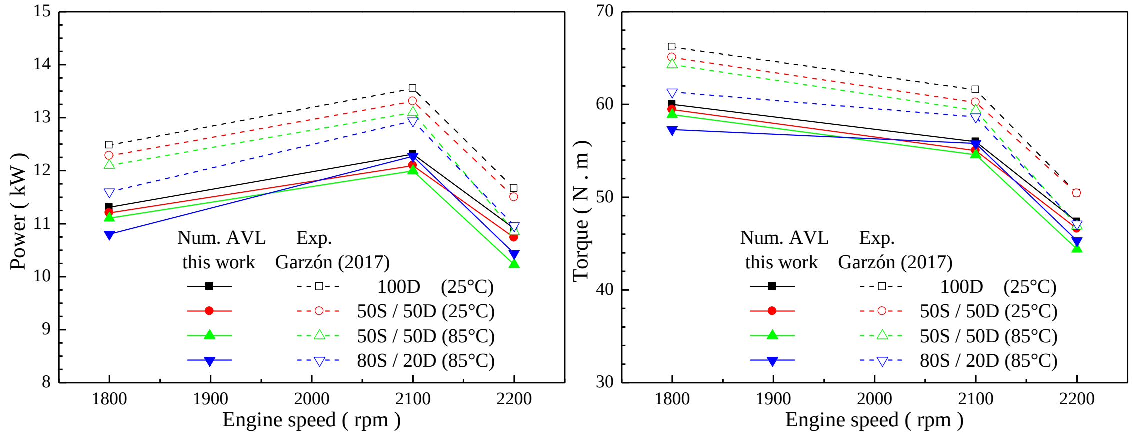

Figure 9. AVL-BOOST™ numerical predictions of performance parameters. (a) Power, (b) Torque

5. CONCLUSSION

All the numerical results in this work can confirm the AVL-BOOST™ as a powerful computational tool for engine modeling. In this case, adopting a zero/one-dimensional approach, the results show that despite the simplicity of the engine model implementation in the AVL-BOOST™ module (apparent simplicity when compared to full 3D simulation process as AVL-FIRE™ package, for example) consistent results plausible of comparison to experimental data with acceptable error percentage for engines development and analysis process. This work shows a numerical approach and validation against experimental data of performance parameters of a CI-ICE fueled with fossil diesel oil and straight soybean oil blends available in the literature. Four fuel blends named 100D(25°C), 50S50D(25°C), 50S50D(85°C) and 80S20D(85°C) regarding the soybean oil and diesel volumetric percentage and fuel temperature were simulated for 1800 rpm, 2100 rpm and 2200 rpm at the conditions of BMEP tested on dynamometric test bench. Performance parameters as torque, power, mean effective pressure (MEP), specific fuel consumption (sfc), volumetric efficiency, thermal efficiency and fuel mass injected, as well as the heat release rate, are reported and compared to available data reported in the literature. The developed AVL-BOOST™ engine model involve all the main engine components in terms of intake and exhaust collectors, including pipe lines, plenums, cylinder and valves geometrical parameters and operation. The numerical results of this work show good agreement with experimental data and differences are reported in terms of error percentages.

6. ACKNOWLEDGEMENTS

The authors would like to acknowledge to the AVL-AST University Partnership Program (UPP) for the use and support of AVL-AST software. The support of UFSC Joinville TI team is also highly appreciated.

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