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Abstract

Using diesel-biodiesel blends in Diesel engine has been highlighted to offer a good opportunity in reducing the exhaust emissions for particular engine operating conditions. The blended diesel-biodiesel with up to 20% biodiesel in petroleum diesel fuel (B20) is in production and available for use in USA being considered as viable path to be followed in the effort to reduce the effect of the greenhouse gas emissions issued by the operation of heat engines. Although previous studies investigating the effect of B20 on engines emissions led to some contradictory results, the life cycle analysis performed on the well-to-wheel base shown that 35% reductions of CO_2 emission in respect to fossil fuels operation are possible. The association of this alternative fuel use with some new combustion concepts as homogeneous charge compression ignition (HCCI) premixed charge compression ignition (PCCI) or partially premixed combustion modes (PPC) may amplify these potential reductions.

The present study continued the investigation on B20 effects by performing a set of comparative experimental tests on a conventional direct injection tractor diesel engine running alternatively with B20 and petroleum diesel fuel at 2400 rpm speed and 60% load. The possibilities to enhance the effects of B20 fuel by PPC combustion operation mode were explored by numerical simulations using the AVL BOOST v2013.2 code. It was basically found that further improvements in decreasing emissions of Diesel engines operating with B20 can be obtained without significant changes in Diesel engines structure and adjustments.

Introduction

PPC (Partially Premixed Combustion) operating mode for CI (compression ignition) engines has been promoted as an intermediary stage between HCCI (homogenous charge compression ignition) mode, in which the entire fuel quantity is premixed with the air (almost like for classic SI engines) and Diesel classic injection operating method, for which combustion turns to its diffusive characteristic (Tanov et al. 2014). Every mode of air fuel mixture formation proves its benefits or disadvantages, as these different modes are illustrated in Figure 1, separated by different injection timings. Operating in HCCI mode means that SOI (start of the injection) is very early positioned on cycle and combined with the use of a fuel with higher auto-ignition resistance conducts to an almost complete homogeneity of the vaporized fuel in the combustion air. This could adjust the rough engine operation and its noise together with the NOx emissions level, but it strongly encourages the knock (for low octane number fuels) CO and smoke to occur (Kalghatgi et al. 2011). On the contrary, standard diesel injection timings, followed by rapid and then diffusive combustion increases the rate of the cycle heat release together with the gases temperature, leading to the rise of the NOx emissions, if not controlled by EGR means (Fridriksson 2011; Maurya and Agarwal 2013).

Related to another concept, the gasoline compression ignition concept (GCIC), it has been proven that gasolines with 70 to 75 RON (research octane number) are suitable to be used in CI engines due to their weak resistance, especially when referring to operational conditions consisting in low loads and high speeds (Kim et al. 2014; Manente at al. 2010).

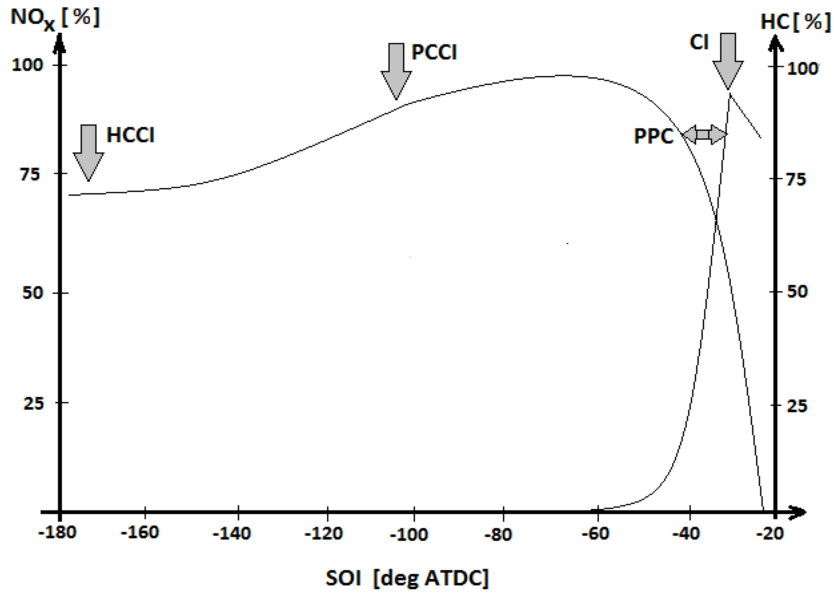

Figure1: Different modes in CI engines operation with regard to emissions levels (Tanov et al.2014)

PPC operating mode could be a promising way to get minimized the emissions (soot, CO and NOx), keeping the engine in-between its efficiency limits at high loads and speeds either. In case of Diesel (DI) engine, the benefit comparing to HCCI combustion is that the fuel injection timing could be use itself as a main control parameter for the combustion period. Negative valve overlap (NVO) is an operating mode that enables LTC (low temperature combustion) in automotive engines. In addition to retain a large fraction of residuals, NVO operation also enables partial fuel injection during the recompression period as a means of enhancing and controlling main combustion. Thermal effects of NVO fueling on main combustion are well understood, but chemical effects of the products of NVO reactions remain uncertain (Steeper and Davisson 2014). According to other authors, the thermal control of the heat transfer could also be obtained using NVO (negative valve overlap) with symmetric OVC (outlet valve closing) and IVO (inlet valve opening) cycle positions to TDC (top dead center) (Cao et al. 2007).

Using biodiesel and biodiesel-diesel blends as alternative fuels in diesel engines, the above described aspects could be improved regarding the influences of this particular fuel on air-fuel mixture formation and combustion, as it has been highlighted in previous studies (Gharbanpour and Rasekhi 2013). There are evident differences between chemical and thermo-physical properties of biodiesel compared to those of fossil fuel. For example, the higher viscosity of biodiesel can affect the spray characteristics, the air-fuel mixture formation and its combustion. However, the ignition delay when using biodiesel is shortened due to its higher cetane number. The lower heating value and the higher oxygen content for biodiesel conduct to a shorter ignition delay and a shorter combustion duration, reducing the heat release rate and the maximum rate of the pressure rise and finally reducing the engine rated power comparing to the classic Diesel fuel engine operation. Regarding the exhaust emissions, simulation and experimental results showed that the use of biodiesel (B20) in compression engines generally led to lower CO_2, CO and HC exhaust gas emissions (Voicu et al. 2013).

The present work describes an appropriate way to decrease Diesel main emissions' levels by using both PPC operating mode, by modifying fuel injection timing and the use of B20 diesel-biodiesel blend fuel when testing a direct injected and normal aspirated tractor diesel engine.

Experimental Infrastructure, Engine Operating Modes and Simulation Procedures

A fully operational test bed equipped with required measurement devices has been adapted on testing the performances and the emissions of the naturally aspirated direct injection tractor diesel engine, with 4 cylinders in line, having the total displacement of 3759 cm^3, nominal power of 50 kW at 2400 rpm, maximum torque of 228 Nm at 1400 rpm, bore/stroke ratio of 102 mm/115 mm and the compression ratio of 17.5. The schematic of the test bed is showed in Figure 2 (Chiriac et al. 2015).

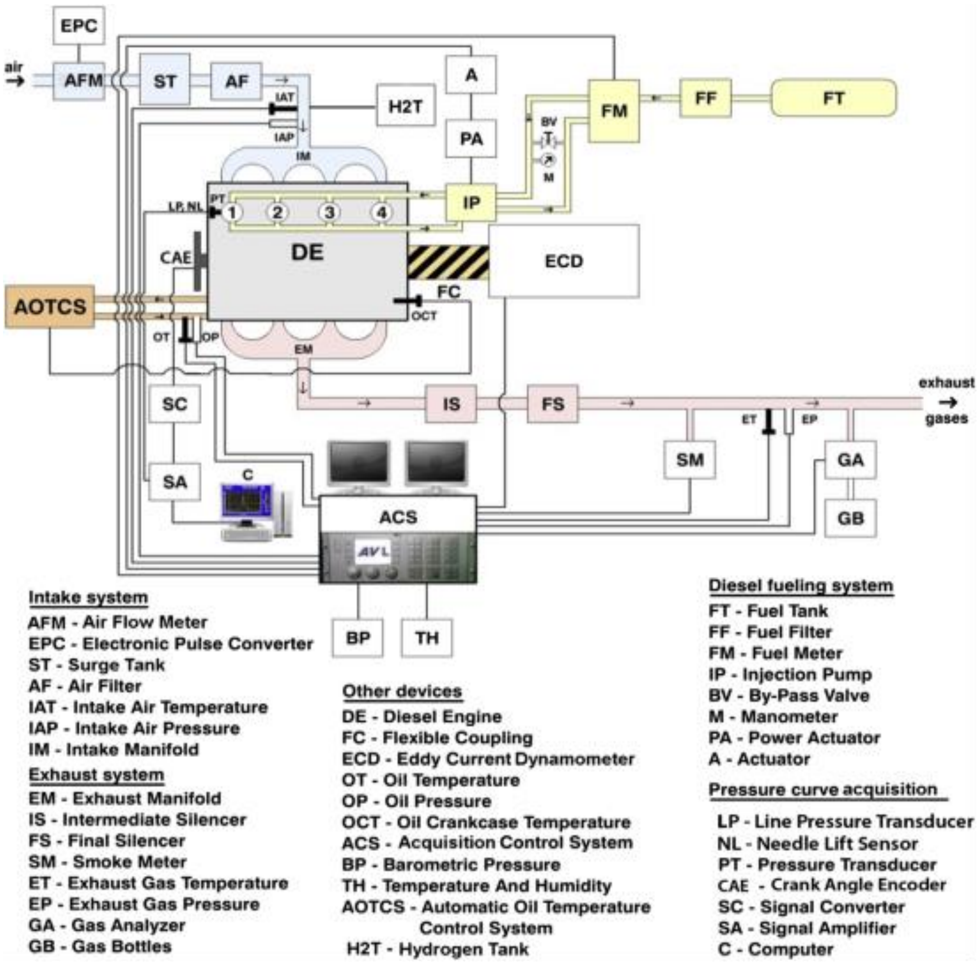

Figure 2: The schematic of the test bed

The multiple fuels operating test bed has been adapted under the purpose of the present work, allowing the engine fueling alternatively with the both tested fuels, Diesel and biodiesel B20. The standard value for diesel injection timing was -7 °CA start of injection (SOI) relative to TDC. The performances and the emissions of the operated engine were tested at 2400 rpm speed, 60% load and different injection timings and decreasing SOI points (down to 20 deg) (Voicu et al. 2013).

In order to develop the analysis regarding the engine operation and performances under PPC mode, the engine calibration parameters and cylinder processes simulation were described by using AVL BOOST code v2013.2 (AVL BOOST Theory and AVL BOOST UsersGuide). By means of the Boost interface, all the engine components, such as the cylinders, the intake and exhaust manifolds, as well as some auxiliary equipment, the system boundaries, the air filter, the catalyst etc, have been modelled (Chiriac et al. 2015). All the components are linked together by pipes as it can be seen in Figure 3.

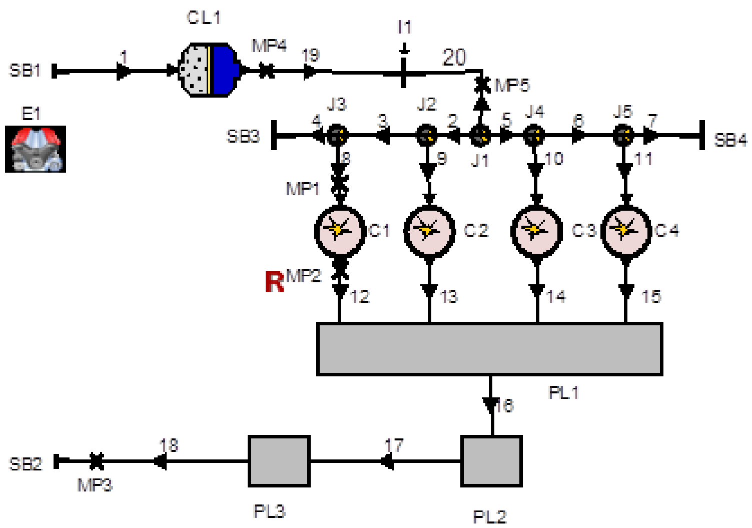

Figure 3: The schematic of the engine symbolic model (AVL BOOST Theory and AVL BOOST UsersGuide)

The values of calibration parameters were chosen considering the AVL – MCC combustion model, for which the operating data on injection and combustion characteristics would provide acceptable relative errors. These parameters values are specified in Table 1:

Parameter	Diesel	B20
Number of Injector Holes [-]	5	
Hole Diameter [mm]	0.24	
Discharge Coefficient (DisC) [-]	0.7	
Rail Pressure (RaiP) [bar]	350	
Ignition Delay Calibration Factor (IgnDel) [-]	0.72	0.66
Combustion Parameter (ComPar) [-]	1.62	1.68
Turbulence Parameter (TurPar) [-]	1	
Dissipation Parameter (DisPar) [-]	1	
Premixed Combustion Parameter (PremixPar) [-]	0.91	0.65
NOx Kinetic Multiplier (NOxKM) [-]	2.05	1.17
NOx Post Processing Multiplier (NOxPM) [-]	0.35	0.34
CO Kinetic Multiplier (COKM) [-]	0.0245	0.0241
Soot Production Constant (SPC) [-]	150	290
Soot Consumption Constant (SCC) [-]	330	60
EGR Parameter (EGRPar) [-]	1	
Evaporation Parameter (EvapPar) [-]	0.70353	

Table 1: Calibration parameter values

The simulation analysis has considered the engine operating condition previously mentioned: 60% load, 2400 rpm speed and a series of SOI timings (°CA) starting with -7 (standard) and down to -44, in order to obtain a reasonable approach to the conditions characterizing the PPCC and PPC air-fuel mixture formation modes. The obtained results should highlight a particular value of the injection timing, for which engine emissions (NO_x, CO, smoke) could register significantly decreasing compared to the standard operating condition, meanwhile preserving engine performance (P_e) and efficiency (BSFC).

Results and comments

A first group of results after setting the values of calibration parameters for the calculation model is related to the best approach between the two pressure traces, experimental and simulation for the two fuels operation, pure Diesel fuel and biodiesel B20. Figure 4 shows the comparative variation of the pressure curves plotted when engine operated in reference condition (2400 rpm speed, 60% load and injection timing corresponding to -7 °CA SOI, Start of Injection).

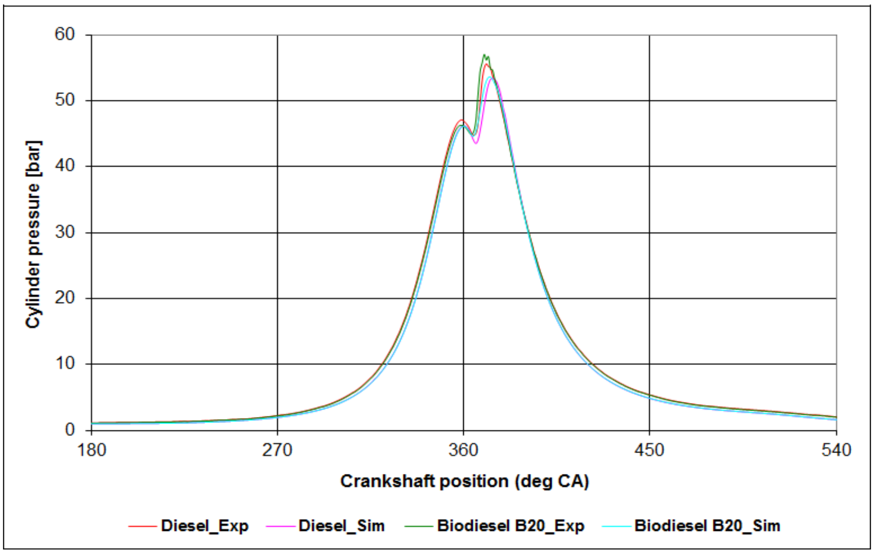

Figure 4: Comparison between experimental and simulation pressure traces for 60% load, 2400 rpm speed and -7 °CA, SOI.

In Table 2 are listed the simulation results, including the calculated relative deviations concerning engine emissions levels, performance and efficiency indicators, related to the experimental values, for Diesel fuel and biodiesel B20 at reference operating engine condition.

	NO_x [g/kWh]		CO [g/kWh]		Soot [g/kWh]		p_{max} [bar]		$(dp/d\alpha)_{max}$ [bar/deg]		P_e [kW]		BSFC [g/kWh]		Lambda [-]		Fuel Cons. [kg/h]	
	Sim	Exp	Sim	Exp	Sim	Exp	Sim	Exp	Sim	Exp	Sim	Exp	Sim	Exp	Sim	Exp	Sim	Exp
Diesel	6.45	6.40	5.75	6.20	0.236	0.23	53.3	53.5	2.03	1.9	26.6	26.83	288.5	285.8	2.29	2.36	7.7	7.66
St. dev.	**0.8%**		**-7.2%**		**2.6%**		**-0.4%**		**6.8%**		**-0.5%**		**0.9%**		**-2.9%**		**0.5%**	
B20	6.14	6.24	6.06	6.55	0.18	0.2	53.5	52	1.88	2.08	26.75	26.83	295.7	298	2.46	2.38	7.92	7.99
St. dev.	**-1.6%**		**-7.4%**		**-9.5%**		**2.9%**		**-9.6%**		**-0.3%**		**-0.5%**		**3.3%**		**-0.8%**	

Table 2: Comparison between simulation and experimental results at reference condition

As a consequence of model calibration, the model parameters have been adopted considering acceptable relative deviations of the significant performance and emissions indicators which are in the domain -9.6%; +3.3% (see Table 2). The highest values for the relative deviations were registered for peak pressure rise and for soot emission.

The simulation work has been extended to evaluate the engine emissions values (CO, Soot and NO_x,) at different injection timings, with SOI values of: -32, -35, -38, -41 and -44 °CA, under the same engine operating conditions of load and speed keeping the values adopted for model parameters which were established in the calibration stage.

Figures 5, 6 and 7 highlight the levels of CO, Soot and respectively NO_x as simulation results in comparison with the values obtained for -7 °CA SOI as the reference operating condition. Like a general remark, all the emission values for extended injection timings are significantly lower than the experimental reference values. The lowest values for CO and Soot emissions are related to the injection timing of 38 °CA, bTDC (before top dead center). The reduced emissions for both fuels in the range of advanced injection timings (-44 to - 32 °CA) could be explained by better homogenization of the air-fuel mixture preventing the occurrence of over rich zones which are responsible for soot formation and carbon incomplete oxidation and also of stoichiometric zones which are involved in the NO formation. In the same time, faster combustion with an increased premixed character associated with reduced residence time diminishes the NO to NO_2 conversion rate, falling thus the NO_x emissions in respect with the reference condition.

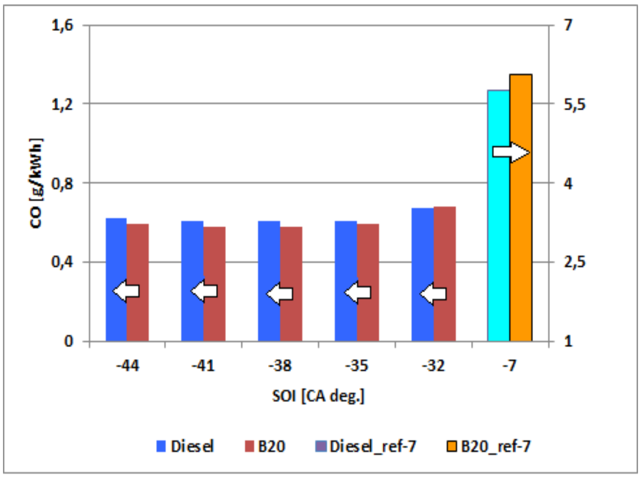

Figure 5: CO comparison values

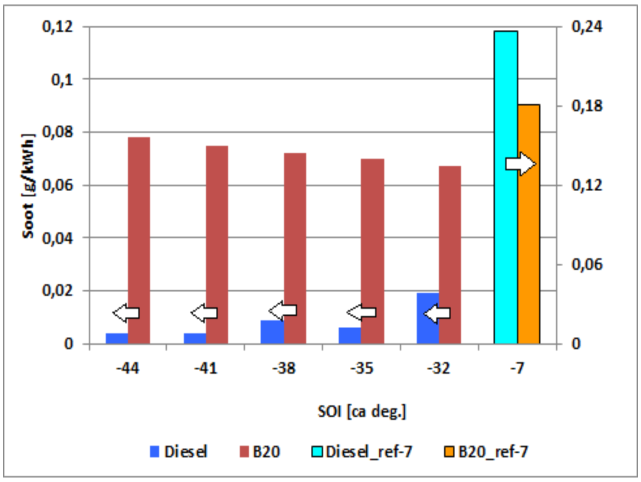

Figure 6: Soot comparison values

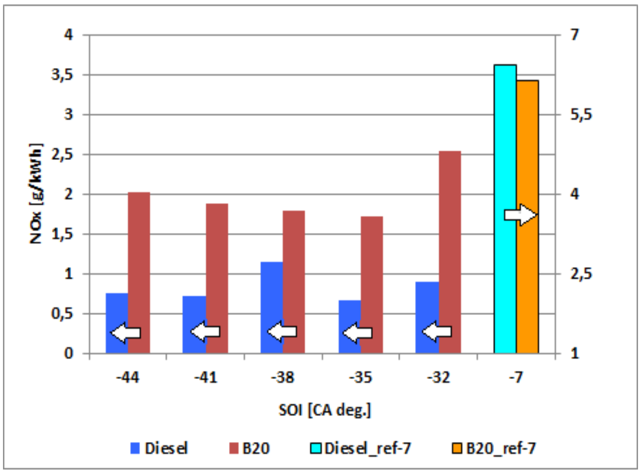

Figure 7: NO_x comparison values

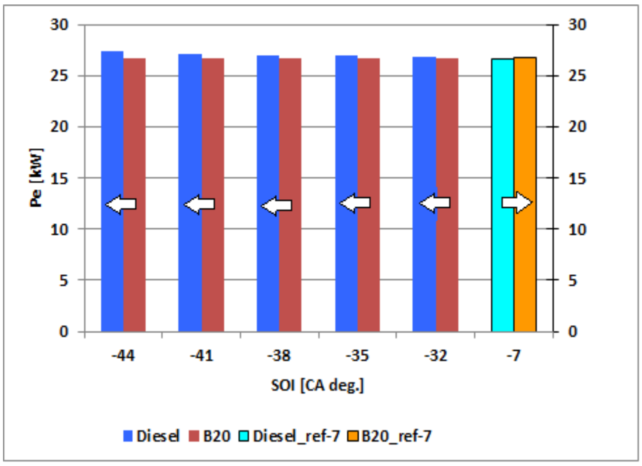

Figure 8: Output power comparison values

The comparison between the two tested fuels emphasizes for biodiesel B20 lower CO values than for pure Diesel in the whole investigated range of injection timings (fig. 5). This behavior can be attributed to the reduced carbon content of biodiesel which leads for the same air consumptions of the engine to leaner mixtures ensuring thus better carbon oxidation (the rate of CO formation/destruction mechanism is calculated by the group of reactions proposed by Onorati) (Ismail et al. 2013). There is an important decreasing by 10 times when this emission is referred to those obtained for the reference condition.

In the extended injection timings range soot is higher for biodiesel relative to pure Diesel fuel (fig. 7) although it remains approximately 3 times lower than reference. The model considers two reactions (model of Schubiger which is used for the 2-zone calculation) for the soot formation mechanism that is governed by reaction kinetic mechanisms (Schubiger et al. 2002). The soot formation reaction is attributed to the combustion rate of the diffusion combustion. The oxidation reaction depends on the actual net soot mass in the cylinder and the oxygen availability in the burned zone. The concurrence between these formation-oxidation reactions gives the magnitude of soot emission in the burned gases. The results show an increase of total combustion duration in the case of biodiesel and approximately the same duration for the premixed stage as for Diesel fuel this leading finally to longer diffusion stage for biodiesel B20. In such conditions the formation reaction is stronger than the oxidation one and soot emission becomes higher for biodiesel.

On the contrary, the NO_x emissions (calculated using Pattas and Häfner model based on the extended Zeldovich mechanism) (Gärtner et al.2004) seem to increase for the same injection timings as a result of temperature rise when a higher premixed degree of the cylinder charge occurs (fig. 6), but in respect to reference condition it is approximately 3 times lower. Higher peak fire pressure and higher peak temperature burned registered for biodiesel B20 lead to superior NO_x values.

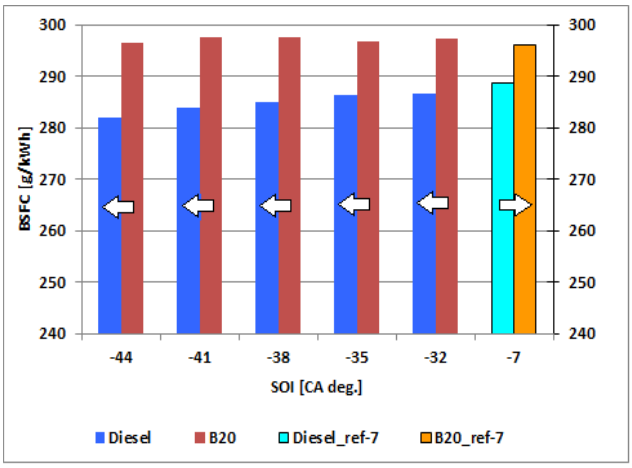

Figure 9: BSFC comparison values

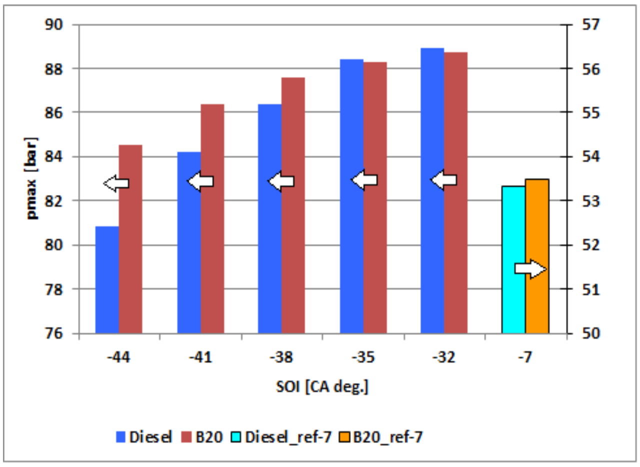

Figure10: Peak fire pressure comparison values

Figures 8, 9 and 10 reveal the values simulated for effective power (P_e), break specific fuel combustion (BSFC) and maximum pressure (p_{max}), for the same range of injection timings, keeping the same reference -7 °CA, SOI reported for the engine operation set up.

As a preliminary condition for the calculation model, the effective power was considered to remain practically unchanged (fig. 8). The resulted values concerning BSFC, for each type of fuel apart, show a small decrease by 3.5 % in the case of Diesel fuel and a small increase by 1% in the case of biodiesel. The decrease of the engine efficiency when using Biodiesel B20 is related to its LHV (lower heating value) compared to Diesel fuel (fig. 9). At constant load and speed, the peak fire pressure and peak pressure rise values are higher for earlier injection timings for both fuels because the whole combustion process is shifted towards compression with an increased ignition delay (two times longer) and stronger premixed combustion, two times longer too. Thus for both fuels, the effect of much more accumulated mass of fuel injected in the auto-ignition delay is visible as an approximately two times higher peak fire pressure. This effect diminishes towards superior injection timings where for Diesel fuel maximum pressure levels fall down under biodiesel levels by approximately 5%. The differences between the tested fuels are evident in absolute values of peak fire pressure (fig. 10).

In the extended Table 3, there are listed the simulation results for performance and emission indicators at -38 °CA SOI, considered as the optimum point in the partially premixed charge operation and the experimental results as reference for -7 °CA SOI. The calculated relative deviations for two important changes, fuel nature and injection timing show that CO emission can be substantially reduced by 91.1%, soot by 64% and NOx by 71.3%, keeping the performance parameters almost at initial values, effective power reduced only by 0.7% and efficiency reduced just by 0.1%. The important drawback of this operation mode is related to the peak fire pressure which is increased 68.4% and to the peak pressure rise increased by 104.3% in condition of similar air-fuel ratio leaned by 3.3% with reduced fuel consumption by 1%.

	NO_x [g/kWh]		CO [g/kWh]		Soot [g/kWh]		p_{max} [bar]		$(dp/d\alpha)_{max}$ [bar/deg]		P_e [kW]		BSFC [g/kWh]		Lambda [-]		Fuel Cons. [kg/h]	
	Sim	Exp	Sim	Exp	Sim	Exp	Sim	Exp	Sim	Exp	Sim	Exp	Sim	Exp	Sim	Exp	Sim	Exp
SOI [degCA]	-38	-7	-38	-7	-38	-7	-38	-7	-38	-7	-38	-7	-38	-7	-38	-7	-38	-7
Diesel	1.14	6.40	0.6	5.75	0.099	0.23	86.4	53.5	4.12	1.9	27	26.83	285	285.8	2.3	2.36	7.73	7.66
St. dev.	**-82.1%**		**-89.5%**		**-43.6%**		**61.3%**		**116.8%**		**0.6%**		**-0.2%**		**-2.5%**		**0.9%**	
B20	1.79	6.24	0.58	6.55	0.072	0.2	87.6	52	4.25	2.08	26.62	26.83	297.5	298	2.46	2.38	7.91	7.99
St. dev.	**-71.3%**		**-91.1%**		**-64%**		**68.4%**		**104.3%**		**-0.7%**		**-0.1%**		**3.3%**		**-1%**	

Table 3: Comparison between simulation and experimental values obtained at 2400 rpm speed and 60% load

Conclusions

- Simulation tests conducted on the normal aspirated tractor diesel engine at high speed, 2400 rpm and high load, 60%, have concluded that early start injection process influences the improvement of gas emissions levels comparing to diesel classic injection operation.
- This significant drop of the emissions levels is linked directly to the combustion particularities of the partial premixed mode of air-fuel formation and combustion for both types of fuels, Diesel and biodiesel B20. For this particular combustion mode, hydrocarbons and CO are presumed more likely to be consumed but meanwhile the high rate of fire pressure and temperature rise is leading to NO_x formation.
- Although biodiesel is renewable, nontoxic, and biodegradable and has low emission profiles, the main drawback of using biodiesel in diesel engines is still the high proportion of NO_x emissions at low and medium loads.
- Performance and efficiency of the tested engine maintain their reference values, the only difference consisting in the slight increase of BSFC when using B20 because of its lower LHV comparing to Diesel fuel.

- The significant augment of the peak pressure rise in correspondence with the peak fire pressure at early injection timings has to be further analyzed due to the risk of major engine vibrations and noise which could challenge reasonable limitations.
- Considering the importance of the actual results the potential of proposed solution was confirmed but an experimental and theoretical study has to be extended to other engine operating conditions.

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References

AVL BOOST Theory and AVL BOOST UsersGuide, https://www.avl.com/ro/boost
Cao, L. et al. (2007) Analysis of Controlled Auto-ignition/HCCI Combustion in a Direct Injection Gasoline Engine with Single and Split Fuel Injectors, *Combustion Science and Technology* 180:1, pp.176-205.
Chiriac, R. et al. (2015) On the Possibility to Reduce CO2 Emissions of Heat Engines Fueled Partially with Hydrogen Produced by Waste Heat Recovery, *International Journal of Hydrogen Energy*, DOI**:** 10.1016/j.ijhydene.2015.06.064
Fridriksson, H. (2011) CFD Investigation of heat transfer in a Diesel engine with Diesel and PPC Combustion Mode, *Master Thesis*, Lund University.
Gärtner, U. et al. (2004) Development and Application of a Semi-Empirical NOx Model to Various HD Diesel Engines, *Thermo-and Fluid Dynamic Processes in Diesel Engines 2* (2004), pp.285-312, *Springer Verlag*
Gharbanpour, M. and Rasekhi, R. (2013) A parametric investigation of HCCI combustion to reduce emissions and improve efficiency using a CFD model approach, *ELSEVIER Fuel* 106(2013) pp.157-165
Ismail, H.M. at al. (2013), Development of a reduced biodiesel combustion kinetics mechanism for a CFD modelling for a light-duty diesel engine, ELSEVIER, Fuel vol.106, pp.388-400.
Kalghatgi, G.T. et al. (2011) Auto-ignition quality of gasoline fuels in partially premixed combustion in diesel engines, *ELSEVIER Proceedings of the Combustion Institute* no.33(2011), pp.3015-3021.
Lim, K. et al. (2014) Load expansion of naphtha multiple premixed compression ignition (MPCI) and comparison with partially compression ignition (PPCI) and conventional diesel combustion (CDC), *ELSEVIER Fuel* 136(2014), pp.1-9
Manente, V. et al. (2010) An advanced internal combustion engine concept for low emissions and high efficiency from idle to maximum load using gasoline partially premixed combustion, *SAE Technical Paper* 2010-01-2198, 2010, doi:10.4271/2010-01-2198, 21 pp.
Maurya, R.K. and Agarwal, A.K. (2013) Investigations on the effect of measurement errors on estimated combustion and performance parameters in HCCI combustion engine, *ELSEVIER Measurement* 46(2013), pp.80-88
Schubiger, R.A. et al. (2002) Soot Emission from Diesel Combustion, *MTZ* no.5/2002, vol.63, pp. 342-353.
Steeper, R. and Davisson, M. (2014) Analysis of Gasoline Negative-Valve-Overlap Fueling via Dump Sampling, *SAE Int. J. Engines* 7(2):762-771, 2014, doi: 10.4271/2014-01-1273
Tanov, S. et al. (2014) Combustion stratification with Partially Premixed Combustion, PPC, using NVO and Split Injection in a LD- Diesel Engine, SAE Paper no. 2014-01-2677, *SAE Int. J. Engines* 7(4), *pp.1911-1919*, doi: 10.4271/2014-01-2677
Voicu, I. et al. (2013) Effects of Hydrogen induction in a Diesel engine operating with Biodiesel B20 at different injection timings, *Termotehnica Revue*, XVII (2)/2013, pp. 43-54.