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MODELLING AND SIMULATION OF IGNITION COMPRESSION ENGINES (ICE) FUELED WITH DIESEL, BIODIESEL AND NATURAL GAS FUEL MIXTURES

First Author's Name: Vilmar Graciano
Second Author's Name: José Viriato Coelho Vargas
Programa de Pós graduação em Engenharia Mecânica e de Materiais PGMEC - Universidade Federal do Paraná – UFPR
Bloco IV do Setor de Tecnologia, Centro Politécnico da UFPR, Bairro Jardim das Américas
Caixa postal 19011, CEP 81531-980, Curitiba – PR.
vgracianos@ig.com.br; jvargas@demec.ufpr.br

Abstract. *This paper introduces a general computational model for compression ignition engines (ICE) fueled by diesel, biodiesel and/or natural gas. A simplified mathematical model for the transient and steady state operation, which combines principles of classical thermodynamics and heat transfer, was developed for the working space of the engine (engine cylinder) in order to provide quick responses during system design. The model is based on geometric and operating parameters (e.g., rpm, piston and cylinder diameter, stroke, engine operating temperature, engine compression ratio, air-to-fuel ratio), and is capable of calculating the engine mean indicated pressure, indicated power and indicated torque with respect to crank speed. Friction losses are quantified based on existing empirical correlations for ICE engines with direct injection of fuel, so that engine net power and torque are also assessed. The model was adjusted and validated by direct comparison of the obtained results with previously published experimental data and characteristic curves in engine catalogs. The obtained numerical results demonstrate that the model is expected to be an important and simple tool for design, control and optimization of ICE engines driven by diesel, biodiesel and natural gas fuel mixtures, combining accuracy with low computational time.*

Keywords*: ICE engines. ICE engines simulation. Computational simulation.*

1. INTRODUCTION

Today there is a global search for bettert use of energy sources, like the cogeneration and trigeneration energy systems, including that based in diesel engines. In Abusoglu et al. (2006), Katri et al. (2010) and Temir at al. (2004), it may be felt this trend. In the other hand, in Bueno et al. (2011), Jimenez et al. (2012), Lebedevas et al. (2011) and Papagiannakis et al. (2010), there is much interest in new and alternative sources of energy in combustion machine prime movers, where diesel engines are included and in this context the biodiesel and natural gas are inside the most promising alternatives, There is too, research/development of componentes, like combustion chambers, or fuel injectors, to became the traditional diesel engines, more efficient, through the developement of new models, what can be seen in Gogoi et al. (2010), Giakomis et al. (2007) and Rakopoulos et al. (2004). Accoding to that references it is some characteristics may be cited:

- The models developed for upgrade in the project and design of traditional diesel engines, are complex, not easily handled, requiring complex aparatus and high computional time;
- There is a low number of computacional programs that shows the four strokes of diesel engines, including the combustion, in a easy form but full with low consumption of computacional time and giving satisfactory precision responses, and
- There is a low number of computacional programs for alternative fuels that spents low computacional time.

Keeping in mind the items above, this paper proposes a simplified mathematical model for diesel engines, wich considers the transient operational conditions of the engine, using diesel, biodiesel and natural gas as alternative fuels. In adittion, the model permits parametric analysis to be made, such as valve diameter or connecting rod length variation.

1. THEORY

2.1. Introdution

The working space for modeling is formed by the head of the piston and the head and walls of the cylinder.

The model was developed with the following assumptions: the gas moisture inside the cylinder, in all the engine cycles, is formed of ideal gases; the pressure and temperature inside the cylinder is considered levelized in al the working space; inercial forces of gases or engine parts, as well potential and kinetic energies, is not considered; the pressure, temperature and elevation from sea level, are as follows: p_0 = 101,300.00 N m^{-2} ($p_0 = p_{atm}$), T_0 = 298.15 K and $z_0 = 0$, where"0" refers to standard conditions, p_{atm} is atmospheric pressure and z is the elevation from sea level.

Figure 1 below, decpits the working space:

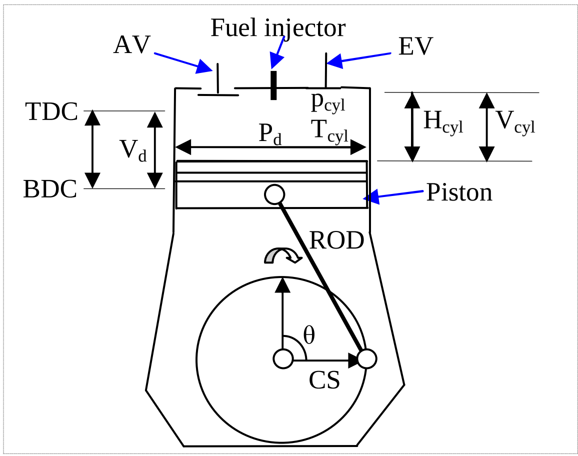

Figure 1. Working space, in the "in cylinder" approach

AV – admission valve; *BDC* – bottom dead center; *CS* – crankshaft diameter, [m]; *EV* – exhaust valve; H_{cyl} – cylinder height, [m]; p_{cyl} – gas moisture pressure, [N m^{-2}]; *ROD* – connecting-rod length, [m]; P_d – piston diameter, [m]; T_{cyl} – gas moisture temperature, [K]; θ – crankshaft angle, [rad]; *TDC* – top dead center; V_{cyl} – cylinder volume, [m^3]; V_d – displaced cylinder volume, [m^3].

2.2. Mathematical equations

The height of the piston, relative to the top of the cylinder and the instantaneous volume of the cylinder:

$$H_{ciyl} = ROD + CS\,(1 - \cos(\theta)) - \sqrt{ROD^2 - (CS)^2 \sin^2(\theta)} \tag{1}$$

$$V_{cyl} = H_{cyl}\left(\frac{\pi (P_d)^2}{4}\right) \tag{2}$$

2.2.1. Admission and exaustion strokes

The mass flow rate to or from the cylinder:

$$\dot{m}_{air} = \mathrm{C_d} A_{min} \sqrt{2\rho_{air}\,(p_{atm} - p_{cyl})} \tag{3}$$

A_{min} – valve area [m^2]; C_d – discharge coefficient; $\dot{m}_{air}$ – air flow rate, [kg s^{-1}]; ρ_{air} – air density, [kg m^{-3}].

The integration of Eq. (1) in time, gives the mass of air (m_{air}), trapped in cylinder. The mass of fuel injected:

$$m_f = \frac{m_{air}}{AFR} \tag{4}$$

AFR – air to fuel ratio; m_{air} – mass of air trapped in the cylinder, [kg]; m_f – mass of fuel injected, [kg].

The total mass of gas moisture inside the cylinder:

$$m_{gm} = m_f + m_{air} \tag{5}$$

m_{gm} – total gas moisture mass inside the cylinder, [kg].

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The instantaneous pressure inside the cylinder:

$$p_{cyl} = \frac{m_{gm} R_{gm} T_{cyl}}{V_{cyl}} \tag{6}$$

$$R_{mg} = c_{p_{mg}} - c_{v_{mg}} \tag{7}$$

$$c_{j_{gm}}(T_{cyl}) = \frac{\sum_{i=1}^{n} x_i M_i c_{j_i}(T_{cyl})}{\sum_{i=1}^{n} x_i M_i} \tag{8}$$

R_{gm} – universal constant of gas moisture inside the cylinder, [J kg^{-1} K^{-1}]; $c_{j_i}(T_{cyl})$ – gas specific heat at constant pressure or volume, [J kg^{-1} K^{-1}]; $c_{j_{gm}}(T_{cyl})$ – gas moisture specific heat at constant pressure or volume, [J kg^{-1} K^{-1}]; M_i – molecular mass of gas component, [kg Mol^{-1}]; x_i – molar fraction of gas component, [%].

The mass of fuel injected is considered in the present stroke and the energy released by combustion, in next item. Applying the conservation of the energy in the working space the instantaneous temperature attained is:

$$\frac{dT_{cyl}}{dt} = \left(\frac{\dot{Q}_p - p_{cyl}\frac{dV_{cyl}}{dt} + h_{f,in}\dot{m}_f + h_{air,in}\dot{m}_{air,in} - c_{v,air} T_{cyl}\dot{m}_{air} - c_{v,f} T_{cyl}\dot{m}_f}{c_{v_{air}} m_{air} + c_{cv_f} m_f} \right) \tag{9}$$

$h_{air,in}$ $h_{f,in}$ – air and fuel enthalpy in the engine cylinder, [J kg^{-1}];

$\dot{Q}_p$ – Heat transfer from or to the engine cylinder walls, [W]

The enthalpies of the air and fuel:

$$h_{air}(T_{cyl}) = c_{p\,air}(T_{cyl}) \tag{10}$$

$$h_f(T_{cyl}) = c_{p\,f}(T_{cyl}) \tag{11}$$

$c_{p\,air}, c_{p\,f}$ – air and fuel specific heats at constant pressure, [J kg^{-1} K^{-1}]; $h_{air}(T_{cyl})$, $h_f(T_{cyl})$ – air and fuel enthalpies, [J kg^{-1} K^{-1}].

2.2.2. Heat transfer from or to cylinder walls

The heat transfer from or to the cylinder walls, is attained using Dittus and Boelter correlation, in the equation's set:

$$\dot{Q}_p = -(\dot{Q}_{conv} + \dot{Q}_{rad}) \tag{12}$$

$$\dot{Q}_{conv} = h_{conv} A_{cil} (T_{cyl} - T_w) \tag{13}$$

$$h_{conv} = 0{,}023.(Re)_{P_d}^{4/5} (Pr)^{1/3} \frac{\kappa_{gas}}{P_d} \tag{14}$$

$$(Re)_{P_d} = \frac{\overline{S}_p P_d}{\nu_{gas}} \tag{15}$$

$$\overline{S}_p = \frac{(CS)N}{60} \tag{16}$$

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$$A_{cyl} = \left(H_{cyl}\right)\pi.P_d + \frac{\pi.P_d^2}{4} \qquad (17)$$

$$\dot{Q}_{rad} = \varepsilon_w A_{cil}\sigma(T_{cyl}^4 - T_w^4) \qquad (18)$$

A_{cyl} – cylinder heat transfer area, [m^2]; h_{conv} – convection heat transfer coefficient, [W m^{-2} K^{-1}]; N – crankshaft rotations per minute, [rpm]; Pr – Prandtl number of gas moisture; $(Re)_{P_d}$ – piston diameter Reynolds number; $\dot{Q}_{conv}$, $\dot{Q}_{rad}$, $\dot{Q}_p$ – cylinder walls heat transfer by convection, radiation and total, [W]; $\bar{S}_p$ – mean piston speed, [m s^{-1}]; T_w – wall cylinder temperature, [K]; ε_w – cylinder walls emissivity, [%]; κ_{gas} – gas moisture thermal conductivity, [W m^{-1} K^{-1}]; ν_{gas} – gas moisture cinematic viscosity, [m^2 s^{-1}]; σ – Stefan-Boltzmann constant, [W m^{-2} K^{-4}].

2.2.3. Compression and expansion strokes.

In the compression and expansion strokes, the pressure may be attained using Eq. (6) up to (8). The instantaneous temperature is attained applying the conservation of energy principle and the resulting equation is:

$$\frac{dT_{cyl}}{dt} = \frac{\left(\dot{Q}_p - \frac{dW_p}{dt}\right)}{m_{mg}c_{v_{mg}}} \qquad (19)$$

W_p – piston engine work, [J].

2.2.4. Combustion

The combustion reaction, considering the existence of diesel, biodiesel and natural gas in the fuel mixture, is as follows:

$$yC_\delta H_\zeta O_\xi + (1-y)(xC_\alpha H_\beta + (1-x)CH_4) + \lambda a(O_2 + 3{,}76N_2) \rightarrow bCO_2 + cH_2O + 3{,}76\lambda aN_2 + (\lambda - 1)aO_2 \qquad (20)$$

δ, ξ, ζ – respectively, carbon, oxygen and hydrogen number of atoms in the biodiesel molecule; α, β – carbon and hydrogen number of atoms in the diesel molecule; a, b, c – stoichiometric coeficients in the combustion reaction, [%]; x, y – percentual of diesel and biodiesel in fuel mixture, [%].

$$\lambda = \frac{AFR_r}{AFR_{est}} \qquad (21)$$

AFR_{est} , AFR_r – air to fuel ratios, stoichiometric and real; λ – the inverse of equivalence ratio.

The heat released in the combustion in molar basis, is the diference in the entalpies of products and reagents:

$$\overline{Q}_{comb} = \overline{h}_{prod} - \overline{h}_{reag} \qquad (22)$$

$\overline{Q}_{comb}$ – combustion released heat, molar basis, [J mol^{-1}]; $\overline{h}_{prod}$, $\overline{h}_{reag}$ – products and reagents molar enthalpies, [J mol^{-1}].

$$Q_{comb} = \frac{\overline{Q}_{comb}}{mol_f} \qquad (23)$$

$$mol_f = ymol_{biodiesel} + (1-y)(xmol_{diesel} + (1-x)mol_{nat\ gas}) \qquad (24)$$

Q_{comb} – combustion released heat, mass basis, [J kg^{-1}]; mol_i – diesel, biodiesel, nat. gas molecular mass, [kg mol^{-1}].
The heat released in the combustion, time basis is:

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$$\dot{Q}_{comb} = m_f \frac{Q_{comb}}{t_{comb}} \tag{25}$$

$$t_{comb} = \frac{60\psi}{2\pi N} \tag{26}$$

$\dot{Q}_{comb}$ – combustion released heat, time basis [W]; t_{comb} – combustion time, [s] ; ψ – combustion angle, [rad].

During combustion, the instantaneous pressure inside the cylinder is given by:

$$\frac{dp_{cyl}}{dt} = \frac{R_{gm}\left(\dot{Q}_{comb} + \dot{Q}_p - p_{cyl}\frac{dV_{cyl}}{dt} - \frac{p_{cyl}c_{v_{gm}}}{R_{mg}}\frac{dV_{cyl}}{dt}\right)}{V_{cyl}c_{v_{gm}}} \tag{27}$$

2.2.5. Mean indicated pressure, mean friction pressure, mean effective pressure, power developed by engine piston, power and torque developed by engine and net output of the engine.

The mean indicated pressure in the head of the engine piston is:

$$p_{med_{ind}} = \frac{W_{\mathrm{ind_p}}}{V_d} \tag{28}$$

$p_{med_{ind}}$ – mean indicated pressure in piston engine, [N m^{-2}]; $W_{\mathrm{ind_p}}$ – indicated work developed by engine piston, [J].

The mean friction pressure may be attained using empirical correlations for ICE engines with direct injection of fuel according to Heywood (1988), which expresses the losses (e.g., friction, pumping) in form of a pressure drop:

$$p_{med\ fr} = C_1 + \frac{48\,N}{1000} + 0{,}4\bar{S}^{\,2}_{pist} \tag{29}$$

$$p_{med\ ef} = p_{med\ ind} - p_{med\ fr} \tag{30}$$

$p_{med_{fr}}$ – mean friction pressure, [N m^{-2}]; C_1 – constant, 75.000 N m^{-2}; $p_{med_{ef}}$ – mean effective pressure, [N m^{-2}].

The net power developed by engine piston and the power, torque and net output developed by the engine, are:

$$\dot{W}_{ef\ p} = \frac{p_{med\ ef}V_d}{\Delta t_{cm}} \tag{31}$$

$$\dot{W}_{ef} = \dot{W}_{ef_p} NP \tag{32}$$

$$\tau_{ef} = \frac{60\dot{W}_{ef}}{2\pi N} \tag{33}$$

$$\eta_{ef} = \frac{\dot{W}_{ef}}{\dot{Q}_{comb}} \tag{34}$$

$\dot{W}_{ef_p}$, $\dot{W}_{ef}$ – net power developed by engine piston, and by engine, [W]; Δt_{mc} – time spent in every engine cycle, [s]; τ_{ef} – engine torque, [N m^{-1}]; η_{ef} – engine net output, [%]; NP – number of engine pistons.

3. RESULTS AND DISCUSSION

3.1. Results

The tolerence for model validation was established as 10% in modulus for the value of model parameters. The fuel consumption gaves results inside the tolerance range, but the model must be adjusted, for power, torque and effective output. The adjustament was made over the Eq. (12) and over the Eq.(29), through constant coeficients as multipliers:

$$\dot{Q}_p = -\left(\dot{Q}_{conv} + \dot{Q}_{rad}\right) 1.45 \tag{35}$$

1.45 – adjustament constant of Eq. (12).

$$p_{med\ fr} = \left(C_1 + \frac{48\ N}{1000} + 0{,}4\,\overline{S}^{\,2}_{pist} \right) 0.8 \tag{36}$$

0.8 – adustament constant of Eq. (29).

The model was tested with Lintec 3LD 1500 and 4LD 2500, Agrale M790 and MWW 229.6 diesel engines. The simulations with alternative fuels proposed, were made with the Lintec 4LD 2500 and MWM 229.6 diesel engines.

Tables 1 and 2 presents, respectively, the values of fuel consumption (*FC*), power, torque and effective output for Agrale M 790 and Lintec 4LD 2500 engines, using diesel as fuel and showing the errors between the values attained by the model (*m*) and the real (*r*) values from catalogs, demonstrating the model effectiveness adjustments. (The parameter D_{val} is the nominal valve diameter, which the engine was simulated for model validation).

Table 1. Fuel consumption, power, torque and net output with erros for Agrale M 790 engine simulation using diesel.

N [rpm]	D_{val} [mm]	FC_m [kg h^{-1}]	FC_r [kg h^{-1}]	*Error* [%]	$\dot{W}_{ef_m}$ [W]	$\dot{W}_{ef_r}$ [W]	τ_{ef_m} [N m]	τ_{ef_r} [N m]	η_{ef_m} [%]	η_{ef_r} [%]	*Error* [%]
1600	170,0	2,91	2,97	-2,0	12091,1	12063,7	72,2	72,0	37,2	37,1	0,2
2000	170,0	3,71	3,88	-4,4	16149,4	16126,8	77,1	77,0	39,1	39,0	0,1
2400	170,0	4,47	4,49	-0,4	19317,3	19352,2	76,9	77,0	38,7	38,8	-0,2
3000	170,0	5,42	5,93	-8,6	21949,0	21991,1	69,9	70,0	36,5	36,6	-0,2

Table 2. Fuel consumption, power, torque and net output with erros for Lintec 4LD 2500 engine simulation using diesel.

N [rpm]	D_{val} [mm]	FC_m [kg h^{-1}]	FC_r [kg h^{-1}]	*Error* [%]	$\dot{W}_{ef_m}$ [W]	$\dot{W}_{ef_r}$ [W]	τ_{ef_m} [N m]	τ_{ef_r} [N m]	η_{ef_m} [%]	η_{ef_r} [%]	*Error* [%]
1600	175,0	6,11	6,46	-5,9	26211,9	26473,2	157,8	158,0	36,5	36,4	0,1
2000	175,0	7,58	7,63	-0,7	31768,8	31415,9	153,7	152,0	35,8	35,4	1,12
2400	175,0	9,37	9,06	3,4	37192,9	37213,8	148,0	148,1	35,1	35,0	0,06
2800	175,0	10,8	9,82	9,9	40310,4	39877,3	137,5	136,0	34,2	33,8	1,09

In tables 1 and 2, the efficacy of the adjustment can be seen, considering that the errors attained comparing the model values and the real ones, are less than 10.0%, in modulus.

Table 3, presents a exemple of parametric analysis, showing the effects of valve diameter variation in the power developed by Lintec 4LD 2500 engine and Fig. 2 presents the corresponding graphic:

Table 3. Power attained with valve diameter variation, for Lintec 4LD 2500 engine simulation using diesel.

N [rpm]	$\dot{W}_{ef}$ [W]; D_{val} = 150 mm	$\dot{W}_{ef}$ [W]; D_{val} = 175 mm	$\dot{W}_{ef}$ [W]; D_{val} = 185 mm	$\dot{W}_{ef}$ [W]; D_{val} = 230 mm
1600	23538,3	26211,9	29558,6	29955,2
2000	28966,4	31768,8	34281,8	34955,3
2400	32675,9	37192,9	40364,2	41184,2
2800	37662,7	40310,4	45468,5	46576,6

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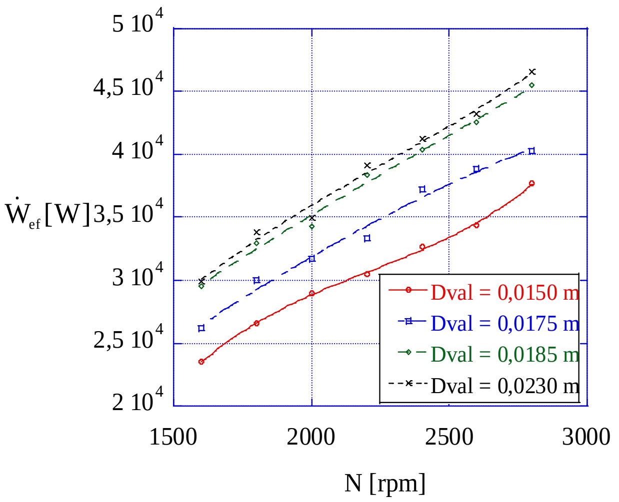


Figure 2. Power simulation, Lintec 4LD 2500 engine, with valve diameter variation

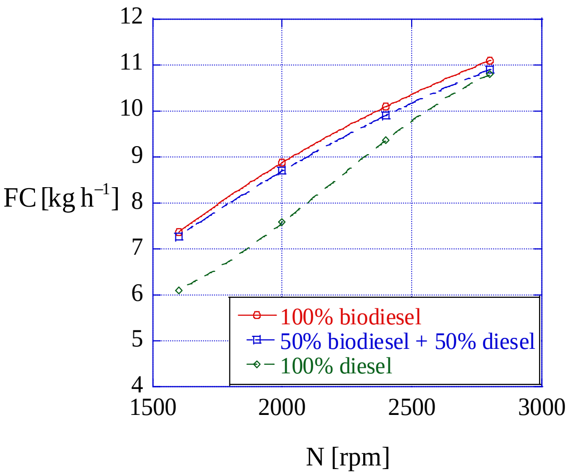
Based on the graphic of the Fig. 2, it may be concluded that for a lower valve diameter, less air flows to cylinder and how AFR at each simulation is constant, less fuel is injected in the cylinder, resulting less power produced. When valve diameter rises, the power rises too, but not proportionally, in such a way that for bigger values of valve diameter the increasing in power is not significant. This may be seen directly from the graphic of Fig. 2.

Table 4, shows the values of fuel consumption, power and effective output for Lintec 4LD 2500 diesel engine, simulated with the model, using mixtures of diesel and biodiesel as fuel:

Table 4. Fuel consumption, power and net output for Lintec 4LD 2500 engine simulation using diesel and biodiesel.

N [rpm]	100% biodiesel			50% biodiesel + 50% diesel			100% diesel		
	$\dot{W}_{ef}$ [W]	η_{ef} [%]	FC [kg h^{-1}]	$\dot{W}_{ef}$ [W]	η_{ef} [%]	FC [kg h^{-1}]	$\dot{W}_{ef}$ [W]	η_{ef} [%]	FC [kg h^{-1}]
1600	25884,3	35,7	7,37	26044,8	36,3	7,28	26211,9	36,5	6,11
2000	31346,2	35,3	8,88	31553,4	35,5	8,70	31768,8	35,8	7,58
2400	36587,3	34,6	10,1	36897,4	34,9	9,91	37192,9	35,1	9,37
2800	39655,6	33,7	11,1	39945,3	33,9	10,9	40310,4	34,2	10,8

Figure 3 shows graphics of fuel consumption and power versus rotation of model simulations for Lintec 4LD 2500 engine, using mixtures of biodiesel and diesel as fuel, based on data of table 4:



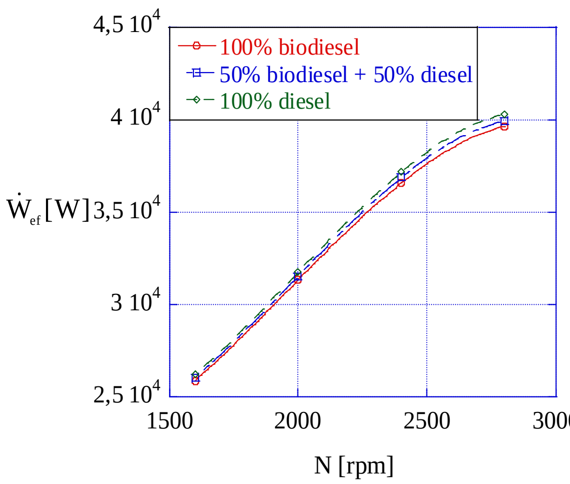


Figure 3. Fuel consumption and power simulation, Lintec 4LD 2500 engine, using biodiesel and diesel

In the graphics of Fig. 3, the biodiesel released less power then diesel and its fuel consumption is sensitively greater then using diesel. In the other hand, this elevation in fuel consumption compensates the loss in power supplied by the engine, which in such cases is about 1.0% lower compared with the power supplied using only diesel.

Table 5, shows the values of fuel consumption, power and net output for Lintec 4LD 2500 diesel engine, simulated with the model, using mixtures of natural gas and diesel as fuel:

Table 5. Fuel consumption, power and net output for Lintec 4LD 2500 engine simulation using natural gas and diesel.

N [rpm]	100% natural gas			50% natural gas + 50% diesel			100% diesel		
	$\dot{W}_{ef}$ [W]	η_{ef} [%]	FC [kg h^{-1}]	$\dot{W}_{ef}$ [W]	η_{ef} [%]	FC [kg h^{-1}]	$\dot{W}_{ef}$ [W]	η_{ef} [%]	FC [kg h^{-1}]
1600	25776,0	35,7	5,19	25983,5	36,2	5,56	26211,9	36,5	6,11
2000	31278,8	35,1	6,55	31464,4	35,4	7,12	31768,8	35,8	7,58
2400	36296,2	34,4	7,76	36792,4	34,7	8,68	37192,9	35,1	9,37
2800	39241,7	33,5	8,48	39832,2	33,8	9,69	40310,4	34,2	10,8

Figure 4 shows grafics of fuel consumption and power versus rotation of model simulations for Lintec 4LD 2500 engine, using mixtures of natural gas and diesel, based on data of table 5:

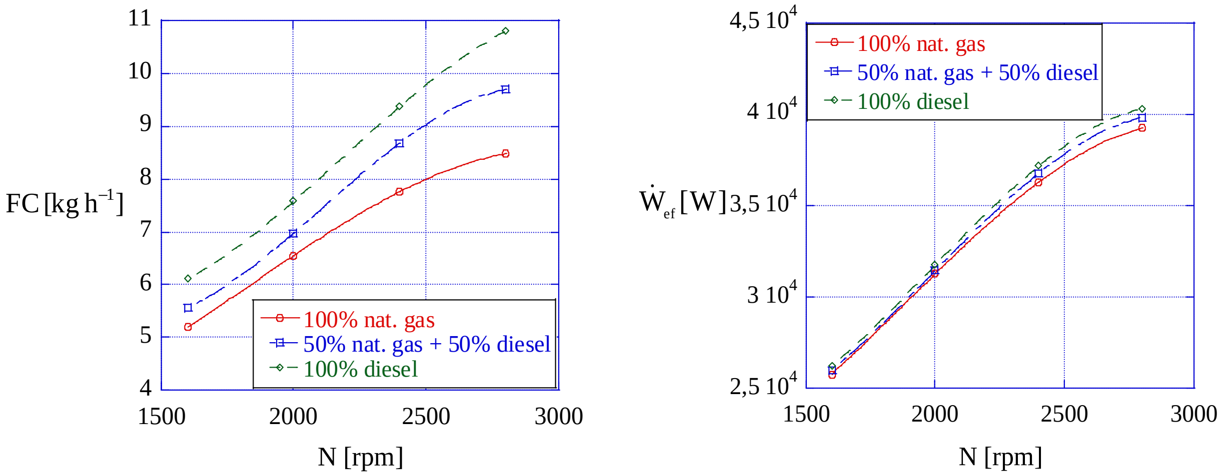


Figure 4. Fuel consumption and power simulation, Lintec 4LD 2500 engine, using natural gas and diesel

In the graphics of Fig 4, it may be seen that natural gas released more power then diesel and consequently its fuel consumption is lower then the diesel. With the power occurs that, in spite of the high energy released in combustion and the lower fuel consumption, the value of power supplied by the engine about 1.5 % lower compared with the power supplied using only diesel.

Table 6, shows the values of fuel consumption, power, torque and net output for Lintec 4LD 2500 diesel engine, simulated with the model using mixtures of natural gas and biodiesel as fuel:

Table 6. Fuel consumption, power, torque and net output for Lintec 4LD 2500 engine simulation using natural gas and biodiesel.

N [rpm]	100% natural gas			50% natural gas + 50% biodiesel			100% biodiesel		
	$\dot{W}_{ef}$ [W]	η_{ef} [%]	FC [kgh^{-1}]	$\dot{W}_{ef}$ [W]	η_{ef} [%]	FC [kgh^{-1}]	$\dot{W}_{ef}$ [W]	η_{ef} [%]	FC [kgh^{-1}]
1600	25776,0	35,7	5,19	25884,3	35,7	7,36	25884,3	35,7	7,37
2000	31278,8	35,1	6,55	31346,2	35,3	8,87	31346,2	35,3	8,88
2400	36296,2	34,4	7,76	36587,3	34,6	10,08	36587,3	34,6	10,1
2800	39241,7	33,5	8,48	39655,6	33,7	11,08	39655,6	33,7	11,1

Figure 5 shows grafics of fuel consumption and power versus rotation of model simulations for Lintec 4LD 2500 engine, using mixtures of natural gas, biodiesel, and diesel based on data of table 6:

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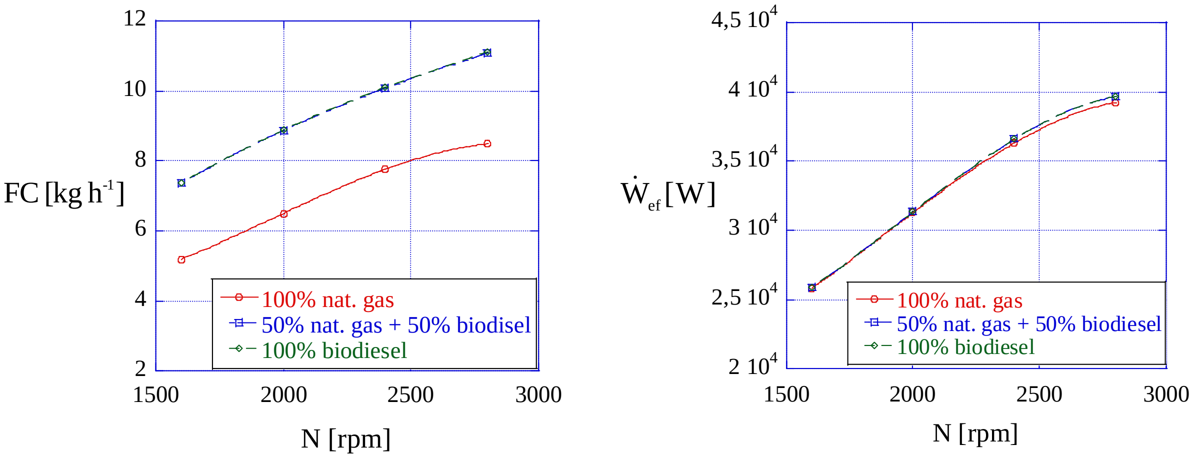

Figure 5. Fuel consumption and power simulation, Lintec 4LD 2500 engine, using natural gas and biodiesel

It may be seen in the graphics of Fig. 5, that natural gas and fuel mixtures of natural gas with biodiesel releases almost the same power and have almost the same fuel consumption. With the power, occurs something similar with the others fuel mixtures where the power produced difers one from each other about 1.0% in value.

Table 7, shows the values of fuel consumption, power, torque and net output for Lintec 4LD 2500 diesel engine, simulated with the model using mixtures of natural gas, biodiesel and diesel as fuel:

Table 7. Fuel consumption, power, torque and net output for Lintec 4LD 2500 engine simulation using natural gas, biodiesel and diesel.

N	30% natural gas + 50% biodiesel + 20% diesel			50% natural gas + 30% biodiesel + 20% diesel			100% diesel		
	$\dot{W}_{ef}$ [W]	η_{ef} [%]	FC [kgh^{-1}]	$\dot{W}_{ef}$ [W]	η_{ef} [%]	FC [kgh^{-1}]	$\dot{W}_{ef}$ [W]	η_{ef} [%]	FC [kgh^{-1}]
1600	25815,1	36,1	5,77	26004,8	36,3	5,19	26211,9	36,5	6,11
2000	31454,2	35,4	7,18	31642,1	35,6	6,55	31768,8	35,8	7,58
2400	36354,4	34,7	8,91	36775,4	34,9	7,96	37192,9	35,1	9,37
2800	39479,9	33,7	10,3	39865,3	33,9	9,28	40310,4	34,2	10,8

Figure 6 shows the grafics of fuel consumption and power versus rotation of model simulations for Lintec 4LD 2500 engine, using mixtures of natural gas, biodiesel, and diesel based on data of table 7:

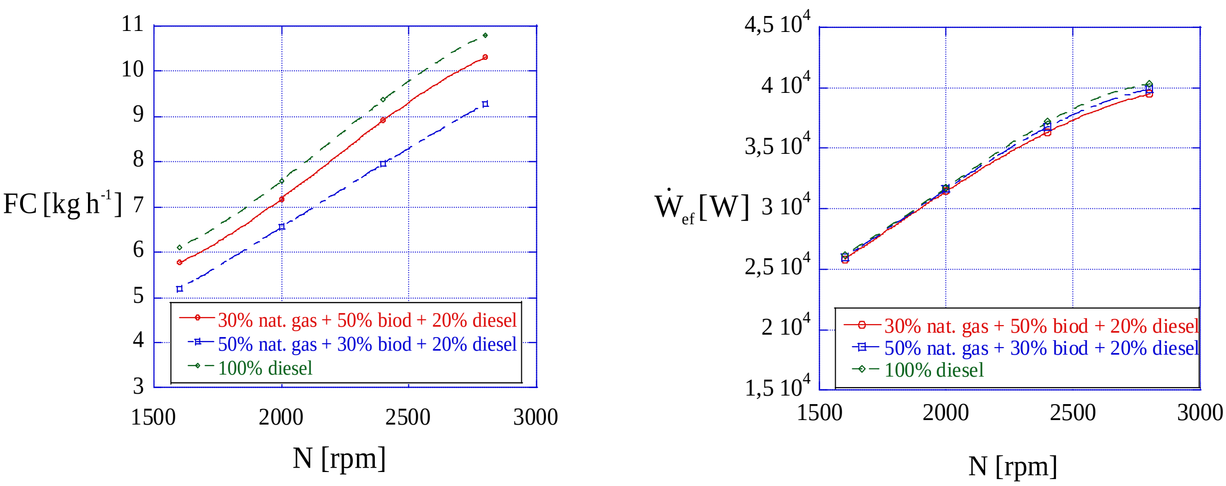

Figure 6. Fuel consumption and power simulation, Lintec 4LD 2500 engine, using natural gas, biodiesel and diesel

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In the graphics of Fig. 6, once biodiesel releases less power and natural gas releases more power then diesel, the fuel consumption undergo a compensation, depending on the percentage of biodiesel and natural gas in the fuel mixture: less natural gas, greater fuel consumption, and vice versa. The greater fuel consumption compensates the loss in power supplied by the engine, what in such case is less than 2% compared with the power supplied using only diesel.

3.2. Discussion

The model was tested, adjusted and experimentaly validated in diesel engines using diesel as fuel. Then, simulations were made using fuel mixtures of natural gas, biodiesel and diesel proving that the model is ready to be used as a computational tool for simulation, control design and optimization of diesel engines working with alternative fuels.

4. CONCLUSIONS

The main conclusions this paper presents, are as follows:

- Using only biodiesel as fuel in diesel engines, the power reduces about 1.0 % and the fuel consumption rises about 12.0% compared to using diesel as fuel;
- Using only natural gas as fuel in diesel engines, the power reduces about 2,0% and the fuel consumption reduces about 13.0% compared to using diesel as fuel;
- Fuel mixtures using 50% of biodiesel and/or 50.0% of natural gas produces power values wich diference is less then 1.0% compared one to each other, and
- Variation in valve diameter causes proportional variation in the power developed by the engine, if the value of the valve diameter is near the nominal value for engine operation. Greater values then the nominal for valve diameter don’t produce senstive variation in the power developed by the engine, increasing only a few Watts in engine power.

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