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CONSTRUCTAL DESIGN OF BIODIESEL / BIOGAS DRIVEN COMPRESSION IGNITION INTERNAL COMBUSTION ENGINES

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Abstract.
This paper aimed at to present and discuss the minimal exergy destruction for 4-strokes compression ignition internal combustion engines (CI-ICE) driven by biodiesel/biogas mixtures, subject to physical constraints, i.e., the CI-ICE constructal design. Geometric and operating parameters (e.g., rpm, piston and crank diameter, conrod-crank ratio, bore-stroke ratio, compression ratio, cutoff ratio, etc.) are the basis for the model equations, which are capable to calculate the indicated mean effective pressure, indicated power and torque with respect to crank speed. The second law equation (entropy generation rate) is developed at a mathematical model previously experimentally validated. The system mathematical model allows for the exploration of operational and design parameters following a constructal approach, where it is possible to achieve a functional and optimal design, i.e., maximum thermal efficiency and minimal exergy destruction. In this paper, the optimal cutoff ratio is shown as well as the compression ratio as a function of conrod-crank and bore-stroke ratio. In addition, the results are illustrated comparing the diesel and renewable alternative fuels (biodiesel/biogas mixtures only). In conclusion, this model can be a useful tool for simulation, design, and optimization of CI-ICE for any combination of diesel, biodiesel, and biogas fuels.

Keywords: *biodiesel/biogas, minimal exergy destruction, constructal design, optimal cutoff ratio, optimal compression ratio.*

1. INTRODUCTION

One of the greatest challenges of the 21st century for humanity's future is meeting global energy demand in a sustainable way. This means that the correct use of renewable alternative fuels, preferably of biological origin, is urgently required, as well as the conscious energy use. Taking into consideration fuel sources, the scientific community has been looking for them, for example, biodiesel, biogas or natural gas. Biodiesel is a biodegradable and renewable fuel which can be obtained from different sources, such as vegetable oils, animal fat, waste oils, etc., without affecting food production. Biogas is a valuable source of fuel and it is considered one of the best economical alternatives since its adaptability for gasoline and diesel fuels are easily met. Diesel engines are one of the main sources of rapid growth in energy / fuel consumption around the world due to its application in transport solutions. Consequently, pollution levels have increased rapidly, accelerating planet's environmental deterioration. In order to minimize these effects, regulations for limiting pollutants emissions are increasingly stringent. Therefore, the science has sought through research to provide answers and solutions for these problems. The concept of the second law of thermodynamics, for example, might be applied to reduce consumption materials and energy sources such as fuels. This is possible because the second law shows which are the paths for a CI-ICE to operate at its optimum point during applications, Bejan (2002). Moreover, with the approach of the second law, it is entirely possible to restrict the mathematical model of a CI-ICE to depend on the environmental conditions and their physical constraints, i.e., geometrics or design parameters, Bejan (2000). Thus, this modeling might be a powerful tool to find the best performance for a CI-ICE. Therefore, if the optimal point of engine operation for a certain application is known, it might provide a desired net power in the crankshaft of the engine with the lowest fuel consumption and it can still provide benefits to the environmental issues of our planet, as greenhouse effect.

Knowing the finite weight, cost and space in any engineering project, a constraint accounting for the total volume occupied by the engine is imposed or required. Therefore, engineering tools are needed to simulate and to understand the thermal and physical behaviors subject to constrains during the development of engine design. Simulation tools are then utilized to optimize some engine operating and design parameters definition to achieve the maximum cycle efficiency with the smaller possible project cost. When a mathematical model thermodynamic is applied on a physical system and it is optimized, the design result for this specific condition is the better structure (i.e., geometry, architecture, configuration, patters and topology). A complex physical system configuration is generated, shape and configuration for example, as a constructive finite-size method, and this generation in engineering has been called constructal design by Bejan (2002) at constructal theory.

Some authors have studied CI diesel engines feed with biodiesel + biogas or natural gas. Firstly, follows the authors that published papers related to dual fuel mode diesel engine with biodiesel + biogas or natural gas (NG) only. Luijten and Kerkhof (2011), studied the Jatropha biodiesel + biogas for rural electrification application and they evaluated the brake thermal efficiency (BTE) and volumetric efficiency with fuels blends. In the same time, Yoon and Lee (2011) introduced exhaust emission, BTE, brake-specific fuel consumption (BSFC) and injection delay analysis as well. Ryu (2013a, 2013b) and Tarabet et al. (2014) analyzed biodiesel + NG with its mixtures. They analyzed BTE, BSFC and emission in common, but Tarabet et al. (2014) added the air-to-fuel ratio (AFR) analysis. Finally, Barik et al. (2017), Bora and Saha (2017) and Kalsi and Subramanian (2017), studied biodiesel + biogas blends (KME, RBB and Pongamia pinnata, respectively), and biogas with its blends (CH_4+CO_2). Both Barik et al. (2017) and Bora and Saha (2017), introduced compression ratio (CR) analysis relating to exhaust emission and Kalsi and Subramanian (2017) itemized dual-fuel mode with biogas content. These authors have developed mathematical model to analyze the performance of a diesel engine, considering the first law of thermodynamics approach. For second law approach or exergy analysis, many other authors have studied it for other dual fuel mode. Among the more recently publications related to diesel + biodiesel are Azoumah et al. (2009), Debnath et al. (2013), Jafarmadar and Nemati (2016), Nemati et al. (2016) and Rakopoulos and Giakoumis (2006), which the last one did a summary of second law researches. Publication related to diesel + biogas (NG) fuel mode, Mahabadipour et al. (2017), Verma et al. (2017) and Ramos da Costa et al. (2012) studied exergy efficiency for diesel engine. In addition, some authors have introduced a third fuel mode, Barik et al. (2017) already cited above have studied the combination (Karanja methyl ester (KME) + biogas + diethyl ether (DEE)). Khoobbakht et al. (2016) researched the blends (diesel + biodiesel + ethanol) for a DI engine and Krishnamoorthi and Malayalamurthi (2018) investigated the follow blends (diesel + biodiesel (bael oil) + DEE) almost similar to Barik et al. (2017).

Regarding CI-ICE numerical simulation, one of the most utilized simulation tools is the code KIVA, Amsden D.C. and Amsden A.A. (1993). It is a reliable and effective complex code for CI-ICE simulation, but computational time could be an issue. Isermann (2013) recently presented a comprehensive review work on internal combustion engines advanced in modeling and identification methods and engine control. However, a simpler code has been proposed by Graciano et al. (2016) for dynamic simulation of the 4-strokes for CI-ICE with alternative fuels mixtures including the combustion process. Thus, this code is capable to analyze the engine performance and optimize its parameters with just fast and excellent accurate computers.

The bibliographic review shows a lack of studies on thermodynamic optimization of CI diesel engine dual-fuel mode and constructal design mainly for biodiesel / biogas mixtures. Therefore, the objective of this paper is to present, compare and discuss the thermal and exergy efficiencies through a dynamic mathematical model developed for CI-ICE powered with diesel and alternative fuels (biodiesel / biogas mixtures), subject to geometric and operational parameters for engine design (i.e., constructal design approach). Operational and design conditions that lead to minimal system irreversibilities or minimal exergy destruction are sought based on Bejan (1982), Bejan et al. (1996), Bejan (2000) and Bejan (2002).

2. MATHEMATICAL MODEL

Figure 1 depicts a schematic diagram of the CI diesel engine configuration considered in this paper, consisting of: piston head, cylinder walls, bore cylinder, slider-crank mechanisms, the system valve (admission and exhaust) and fuel nozzle.

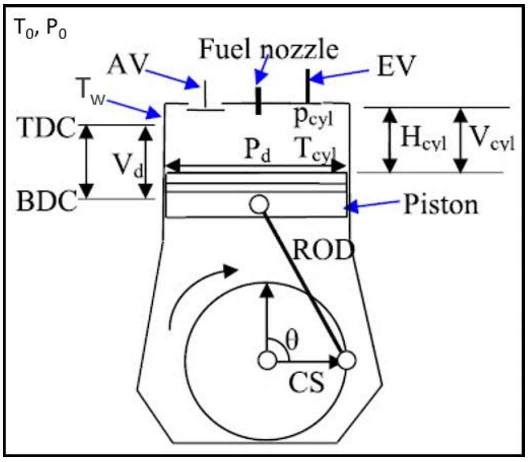

Figure 1. Compression ignition internal combustion engines working space schematic diagram adapted from Graciano et al. (2016).

The stroke volume – V_d (m^3) is defined as been the difference between *TDC* – top dead center and *BDC* – bottom dead center, displacement piston – H_{cyl} (m), piston diameter – P_d (m), crank-shaft radius – *CS* (m), connecting-rod length – *ROD* (m), crank angle – θ (rad), cylinder wall temperature – T_w (K), admission valve – *AV*, exhaust valve – *EV*, and fuel nozzle. In addition, the T_{cyl}, p_{cyl} and V_{cyl} are the instantaneous values for temperature (K), pressure (N. m^{-2}) and volume (m^3), respectively. This schematic diagram has a thermal contact with atmospheric heat reservoir been the ambient temperature – T_0 (K) and ambient pressure – P_0 (N. m^{-2}).

The first law of thermodynamics equation already implemented by Graciano et al. (2016) is neglected in this paper, because the focus of this work is to develop the second law equations for schematic diagram depicted on Fig. 1.

This model is comprised of five ordinary differential equations (ODE) for each stage of the diesel cycle engine: admission, compression, expansion, exhaust strokes and combustion process. All equations are based in the generalized entropy generation differential equation as proposed by Bejan (2000):

$$\dot{S}_{gen,k} = \left(\frac{dS_{vc}}{dt}\right)_k - \left(\sum_{i=1}^{n}\frac{\dot{Q}_i}{T_i}\right) + \left(\sum_{out}\dot{m}s - \sum_{in}\dot{m}s\right)_k \geq 0 \tag{1}$$

The first term of differential equation is the entropy generation within the control volume. This term is calculated utilizing the thermodynamic relation with ideal gas assumption as Eq. (2) shows. The second term is related to all heat transfer rates - $\dot{Q}_i$ interaction with itself temperature boundary surface - T_i and the third term is the entropy mass flow balance inside of the control volume as depicted in Fig. 1, where the $\dot{m}$ is as Eq. (5) and *s* is determined by an approximation polynomial.

$$\left(\frac{dS_{vc}}{dt}\right)_k = \frac{m_k ds}{dt} = m_k\left(\frac{\bar{c}_p}{T_{cyl}}\frac{dT_{cyl}}{dt} + \frac{R}{P_{cyl}}\frac{dP_{cyl}}{dt}\right) \tag{2}$$

Where m_k (kg) is the total mass inside of the control volume and *k* represents each stage of the diesel cycle engine, $\bar{c}_p$ is the average specific heat at constant pressure (J. kg^{-1}. K^{-1}) and *R* is the gas constant (J. kg^{-1}. K^{-1}).

2.1 Admission stroke

In this engine stage the cylinder is moving down and the admission valve – AV is opened for the air to be admitted into the cylinder. The balance of the entropy generation applied in this workspace is:

$$\dot{S}_{gen,adm} = \left(\frac{dS_{vc}}{dt}\right)_{adm} - \left[\left(\frac{\dot{Q}_{air}+\dot{Q}_{fuel}}{T_0}\right) - \left(\frac{\dot{Q}_w+\dot{Q}_{prod}}{T_w}\right)\right] - \left(\sum_{in}\dot{m}s_R\right)_{adm} \geq 0 \tag{3}$$

The entire heat transfer rate within the second term on the right side of the equation ($\dot{Q}_{air}$ – admitted air heat transfer rate, $\dot{Q}_{fuel}$ – fuel heat transfer rate, $\dot{Q}_w$ – total cylinder wall heat transfer rates where it is calculated taking account the radiation and convection transfer modes and $\dot{Q}_{prod}$ – combustion products heat transfer rate) are calculated once known the temperatures and the mass flow for each differential time. In addition, the parameter T_w (K) is an input and it means engine operating temperature. The last term is related to the entropy mass flow that entered in the system and all variants are known or they are inputs, Bejan et al. (1996):

$$\left(\sum_{in} \dot{m}s_R\right)_{adm} = \dot{m}_{vc}\left\{\sum_{i=1}^{n} \gamma_{ri}\left[\bar{s}^{\circ}\left(T_{cyl}\right) - \bar{R}\,\mathbf{ln}\left(\frac{y_i P_{cyl}}{P_0}\right)\right]_{ri}\right\} \tag{4}$$

The *ri's* are all reactants components admitted into to cylinder (e.g., O_2; N_2 and fuel). The γ_{ri} represents molar fraction of reactants, $s^o\,(T_{cyl})$ represents instantaneous entropy as function of the cylinder temperature (J. K^{-1}) and y_i the partial molar concentration of each *i's* components. Also, the mass flow rate into and out of the cylinder is defined as follows:

$$\frac{dm_j}{dt} = \dot{m}_j = \pm C_d A_{\mathbf{min}} \sqrt{2\rho_{air}\left|p_0 - p_{cyl}\right|} = \frac{\dot{m}_{air}}{AFR_r} \tag{5}$$

C_d is the discharge coefficient parameter, A_{min} represents valve minimum opening area (m^2), and *j* might be *air* (admission stroke) or *prod* (exhaust products), ρ_{air} is the admitted air density (kg. m^{-3}), AFR_r is the air-to-fuel ratio and $\dot{m}_j$ represents the mass flow rate (kg. s^{-1}). Equation 6 defines λ – inverse of the equivalence ratio (i.e., excessive admitted air ratio), where AFR_{st} represents air-to-fuel ratio stoichiometric.

$$\lambda = \frac{AFR_r}{AFR_{st}} \tag{6}$$

2.2 Compression stroke

In this engine stage, the cylinder is compressing the admitted air and the admission valve – AV is closed. Therefore, the workspace is treated as a closed system, with no leak assumption. The balance of entropy generation applied it is as follows:

$$\dot{S}_{gen,comp} = \left(\frac{dS_{vc}}{dt}\right)_{comp} + \left(\frac{\dot{Q}_{air} + \dot{Q}_{fuel} + \dot{Q}_w + \dot{Q}_{prod}}{T_w}\right) \geq 0 \tag{7}$$

2.3 Expansion stroke

In this engine stage starts after the combustion process to be finalized. The workspace is also treated as a closed system, with under a no leak assumption. The balance of entropy generation applied is as follows:

$$\dot{S}_{gen,\mathbf{exp}} = \left(\frac{dS_{vc}}{dt}\right)_{\mathbf{exp}} + \left(\frac{\dot{Q}_w + \dot{Q}_{prod}}{T_w}\right) \geq 0 \tag{8}$$

2.4 Exhaust stroke

In this stage of the engine the cylinder is moving upwards and the exhaust valve – EV is opened for the combustion products to be drop out of the cylinder. Thus, the balance of the entropy generation applied in this workspace is:

$$\dot{S}_{gen,exh} = \left(\frac{dS_{vc}}{dt}\right)_{exh} + \left(\frac{\dot{Q}_w + \dot{Q}_{prod}}{T_w}\right) + \left(\sum_{out} \dot{m}s_P\right)_{exh} \geq 0 \tag{9}$$

Where the last term is about the entropy mass flow that drop out of the system and its formulation is as follows:

$$\left(\sum_{out} \dot{m}s_P\right)_{exh} = \dot{m}_{vc}\left\{\sum_{i=1}^{n} \gamma_{pi}\left[\bar{s}^{\circ}\left(T_{cyl}\right) - \bar{R}\,\mathbf{ln}\left(\frac{y_i P_{cyl}}{P_0}\right)\right]_{pi}\right\} \tag{10}$$

Where the γ_{pi} is the molar fraction of each *i's* combustion products (e.g., CO_2, H_2O, N_2, O_2, CO) and other terms are similar of Eq. (4).

2.5 Combustion process

This engine stage starts before the cylinder arrives at the top dead center – *TDC* with fuel injection angle span – $\Delta\psi$ (rad) anticipated, where the cutoff ratio - r_c is calculated and defined in this study taking account, $V(\theta=\Delta\psi)$ the stroke volume (m^3) of fuel injection into the cylinder and V_c represents the dead volume (m^3) above of the *TDC* as Eq. 13. During the combustion process, the balance of entropy generation applied in this workspace is as follows:

$$\dot{S}_{gen,comb} = \left(\frac{dS_{vc}}{dt}\right)_{comb} + \left(\frac{\dot{Q}_{air} + \dot{Q}_{fuel} + \dot{Q}_w}{T_w}\right) + \dot{m}_{vc}\left(S_P - S_R\right) \geq 0 \tag{11}$$

The last term is related to the absolute entropy for the combustion products and reactants and its formulation is as follows in Eq. 11, All parameters within the equation are input or known. The sum of *i=1, n* are all the combustion chemical elements:

$$S_P - S_R = \sum_{i=1}^{n} \gamma_{pi}\left[\bar{s}^\circ\left(T_{cyl}\right) - \bar{R}\,\mathbf{ln}\left(\frac{y_i P_{cyl}}{P_0}\right)\right]_{pi} - \sum_{i=1}^{n} \gamma_{ri}\left[\bar{s}^\circ\left(T_{cyl}\right) - \bar{R}\,\mathbf{ln}\left(\frac{y_i P_{cyl}}{P_0}\right)\right]_{ri} \tag{12}$$

$$r_c = \frac{V\left(\theta = \Delta\psi\right)}{V_c} \tag{13}$$

2.6 CI-ICE outputs variables and post-processing

After the mathematical model convergency is achieved, the code creates an outputs variants list: temperature, pressure, volume and mass rates. In the sequence, the code starts immediately the post-processing analysis where the entropy rate and all variables in the Eq. 14 (first law thermal efficiency or BTE – brake thermal efficiency), Eq. 15 (second law thermal efficiency or exergy efficiency) and Eq. 16 (exergy destruction partial) are so calculated.

$$\eta_{ef} = \frac{\dot{W}_{ef}}{\dot{Q}_{comb}} \tag{14}$$

$$\eta_{ef\_II} = \frac{E_{net}}{E_{rev}} = \frac{E_{net}}{E_{ind} + E_{lost}} = \frac{\dot{W}_{ef}}{\dot{W}_{ind} + T_0 \dot{S}_{gen}} \tag{15}$$

$$E_d = \frac{E_{lost}}{E_{rev}} \tag{16}$$

Where $\dot{W}_{ef}$ is the effective power (W), $\dot{W}_{ind}$ is the indicated power (W), $\dot{Q}_{comb}$ is the heat of combustion (W); E_{net} is the net exergy, E_{rev} is the reversible exergy and E_d is the exergy destruction partial. Also, E_{lost} is the exergy lost (i.e., destroyed) as shows Eq. 17, where the $\dot{S}_{gen}$ represents the total sum of entropy generation rate (W. K^{-1}) during one engine cycle (i.e., $\theta=4\pi$).

$$E_{lost} = T_0 \dot{S}_{gen} = T_0 \int_0^{4\pi} \dot{S}_{gen,k}\, d\theta \tag{17}$$

3. RESULTS AND DISCUSSION

The mathematical model validated experimentally by Graciano et al. (2016) analyzes the indicated pressure, power torque and BSFC, with the first law approach only. The model is validated for the Lintec 4LD 2500 engine and MWM 229.6, but in this paper is considering just the Lintec 4LD 2500 engine, where your specification is: $CR = 16$ – compression ratio, $ROD/CS = 3.45$ – conrod-crank ratio, $B = 0.85$ – bore-stroke ratio, which $CS = 0.06$ (m), $ROD = 0.207$ (m), $P_d = 0.102$ (m) with 6 cylinder, $C_d = 0.9$, $V_c = 6.128x10^{-5}$ (m^3), and $r_c = 4.5$; for other information about it the manufacturer shall be consulted. In addition, this model has been validated experimentally with empirical correlation for friction losses and heat transfer through the cylinder wall. Figure 2 shows the *P x V* diagram simulated for one complete engine cycle, and the Fig. 3 shows the entropy generation for the same condition by time. These curves show the thermodynamic behavior and basic information related to this engine.

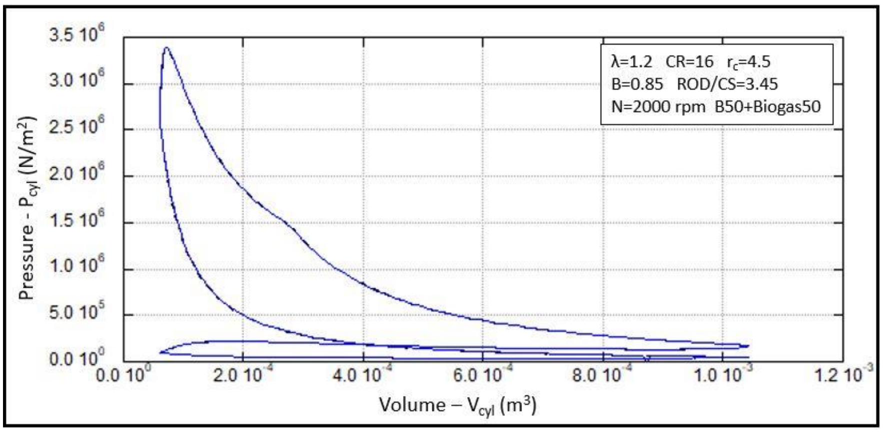

Figure 2. Simulation of P_{cyl} *x* V_{cyl} diagram of CI-ICE (4LD 2500).

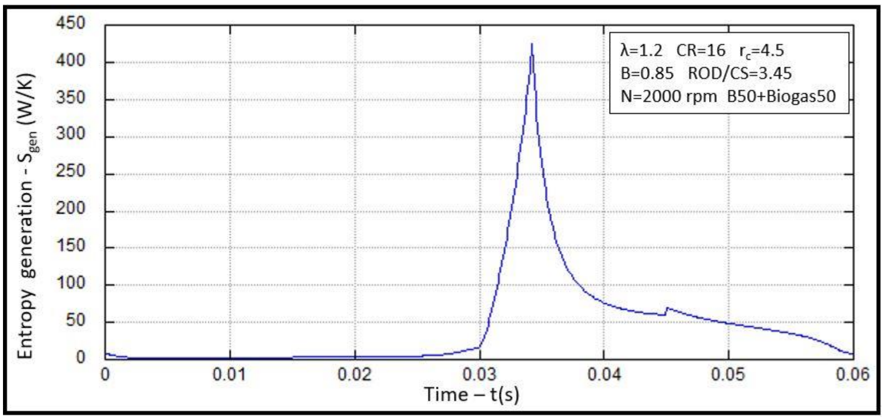

Figure 3. Simulation of $\dot{S}_{gen}$ during one complete engine cycle.

Figure 3 depicts the characteristic entropy curve for a diesel engine cycle. The time between 0 and 0.03 seconds concerns two first stroke stages of engine: admission and compression, after this instantaneous time the combustion process starts, where the entropy generation within the cylinder is quickly increased. After, the piston goes down (i.e., expansion stroke), and the last stage starts when the exhaust valve is opened and the combustion products gases go out of the cylinder: exhaust stroke.

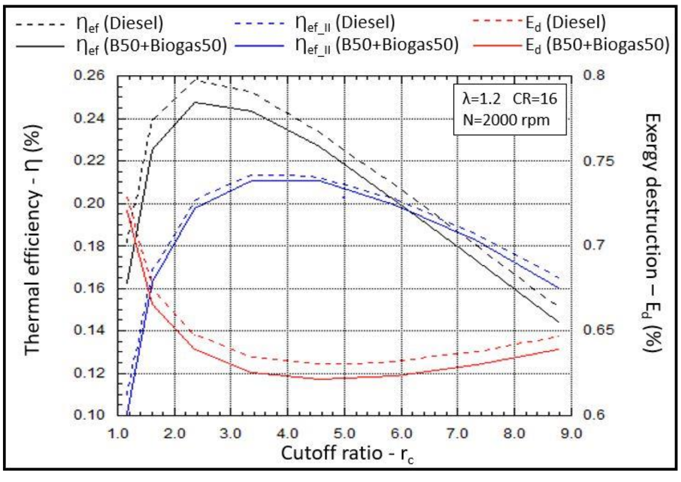

Figure 4. First and second law thermal efficiency and exergy destruction partial as a function of cutoff ratio – r_c.

Figure 4 bring up the impact of the cutoff ratio parameter on engine thermal efficiency and exergy destruction as well. It is possible to see very clear, there is the optimal cutoff ratio – $r_c = \sim 4.5$ (operating parameter) for the 4LD 2500 engine specification. It means a fuel injection angle – $\Delta\psi \sim 50°$ that can be checked on the Eq. 13.

The maximum performance is achieved to respect of cutoff ratio variation (i.e., constructal design). Therefore, the Fig. 4 has more information for this specific behavior. For example, the first law thermal efficiency - η_{ef} is maximum for $r_c \sim 2.2$ ($\Delta\psi \sim 30°$) and for second law thermal efficiency - η_{ef_II} is $r_c \sim 4.5$ ($\Delta\psi \sim 50°$). This lack between them is a very important point to understand that when there is the maximal thermal efficiency – η_{ef}, it doesn't mean the best configuration, physical or operational parameters for the exergy point of view (or constructal design). For example, in the real situation and application, this choice might convert at cost or weight saving. In addition, in this Fig. 4 is depicted the difference between the pure diesel and complete renewable alternative fuels. In this case and for all next figures, the simulation is done in this paper with 50% of biodiesel ($C_{20}H_{36}O_2$) + 50% of biogas (CH_4) = (B50+Biogas50). For r_c variation, the engine's thermal performance is slightly lower with (B50+Biogas50), because the fuel exergy of biodiesel is lower than that of diesel, but the exergy destruction partial – E_d is also lower (i.e., greater usage of fuel exergy).

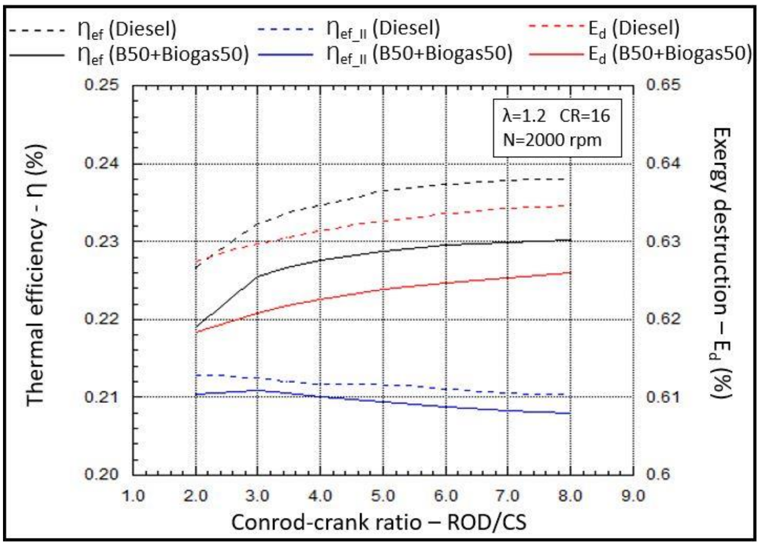

Figure 5. First and second law thermal efficiency and exergy destruction as a function of conrod-crank ratio – *ROD/CS*.

Figure 5 depicts the most famous physical construction of CI-ICE, the conrod-crank ratio mechanisms. In this paper is considered this ratio as *ROD* by *CS (*i.e., *ROD/CS)*. Norton (2004), discuss in his book how this mechanism works and its trade-offs as well. Also, there is a lower limit for this to work, which is $2.0 > ROD/CS$, and the upper limit is determined by kinematics and dynamics constraints. Unfortunately, in the Fig. 5 there is no optimal point for constructal design or exergy destruction partial, but there is advice for using the lowest possible relation $2.0 > ROD/CS > 4$. Obviously, this choice depends on and affect other mechanical parameters and specifications (e.g., kinematics, forces, etc.), but the result of this choice results in greater exergetic destruction or not.

A similar idea is shown in the Fig. 6 for another important physical relation for CI-ICE: bore-stroke – B, as Eq. 18, where the bore cylinder is considered equal to the piston diameter - P_d. These geometries are defined anyway, they depend on project and space constraints as discussed in the introduction to this paper. However, here is considered constructal design only, and the best definition is $2.0 > B > 3.0$ as it is possible to catch in the Fig. 6. It means a super-square engine type, Norton (2004). When $B = 1.0$ the engine type is known as square ratio ($P_d = 2CS$), and if $B < 1.0$ is called under-square and $B > 1.0$ over-square. Thus, for the exergy efficiency point of view the over-square engine type ($P_d > 2CS$) has the better structure or geometric configuration (i.e., constructal design). Norton (2004) says that there is the B range specification recommended and it is related with engine's dynamic performance, this range is $0.75 > B > 1.50$. In addition, the same idea for conrod-crank ratio, where the range specification recommended is about $3.0 > ROD/CS > 5.0$. However, for the conrod-crank ratio, the exergy analysis almost met with the project recommendation, maybe ratio ~ 3.0 looks a very good choice. On the other hand, the optimal B range simulated as Fig. 6 is over of recommendation, this the interesting point and opportunity to understand the constructal design approach as an input to engine design during its development.

$$B = \frac{P_d}{2CS} \tag{18}$$

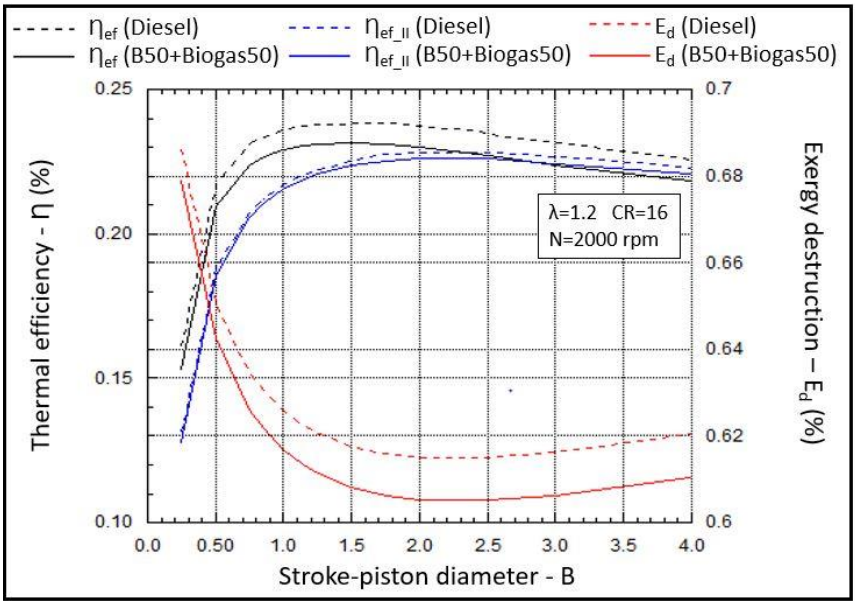

Figure 6. First and second law thermal efficiency and exergy destruction partial as a function of bore-stroke – *B*.

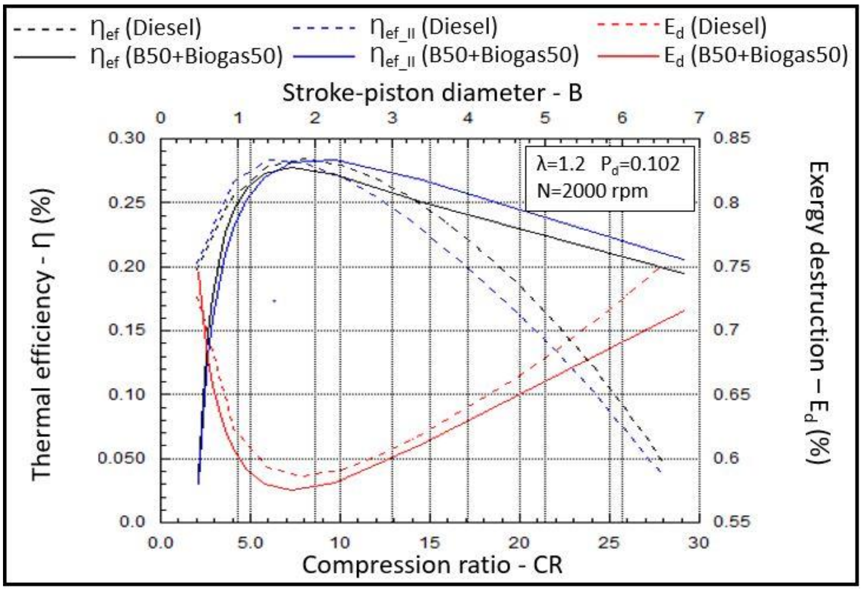

Figure 7. First and second law thermal efficiency and exergy destruction as a function of compression ratio – *CR*, keeping the P_d constant.

In the exergetic analysis on the Fig. 4, Fig. 5 and Fig. 6 the engine compression ratio is kept constant; it means the same stroke volume – V_d (m^3) independents on which parameter is changing. However, on the Fig. 7 and Fig. 8 is given freedom to V_d relating with *CR*. The mathematical model system needs to keep another input parameter fixed. On the Fig. 7 is demonstrated the *CR* behavior keeping the piston diameter – P_d constant. For the Fig. 8 is showed the *CR* behavior keeping the crank radius – *CS* constant. In addition, for this both exergetic analysis the $r_c = 4.5$ for diesel and (B50+Biogas50) fuel mode are also kept constant.

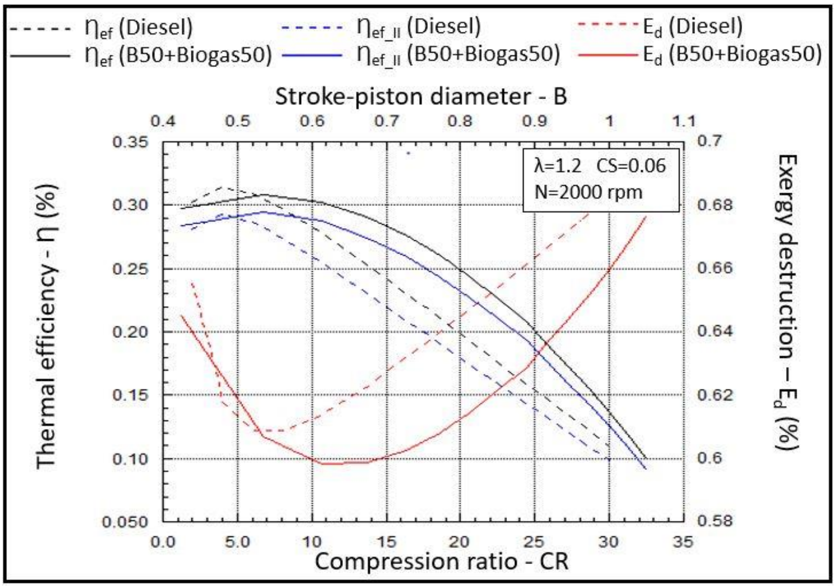

Figure 8. First and second law thermal efficiency and exergy destruction as a function of compression ratio – *CR*, keeping the *CS* constant.

The *CR* behavior is very interesting and intriguing because the diesel engine is known to be more efficient than spark engine due its high *CR*, and it is true, but the exergetic analysis shows what is the best *CR* range for each engine condition imposed (i.e., constructal design). In this paper, it is brought just two examples, Fig. 7 and Fig. 8, however, it is possible to image how many ways and freedom this system needs to deliver the optimal exergetic efficiency as the nature does in its shape and structure due to system flow resistances Bejan (2002).

4. CONCLUSION

This paper provides a general mathematical model for the CI-ICE driven by diesel, biodiesel and biogas mixtures able to estimate the best engine performance considering the physical constraints and environmental conditions. In this way, the engine design can be assessed with low computational time and accuracy that leads to a path to constructal design.

This study shows the maximum thermal performance of the engine through the first and second law of thermodynamics, subject to physical constraints and environmental conditions, indicating the path to thermodynamic optimization as a function of engine design, i.e. the application of constructal design approach to CI-ICE. The cutoff ratio parameter is optimized in certain physical dimensions as well as some mechanical relations, conrod-crank ratio, bore-stroke ratio and the compression ratio. In this paper, an optimal cutoff ratio is indicated ~4.5; although the conrod-crank ratio did not show an optimal point, only a recommended minimum, on the other hand, the bore-stroke ratio had an optimal point ~2.5. The most interesting result is the behavior of the compression ratio, when the stroke volume is given freedom, the optimal point is ~10 and not higher, as is the reputation of diesel engines. As can catch in this paper, a constructal design proves to be a very attractive path to drive an optimal engine design.

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