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ANALYSIS OF LAMBDA, ANGLE OF INJECTION AND SUBSTITUTION RATE IN HYDROCARBONS EMISSIONS IN DIESEL CYCLE ENGINE OPERATING IN DUAL FUEL MODE

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Abstract *In urban centers, the highest pollution rate comes from large engine vehicles and within this scenario, natural gas is an attractive option for use as fuel in Diesel cycle engines, since its application in this type of engine does not require major constructive changes. In terms of atmospheric emissions, it presents a clean burning because it has approximately 90% of methane in its composition, it reduces NOx emissions by around 27%, emissions by 77% and the emission of smoke is practically zero when used in Diesel cycle engines. In this research, engine-operating parameters related to air / fuel ratio, Diesel injection angle and Diesel / natural gas substitution rate were changed in Diesel engine operating in Dual fuel mode. Total hydrocarbon emissions in the Dual fuel mode will be evaluated, aiming to maintain the same efficiency that the engine presents using only Diesel as fuel. Engine tests were performed based on the European Stationary Cycle to define measure point. After that, it was defined the ESC B50 engine operation point and a design of the experiment was elaborated, to determine the best combinations of the proposed parameters. Engine essays were repeated with changes defined in design of the experiment methodology to analyze the emission total hydrocarbon changes.*

Keywords: *Dual fuel. Diesel. Hydrocarbon emissions. Natural gas. Lambda.*

1. INTRODUCTION

Currently there are themes that already show the immediate need research in order to find a better way to future demands, atmospheric emissions and energy efficiency are two topics included in this context, which require technological innovations. Natural gas applied to internal combustion engine of cycle Diesel presents a high thermal efficiency (Fredrik Konigsson (2011a,b)), turning it into a promising fuel alternative. It is also a commercial attraction for the relative ease of adapting to existing internal combustion engines. Natural gas has a 21% share (Fredrik Konigsson (2012)) in the world energy matrix, presents in its composition approximately 90% of methane, which makes it a clean burning fuel. As it reduces around of 27% NOx emissions, 77% emissions of particulate matter and the emission of smoke is practically zero when used in Diesel cycle engine (Fredrik Konigsson (2011a); Karin (2015)).

This Dual fuel technology used, which uses a small amount of diesel to combustion and natural gas to complete it, shows great potential for reduce fuel consumption because it achieves up to 90% Diesel substitution rate for natural gas while maintaining the same levels of engine efficiency when using Diesel only as fuel (DORFER. (2013)). One factor that needs to be developed is the emission of hydrocarbons, which is directly linked to greenhouse gases. This research aims to reduce the emission of hydrocarbons in a Diesel cycle engine in dual fuel mode (Ryu (2013)), being a differential in relation to other studies that focus only on NOx and CO emissions.

2. EXPERIMENTAL PROCEDURE

2.1 EXPERIMENTAL APPARATTUS

The sample used for this study is Diesel cycle internal combustion engine an inline six-cylinder (Figure 1), it has 9.7 liters of displacement. A central natural gas injection system was fitted with six injectors installed in a distribution block (Figure 2) interconnected to the combustion air intake duct of the engine. A throttle valve was adapted to the engine air inlet to control air + natural gas flow (Amin Yousefi (2015)).

Figure 1. Engine dualfuel assembled at Dynamometer

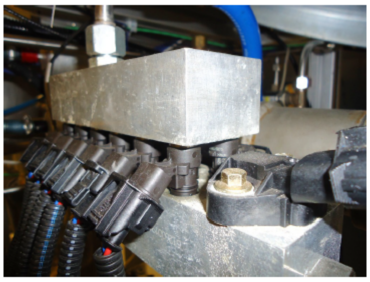

Figure 2. Natural gas distribution block

3. METHODOLOGY

A power curve in Diesel mode was performed in Dynamometer to define measure point using the European Stationary Cycle methodology (Figure 3) whose steps faithfully represent various engine operating conditions, both on road and in urban cycles. Checkpoints (A, B or C) were calculated from the power curve and in this search the point B50 was defined as check point for efficiency and emissions measurements. In this condition air / fuel ratio (Lambda), Diesel injection angle (SOI) and Diesel / natural gas substitution rate will be changed to effects on the efficiency and emissions of the total hydrocarbons of the engine (S. Imran (2014)).

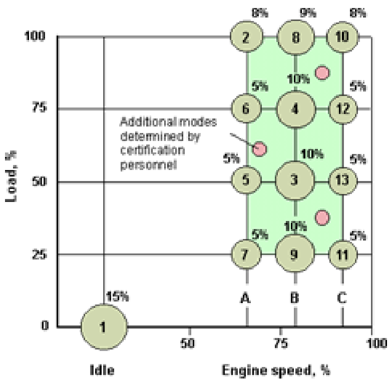

Figure 3. European Stationary Cycle

Table 1. *European Stacionary Cycle* (adapted from *DIESELNET*, 2016)

Step	Engine speed	Load(%)	Weight(%)	Time(min)
1	Idle speed	0	15	4
2	A	100	8	2
3	B	50	10	2
4	B	75	10	2
5	A	50	5	2
6	A	75	5	2
7	A	25	5	2
8	B	100	9	2
9	B	25	10	2
10	C	100	8	2
11	C	25	5	2
12	C	75	5	2
13	C	50	5	2

3.1 CHARACTERISTICS AND PARAMETERS FOR EMISSIONS ANALYSIS

3.2 AIR AND FUEL RATE IN DUAL-FUEL MODE (λ)

Formula of equivalence of air and fuel is given by the equation below:

$$\lambda = \frac{\dot{m}_{ar}}{\dot{m}_{NG} AFR_{NG} + \dot{m}_D AFR_D} \tag{1}$$

The ratio of air and fuel is calculated by dividing the mass flow of combustion air by multiplying the mass of air of fuel by the ratio of air and fuel. In this case dual-fuel (Diesel and natural gas).

3.2.1 BRAKE MEAN EFFECTIVE PRESSURE (BMEP)

Another way to measure performance is the average brake mean effective pressure, obtained by dividing the work, by cycle, by the volume displaced per cycle (Heywood (1988)).

$$BMEP = \frac{Tnr}{Vd} \tag{2}$$

Where T is the torque measured in Nm, nr is engine speed and Vd is the volume displaced inside the combustion chamber in the cycle.

3.2.2 THERMAL EFFICIENCY (η_t)

Thermal efficiency is the work done on the cycle and the product of the fuel consumed multiplied by the lower calorific value of the fuel.

$$\eta_t = \frac{\dot{\omega}}{\dot{m}_{NG} PCI_{NG} + \dot{m}_D PCI_D} \tag{3}$$

Where $\dot{\omega}$ is power, $\dot{m}_D$ is Diesel mass flow and $\dot{m}_{NG}$ is the natural gas flow. PCI_{NG} is the natural gas lower calorific value and PCI_D is the Diesel lower calorific value

3.2.3 HEAT RELEASE RATE (RoHR)

$$RoHR = 1 - EXP\left[-a\left(\frac{\theta - \theta_s}{\theta_d}\right)^{mv}\right] \tag{4}$$

Where the coefficient a is Wiebe efficiency factor, θ is crankshaft angle, θ_s is the begin of combustion angle, θ_d is the duration of combustion angle and mv is the Wiebe form factor.

3.3 THEORICAL CALCULATION OF THE CHEMICAL KINETICS OF EMISSIONS

This sub-chapter is intended to demonstrate the calculations related to kinetics chemistry, in order to analyze the factors that influence the chemical reaction that occurs during combustion (Khalil (1998)). The equilibrium concentration and the kinetic time scale are expressed by the formula:

$$Yi^{n+1} - Yi^{n} = \omega idt = \frac{\tau ki_n}{\tau ki_n + f\tau_{turb}}\Delta Yi \tag{5}$$

Where i, Yi is the current concentration, Yi^* is the equilibrium of concentration. τki_n is the kinetic time scale and τ_{turb} is the swirling kinetic time scale. It is calculated by the formula (6)

$$\tau_{turb} = \frac{C2K}{\varepsilon} \tag{6}$$

Where C2 is the model constant, K the turbulent energy of the model and ε is the dissipation rate.

The mathematical expression of the chemical kinetics of elementary reactions is given by the formula:

$$\sum_{i=1}^{n} \alpha_{ijf} Xi \rightleftharpoons^{Kjf}_{Kjb} \sum_{i=1}^{n} \alpha_{ijb} Xi \tag{7}$$

Where α_{ijf} and α_{ijb} are the stoichiometric coefficient of the resulting species and the product of the reaction j. Xi is the chemical symbol of species i part of the reaction. n is the total number of chemical species involved in the reaction. j is the total number of chemical reactions.

The rate of formation of species i is a function of all the steps of reactions involved, as shown below:

$$-\frac{dC1}{dt} = [\frac{1}{\rho}]\sum_{j}^{j=1}(\alpha_{ijf} - \alpha_{ijb})(R_{jf} - R_{jb}) \tag{8}$$

Where C1 is the concentration of species i in Mol / kg of the mixture. Rf and Rb are the forward and backward reaction rate of the reaction j. ρ is the specific mass of the mixture in $\frac{kg}{m^3}$ and t is the time in seconds.

The reaction rates Rf and Rb are calculated by the expressions:

$$Rjf = K_{jf}\prod_{n}^{i=1}(PCi)^{\alpha}_{ijf} \tag{9}$$

$$Rjb = K_{jb}\prod_{n}^{i=1}(PCi)^{\alpha}_{ijb} \tag{10}$$

Where K jf and Kbjb are constants of the reaction rate, n is the total order of the reaction and B is the constant of the temperature exponent.

Constants of the Arhenius equation are given by:

$$K_{jf} = AjfT^{B}jfexp[\frac{-Ejf}{RgT}] \tag{11}$$

$$K_{jb} = AjbT^{B}jbexp[\frac{-Ejb}{RgT}] \tag{12}$$

Where Ajf and Ajb are pre exponential factors. Bjf and Bjb are constants of temperature exponent. Ejf and Ejb are the activation energy, Rg is the universal gas constant and T is the temperature in Kelvin.

3.4 DESIGN OF THE EXPERIMENT

In order to delineate the experiments required for this research, we used a closed-code software called ASCMO developed by the company ETAS, which uses a statistical learning algorithm based on Gaussian process modeling, which creates a motor model at some specific measurements. With these measurements it automatically determines the model of the set N of base D and weight r that represents the formation of data with the maximum likelihood.

The basic principle used by the software consists of a superposition of functions to describe the output of the system in $y(\overrightarrow{x})$, according to the dimensional D parameter of the vector $\overrightarrow{x}$:

$$y(\overrightarrow{x}) = \sum_{i=1}^{N} C_i.e^{-\frac{1}{2}\sum_{l=1}^{D}\frac{(X_{il}-x_l)^2}{r_l^2}} \tag{13}$$

After verifying the accuracy of the created model, the experiment is delineated. In this research the sequence construction was used using the Sobol method, which is considered a characteristic, to generate a sequence of values x1, x2 ...with these values between 0 <xi <1 (Santner (2003))

Following builds a set of direction numbers v_1, v_2. Each v_i is a binary fraction, expressed as follows:

$$v_i = \frac{m_i}{2^i} \tag{14}$$

Where m_i is an odd integer such that $0 < m_i < 2^i$

For the construction of the sequence *Quasi-Monte*, a primitive polynomial was chosen where each ai is 0 or 1 and P is an arbitrary primitive polynomial:

$$P = x^u + a_1 x^{u-1} + ... + a_u - 1^{x+1} \tag{15}$$

4. RESULTS

4.1 DIESEL POWER CURVE

The performance curve in Diesel mode (Figure 4) follows the guidelines described in the standard NBR ISO 1585 of 06/1996, which determines the test conditions in the engine, such as the temperature range of the combustion air. The torque and power values measured in the Dynamometer during the performance curve allowed defining the partial load points by means of the "European Stationary Cycle" method. Point B50 was chosen as the basis for the parameter variation tests because it is a medium partial load condition and commonly used for comparison of emissions between internal combustion engines with different characteristics.

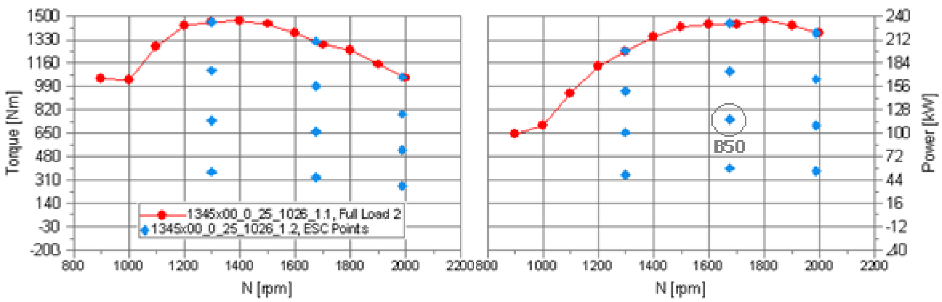

Figure 4. Diesel power curve and B50 point

4.2 EMISSIONS OF HYDROCARBONS

At this stage of the research the engine was tested in the dynamometer at a constant condition of rotation at 1676 rpm and 660 Nm of torque, which corresponds to point B50. The Diesel distribution gallery for the injectors was at a constant pressure of 1600 bar. The replacement rate set for this step was 60 %, the start of the Diesel injection at 5^0 CA and the lambda at 1,50. Following, the parameters of injection start, lambda and substitution rate were changed one at a time, keeping the other parameters in the same conditions described above

4.3 START OF INJECTION VARIATION

By means of software that controls the electronic module of the engine, the beginning of the injection of Diesel was changed from 1^0 to 5^0 CA with steps of 1^0CA. At each step, a measurement of the main parameters was carried out. In this way, it was possible to observe that as the injection angle was increased, the efficiency showed an improvement around 4% (Figure 5), however nitrogen oxide increased by about 50% (Figure 6). Emissions of carbon dioxide, particulate matter and hydrocarbons remained stable.

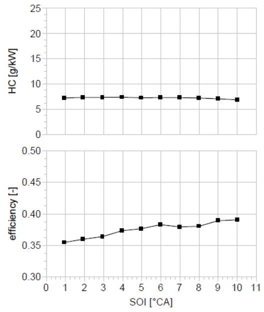

Figure 5. Efficiency and hydrocarbon emissions varying the start of Diesel injection

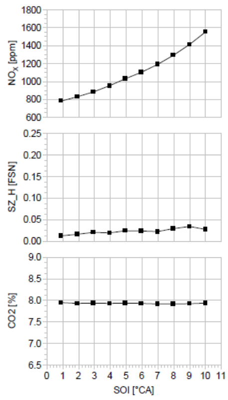

Figure 6. Emissions of nitrogen oxide and carbon dioxide and particulate matter varying the start of Diesel injection

4.4 LAMBDA VARIATION

At this stage the lambda was changed from 1,15 to 1,70 in steps of 0,05. This variation was performed by changing the opening time of the Diesel injectors through the injection system control software. The best efficiency was seen with lambda at 1,35 (Figure 7). As the lambda increased the specific emissions of hydrocarbons increased considerably reaching the value of 15 g/kWh with lambda at 1,70 (Figure 8).

On the other hand the emissions of nitrogen oxide and carbon dioxide were drastically reduced with increasing lambda.

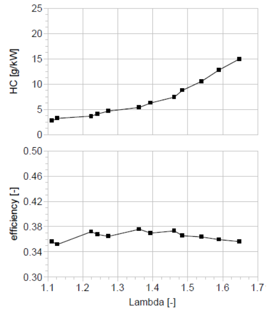

Figure 7. Efficiency and emissions of hydrocarbons varying the lambda

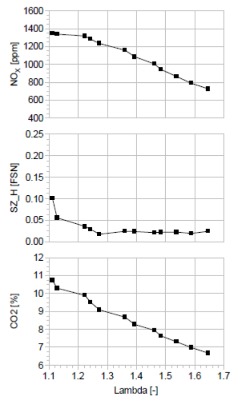

Figure 8. Emissions of nitrogen oxide and carbon dioxide and particulate matter varying the lambda

4.5 VARIATION OF THE SUBSTITUTION OF DIESEL BY NATURAL GAS

This step was performed in the same way as the two previous stages, changing only the rate of substitution of Diesel for natural gas. The measurement points were 1%, 12%, 24%, 48%, 55%, 66%, 71%, 77%, 83%, 84%, 88%. In this condition the best efficiency obtained was 38% with 83% substitution rate (Figure 9). The specific emissions of hydrocarbons remained stable at around 7,5 g / kWh in the range of 55% and 91% of substitution (Figure 10). In the range of 48% to 77%, there was a 33% increase in nitrogen oxide emissions, while carbon dioxide emissions were reduced by 25% in the same range.

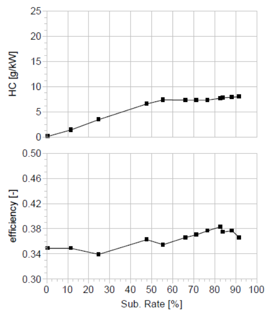

Figure 9. Efficiency and hydrocarbon emissions varying the rate of substitution

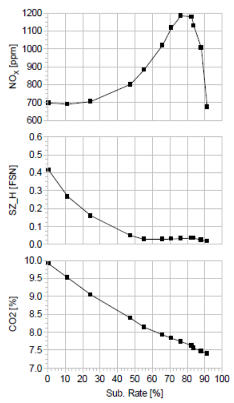

Figure 10. Emissions of nitrogen oxide and carbon dioxide and particulate matter varying the rate of substitution

4.6 ENGINE MATHEMATICAL MODEL BY ASCMO SOFTWARE

With the results measured in the Dynamometer, a computational modeling was carried out through ASCMO software and then a comparison between this model and the data measured in Dynamometer for the emissions of nitrogen oxide (Figures 11 and 13) and hydrocarbons (Figures 12 and 14), varying the start of Diesel injection and lambda.

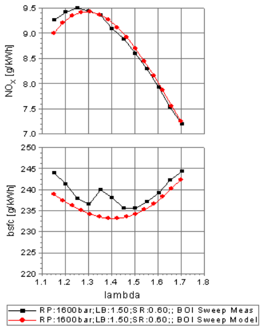

Figure 11. NOx comparison between the measured data and the mathematical model, varying Lambda

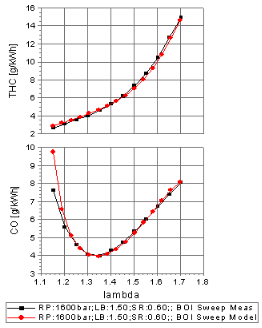

Figure 12. HC comparison between the measured data and the mathematical model, varying Lambda

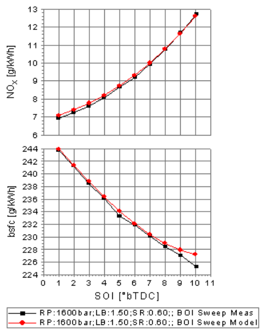

Figure 13. NOx comparison between the measured data and the mathematical model, varying the beginning of the Diesel injection

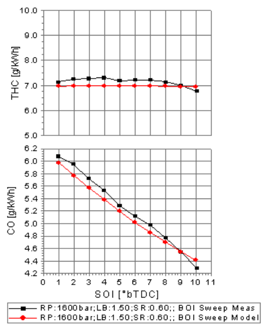

Figure 14. HC comparison between the measured data and the mathematical model, varying the beginning of the Diesel injection

The comparison showed a good precision in the mathematical model for the emissions of nitrogen oxide, carbon dioxide and hydrocarbons, as shown in the graphs.

4.7 CHEMICAL KINETICS OF EMISSIONS

The code to calculate is being prepared

4.8 DESIGN OF THE EXPERIMENT

With the confirmation of the data seen in the previous chapter, it was possible to use the mathematical model to generate the experiment's outline following the Space-Filling Design (Sobol sequence) method to obtain the best combination between lambda, Diesel and the replacement rate for the hydrocarbon emissions referring to point B50.

Table 2. Design of the experiment results

X	**Lambda**	**SOI**	**Z**	η_t	**HC**
1	1,70	3	0,60	0,38	15,39
2	1,35	10	0,60	0,42	4,80
3	1,40	3	0,60	0,40	5,24
4	1,55	6	0,60	0,41	8,74
5	1,20	8	0,65	0,41	3,58
6	1,55	2	0,65	0,40	8,48
7	1,25	6	0,65	0,41	3,94
8	1,50	5	0,65	0,41	7,22
9	1,15	9	0,65	0,41	3,03
10	2,67	-1,85	0		
11	1,60	10	0,70	0,43	9,63
12	1,40	5	0,70	0,42	5,68
13	1,65	2	0,70	0,39	14,40
14	1,40	9	0,70	0,43	5,89
15	1,60	5	0,70	0,41	10,55
16	1,65	7	0,70	0,41	12,41
17	1,30	3	0,70	0,41	4,36
18	1,50	3	0,75	0,42	7,62
19	1,25	7	0,75	0,42	4,36
20	1,20	5	0,75	0,42	3,49
21	1,50	9	0,75	0,43	7,92
22	1,55	5	0,75	0,42	9,24
23	1,20	6	0,75	0,42	3,68
24	1,25	9	0,75	0,43	4,26
25	1,45	4	0,75	0,42	6,66
26	**1,35**	**3**	**0,80**	**0,43**	**5,47**
27	1,65	9	0,80	0,43	12,83
28	1,35	7	0,80	0,43	5,61
29	1,50	5	0,60	0,41	7,19
30	1,45	8	0,65	0,42	6,35

5. TESTING OF THE OPTIMUM POINT

At this stage, the tests were carried out to verify the best condition to keep the hydrocarbon emissions at the lowest possible level, the best efficiency with the highest replacement rate, under the conditions determined in the experiment design point X 26.

In the first assay the injection was maintained at 3^0 CA, Lambda at 1,35 and the replacement rate was changed to 80%, 82% and 85% (Figure 15). This test showed that the substitution rate does not interfere with HC emissions, since they remained stable, but reduced efficiency by 1%, remaining at 37,2% when the rate of increase from 80% to 85% was increased. Hydrocarbon emissions remain stable at around 5,47 g/kWh.

On the other hand, changing the injection start angle from 2^0 to 5^0 CA increases efficiency by 2% reaching 38,4% and hydrocarbon emissions remain at the level of 5,59 g/kWh.

In the second assay, the start of injection with 3^0 CA and the replacement rate was maintained at 85%, and the Lambda was varied with 1,30, 1,46 and 1,50 respectively (Figure 16). In the first step by changing the lambda from 1,30 to 1,46 the efficiency remained at 37,5% but the hydrocarbon emissions increase from 4,81 g/kWh to 7,63 g/kWh which is equivalent

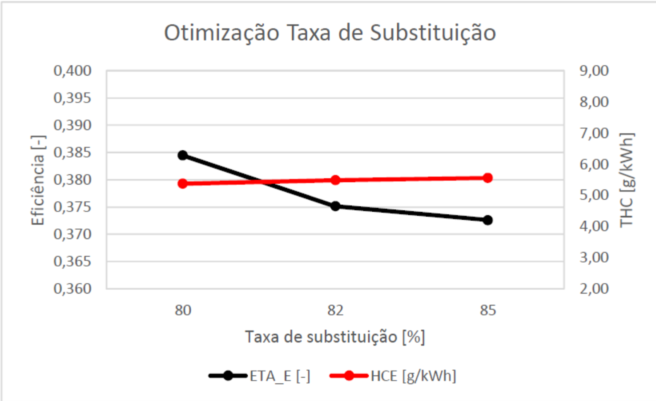

Figure 15. Checking the emissions of HC at the optimum point, varying the substitution rate

to an increase of 37,5%. In this step of this test it was clear that Lambda is the characteristic that has the greatest influence on the emissions of hydrocarbon and the efficiency among the three characteristics that are part of this study.

When the Lambda value was set at 1,50, hydrocarbon emissions fell by 16% compared to the previous step and efficiency increased considerably by approximately 2%, reaching 39,4%.

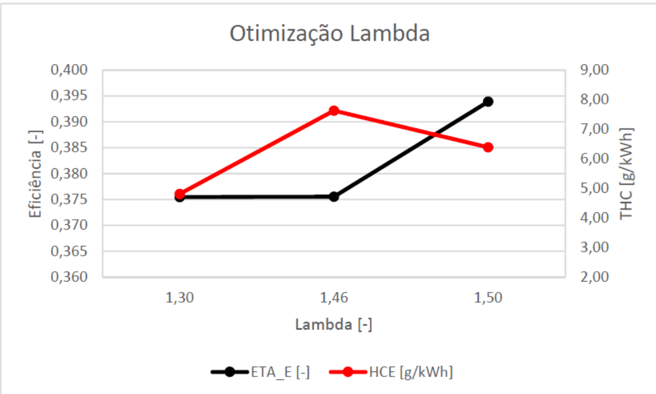

Figure 16. Checking the emissions of HC at the optimum point, varying the Lambda

In the latter assay, the Lambda was maintained at 1,35, the replacement rate was 85%, and the injection start variation was made with the values of 2^0, 4^0 and 5^0 CA (Figure 17). Under these test conditions hydrocarbo emissions remained at a level of 5,5 g/kWh, efficiency increased by about 5% when the start of Diesel injection was changed from 2^0 to 5^0.

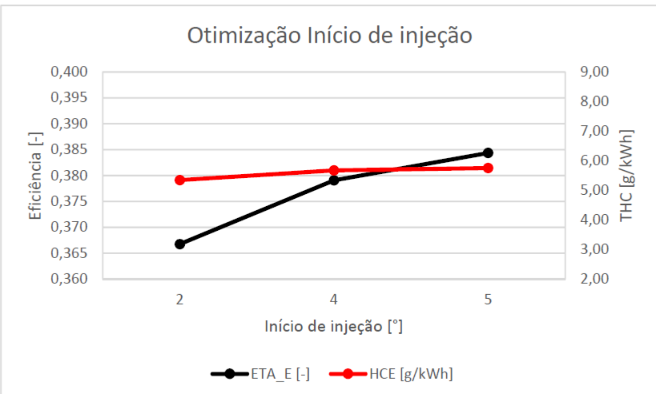

Figure 17. Checking the emissions of HC at the optimum point, varying the start of Diesel injection

6. CONCLUSION AND FINAL CONSIDERATIONS

This research presented results for a better understanding of the influence of the lambda, injection start angle and substitution rate in hydrocarbon emissions and efficiency in a Diesel cycle engine, operating in dualfuel mode. Was adapted a central natural gas injection system in a 9,7 liter Diesel cycle engine, transforming it into dualfuel. This work was carried out in the laboratory of Robert Bosch, located in the city of Curitiba, laboratory consisting of equipment state-of-the-art technologies that have enabled advanced combustion the tests.

Initially a performance curve was evaluated using only Diesel as fuel to verify that the engine would achieve the approved torque and power by the manufacturer and with this test one can calculate the point B50 (1676 rpm and 660 Nm), following the European Stationary Cycle, shortly after a scan was performed by operating the motor in Dualfuel mode and systematically the individual contribution of the lambda, the angle of injection onset, and the hydrocarbon emissions and engine efficiency to identify trends in emissions seen in the literature.

The creation of a mathematical model using ASCMO software and the comparison of this model with the values in Dynamometer demonstrated a good accuracy between both, a fact that contributed to the elaboration of an experimental design following the Space-Filling Design - Sobol sequence methodology, where it was possible to define the contour condition for a refinement in the variation of the study parameters. The tests performed during the refinement showed that an increase in the replacement rate the efficiency is reduced and the hydrocarbon emissions remain stable. On the other hand, by changing the angle of injection efficiency ratio increases considerably and hydrocarbon stable.

Finally in the first step in the lambda variation efficiency remains hydrocarbon emissions increase considerably, already in the second efficiency step was increased and hydrocarbon emissions reduced. In this way, these experiments lead us to conclude that among the studied parameters, which has the greatest influence on hydrocarbon emissions and the efficiency in a Dualfuel engine is the lambda.

7. REFERENCES

Amin Yousefi, Madjid Birouk, B.L.A.G., 2015. "Performance and emissions of a dual-fuel pilot diesel ignition engine operating on various premixed fuels". *Energy Conversion and Management*. 43.

DORFER., D.C.M.P.D.R.K.M.F., 2013. "Dual fuel-potential of combined combustion of cng and diesel fuel". *SAE International*. 31.

Fredrik Konigsson, Per Stalhammar, H.E.A., 2011a. "Characterization and potential of dual fuel combustion in a modern diesel engine". *SAE International*. 10.

Fredrik Konigsson, Per Stalhammar, H.E.A., 2011b. "Combustion modes in a diesel-cng dual fuel engine". *SAE International*. 11.

Fredrik Konigsson, Per Stalhammar, H.E.A., 2012. "Controlling the injector tip temperature in a diesel dual fuel engine". *SAE International*. 09.

Heywood, J.B., 1988. *Internal Combustion Engine Fundamentals*. 02.

Karin, G.A., 2015. *Dual-Fuel Diesel Engines*. CRC press. 03.

Khalil, E.B.F., 1998. *MODELLING THE CHEMICAL KINETICS OF COMBUSTION OF HIGHER HYDROCARBON FUELS IN AIR*. Master's thesis, Calgary, Alberta. 58.

Ryu, K., 2013. "Effects of pilot injection timing on the combustion and emissions characteristics in a diesel engine using biodiesel-cng dual fuel". *Applied Energy*. 34.

S. Imran, D.R. Emberson, B.I.D.W.R.C.T.K., 2014. "Effect of pilot fuel quantity and type on performance and emissions of natural gas and hydrogen based combustion in a compression ignition engine". *ScienceDirect*. 32.

Santner, T.J., 2003. *The Design and Analysis of Computer Experiments*. 01.

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