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THE EFFECT OF BROWN GAS ADDITION IN THE DIESEL ENGINE FUEL COMBUSTION PROCESS

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Abstract: Fuel combustion is the most important moment in the operation of a thermal engine. The efficiency of combustion depends on the efficiency of the engine and the level of pollution produced. A very important element of combustion is the diffusion of fuel into the air mass sucked into the engine, which depends on the type and size of the fuel droplets. Another important element is the speed of flame propagation in the mass of air/fuel mixture. The addition of Brown Gas benefits the quality of fuel combustion because the hydrogen in the mixture has a diffusion velocity 7–8 times higher than gasoline which favours a rapid flame propagation in the fuel mixture volume. The high flame propagation speed in the air/fuel mixture favours better fuel combustion with improved engine operating parameters. The parameters analysed are the net heat release rate, the volume fraction burned in the cylinder and the temperatures in the cylinder during combustion.
Keywords: diesel engine, Brown gas, efficiency, pollution

1. TOPICALITY OF THE THEME

Importantly, over time, compression ignition engine manufacturing technology has had two major advantages: high reliability and low production costs. This contributes to a greater or lesser extent to supporting the use of compression ignition engines but in combination with other types of engines in alternative propulsion systems.

The interest in increasing the performance of compression ignition engines used in transport stems from scientific research into the techniques used in design and manufacture, in identifying ways of making maximum use of energy for the benefit of competitiveness and reducing pollution.

2. RESEARCH METHODOLOGY: AIM, OBJECTIVES AND HYPOTHESES

The aim of the research is to study, experiment and substantiate from a theoretical and practical point of view the operation of the diesel engine with a mixture of diesel and Brown gas. Vehicles with thermal engines have a future and vehicles with compression–ignition engines are constantly undergoing further development to improve their performance.

There have been numerous studies and research aimed at introducing hydrogen into the fuel mixture of diesel engines. The conclusions obtained are diverse and in some cases divergent in terms of the performance of the diesel and hydrogen fuelled engine. The assumption underlying our research is that the addition of HHO to the fuel mixture improves engine operating conditions.

In order to verify this working hypothesis we have run two simulations of diesel engine operation using the AVL BOOSTTM software. The first simulation will be with conventional fuel, diesel, to have a point of comparison and starting and the second simulation with a mixture of diesel and Brown gas. The two simulations will result in a series of data that will be analysed and compared to see if the research hypotheses are confirmed.

3. PRESENTATION OF THE PHYSICAL–MATHEMATICAL MODEL

For a better understanding of the AVL BOOST™ program, the theoretical background [10] includes basic equations for all available elements and does not include detailed explanations for all engine cycle simulations. [3]

BOOST™ calculates gas properties that depend on temperature, pressure and gas composition. Gas properties, such as the gas constant or heat capacities of a gas, depend on temperature, pressure and gas composition. BOOST™ calculates gas properties at each instant, at each time step, with instantaneous composition.

Compression, expansion and evacuation processes and heat release rate

The MCC (Mixing Controlled Combustion) model [7], [8] is used to predict combustion characteristics in direct injection compression ignition engines.

The model considers the effects of premixing controlled combustion (PMC) and diffusion controlled combustion (MCC) processes as a function of:

$$\frac{dQ_{total}}{d\alpha}=\frac{dQ_{MCC}}{d\alpha}+\frac{dQ_{PMC}}{d\alpha}$$

Fuel mixture combustion control:

In this regime, heat release is a function of the amount of fuel available (f1) and the turbulent kinetic energy density (f2):

$$\frac{dQ_{MCC}}{d\alpha}=C_{Comb}\cdot f_1(m_F,Q_{MCC})\cdot f_2(k,V) \tag{1}$$

with f_1

$$f_1(m_F,Q)=\left(m_F-\frac{Q_{MCC}}{LCV}\right)\cdot(w_{Oxygen,available})^{C_{EGR}} \tag{2}$$

and f_2

$$f_2(k,V)=C_{Rate}\cdot\frac{\sqrt{k}}{\sqrt[3]{V}} \tag{3}$$

Q_{MCC} – cumulative heat release for controlled combustion of the mixture [kJ].
C_{EGR} – combustion constant [kJ/kg/deg CA]
C_{Rate} – mixing speed constant [s]
k – local turbulent kinetic energy density [m2/s2]
m_F – mass of fuel vaporised (real) [kg]
LCV – lower calorific value [kJ/kg]
V – cylinder volume [m3]
α – crank angle [deg CA]
$w_{Oxygen,available}$ – mass fraction of available oxygen (aspirated and in EGR) at SOI [–]
C_{EGR} – Constant of influence EGR [–]

Ignition delay model

Ignition delay is calculated using the Andree and Pachernegg model [2] by solving the following differential equation:

$$\frac{dI_{id}}{d\alpha}=\frac{T_{UB}-T_{ref}}{f_{id}\cdot Q_{ref}} \tag{4}$$

As soon as the integral of the ignition delay calculation I_{id} reaches the value 1,0 (=α_{id}) ignition delay τ_{iD} is calculated from $\tau_{id}=\alpha_{id}-\alpha_{SOI}$

I_{id} – delay to full ignition [–]
T_{ref} – reference temperature = 505,0 [K]
T_{UB} – unburnt area temperature [K]
Q_{ref} – reference activation energy, f (droplet diameter, oxygen content, ...) [K]
τ_{id} – ignition delay [s]
α_{SOI} – injection start time [degCA]
α_{id} – ignition delay timing [degCA]
f_{id} – ignition delay calibration factor [–]

Premixed combustion model:

A Vibe function is used to describe the actual heat release due to premixed combustion:

$$\frac{\left(\frac{dQ_{PMC}}{Q_{PMC}}\right)}{d\alpha}=\frac{a}{\Delta\alpha_c}\cdot(m+1)\cdot y^m\cdot e^{-a\cdot y^{(m+1)}} \tag{5}$$

$$y=\frac{\alpha-\alpha_{id}}{\Delta\alpha_c} \tag{6}$$

Q_{PMC} – total fuel heat input for premixed combustion = $m_{fuel,id}\cdot C_{PMC}$
$m_{fuel,id}$ – the total amount of fuel injected during the ignition delay phase
C_{PMC} – premixed combustion parameter [–]
$\Delta\alpha_c$ – duration of premixed combustion = $\tau_{id}\cdot C_{PMC-Dur}$
$C_{PMC-Dur}$ – premixed combustion duration factor
m – shape parameter m = 2,0
a – Vibration parameter a = 6,9

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Droplet heating and evaporation pattern

According to Sitkei [12], the equilibrium temperature for droplet evaporation can be calculated iteratively from:

$$\lambda_c \cdot (T_c - T_d) = \frac{30.93 \cdot 10^4 \cdot \frac{T_d}{p_c}}{e^{\left(\frac{4150.0}{T_d}\right)}} \cdot (20.0 + 0.26 \cdot (T_d - 273.15) + 0.3 \cdot (T_c - 273.15)) \quad (7)$$

Using the equilibrium temperature, the evaporation rate can be calculated with:

$$v_e = 0.70353 \cdot \frac{T_d}{p_c \cdot e^{\left(\frac{4159.0}{T_d}\right)}} \quad (8)$$

The value of 0.70353 can be changed by user input as a user parameter. Finally, the change in droplet diameter (and corresponding change in droplet mass) over time can be calculated:

$$d_d = \sqrt{d_{d,0}^2 - v_e \cdot t} \quad (9)$$

λ_c – thermal conductivity of the cylinder [W/ms]
T_c – temperature in the cylinder [K]
T_d – equilibrium temperature of isothermal droplet evaporation [K]
p_c – cylinder pressure [Pa]
v_e – evaporation rate [m2/s]
d_d – effective droplet diameter [m]
$d_{d,0}$ – initial droplet diameter [m]

4. GENERAL CHARACTERISTICS

For the modelling of the theoretical experimental processes and the running of the Brown gas simulation we have chosen as model a diesel engine whose characteristics are (partially) given below.

General informations [13]

- Brand – Volkswagen
- Engine type – 1.9 TDI

Performance

- Urban fuel consumption – 8.1 l/100 km
- Extra–urban fuel consumption – 5.2 l/100 km (19.23 km/l)
- Mixed fuel consumption – 6.3 l/100 km (15.87 km/l)
- Fuel type – Diesel

Engine

- Power : 130 CP @ 4000 rot/min
- Couple: 310 Nm @ 1900 rot/min
- Engine volume – 1896 cm3
- Number of cylinders – 4
- Cylinder diameter – 79.5 mm
- Cylinder stroke – 95.5 mm
- Compression ratio – 19
- Number of valves per cylinder – 2
- Engine Intake – Turbocharger, Intercooler

The burning process

The operation of the heat engine is based on the combustion of a fuel mixture to obtain a quantity of heat that is converted into mechanical work. For a short time, a few degrees of crank rotation, a maximum heat release of 6.06 J/degree at 800 rpm can be achieved. Figure 2.

Table 1. Mass of fuel mixture in cylinders (diesel)

	Mass Cylinder 800 rpm (g)	Mass Cylinder 1600 rpm (g)	Mass Cylinder 2400 rpm (g)	Mass Cylinder 3200 rpm (g)
Mass A+F	1.68076	0.850387	0.56933	0.427927

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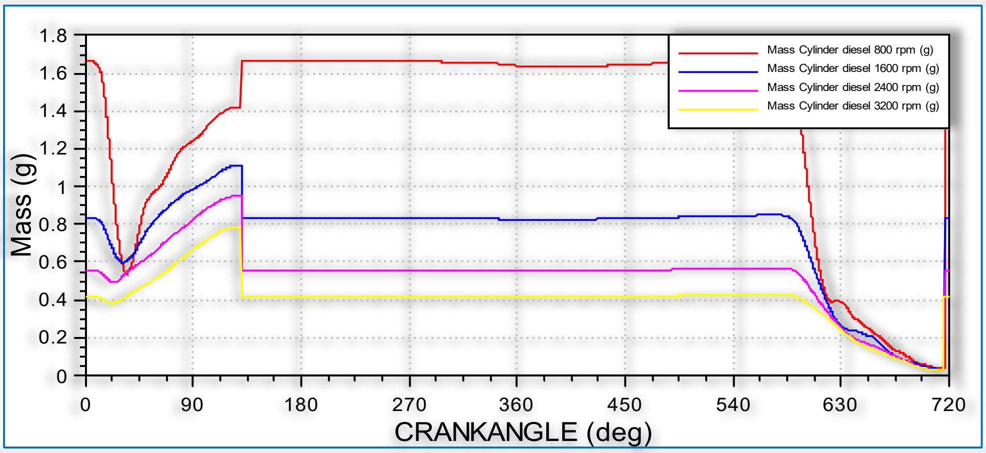

Figure 1. Mass of fuel mixture in cylinders (diesel). Source: own construction

Table 2. Net heat release rate (diesel)

Net Rate of Heat Release Cylinder 800 rpm (J/deg)	Net Rate of Heat Release Cylinder 1600 rpm (J/deg)	Net Rate of Heat Release Cylinder 2400 rpm (J/deg)	Net Rate of Heat Release Cylinder 3200 (J/deg)
6.06209	2.46758	1.38653	0.884153

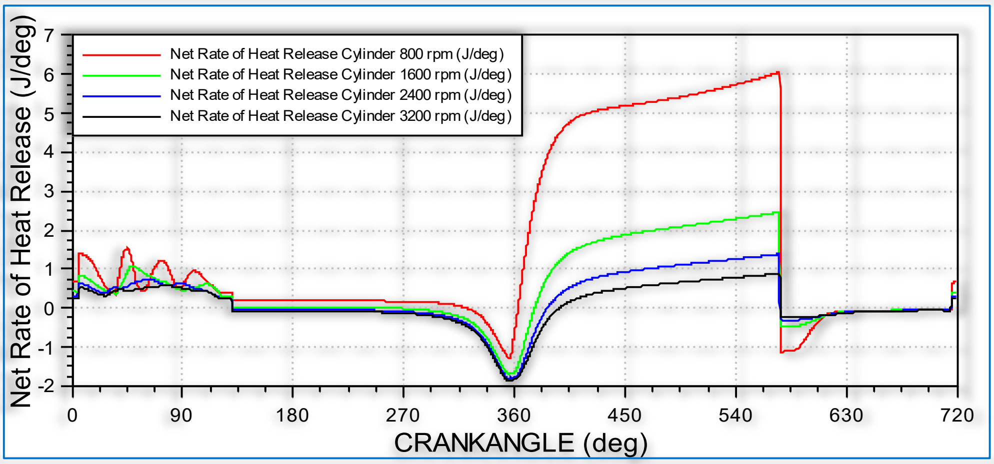

Figure 2. Net heat release rate (diesel). Source: own construction

Another element describing the diesel engine operation at different engine speeds is also the burned volume fraction which decreases with increasing engine speed as shown in Table 3. It can also be seen from Figure 3 that with increasing engine speed the burnt volume fraction in the cylinder decreases but according to the data in Table 1 the amount of fuel mixture used per cycle also decreases. The increase in the level of mechanical work is based on the number of cycles/revolutions per minute that are performed by the engine.

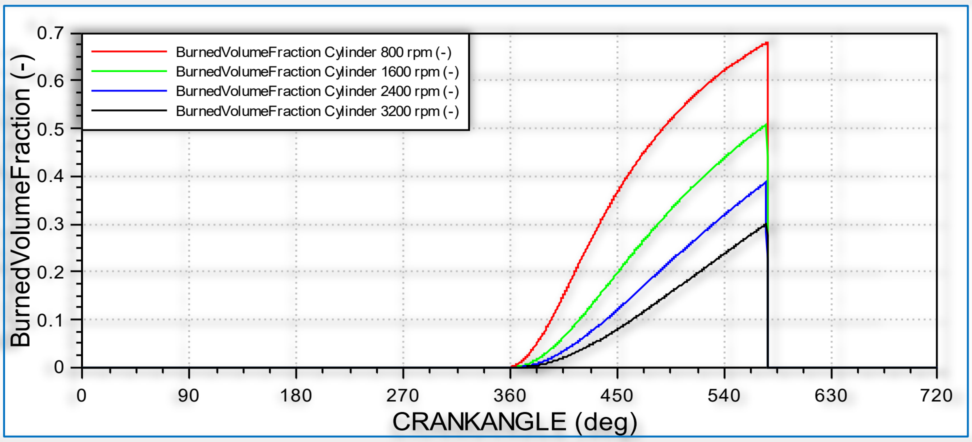

Figure 3. Volume fraction burned (diesel). Source: own construction

Table 3. Fraction of cylinder burned volume (diesel)

	BurnedVolume Fraction Cylinder 800 rpm (–)	BurnedVolume Fraction Cylinder 1600 rpm (–)	BurnedVolume Fraction Cylinder 2400 rpm (–)	BurnedVolume Fraction Cylinder 3200 rpm (–)
%	67,9692	50,8036	38,7288	30,0913

During engine operation there are 2 zones in each cylinder, the combustion zone with a maximum temperature of 2708.03 K at 2400 rpm and the flame failure zone with a maximum temperature of 980.08 K at 3200 rpm. The data are extracted from the excel reports on which the figures are based: Fig. 4, Fig. 5.

From the data in Tables 4 and 5 it can be seen that between the combustion zone and the "unburned" zone there is a temperature difference of at least 1600 K which justifies the incomplete combustion of the fuel mixture in the cylinder.

Table 4. Combustion zone temperature (diesel)

Temperature	TempBurnedZone Cylinder 800 rpm (K)	TempBurnedZone Cylinder 1600 rpm (K)	TempBurnedZone Cylinder 2400 rpm (K)	TempBurnedZone Cylinder 3200 rpm (K)
Zona de combustie	2553.55	2677.52	2708.03	2670.03

Table 5. Temperature of the "nearse" zone (diesel)

Temperature	TempUnBurnedZone Cylinder 800 rpm (K)	TempUnBurnedZone Cylinder 1600 rpm (K)	TempUnBurnedZone Cylinder 2400 rpm (K)	TempUnBurnedZone Cylinder 3200 rpm (K)
The "unburned" area	926.106	894.075	920.863	980.083

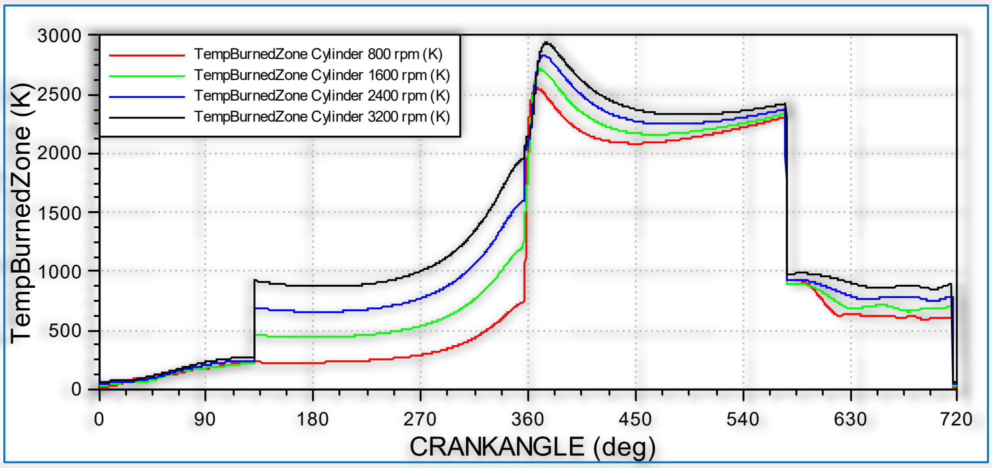

Figure 4 Combustion zone temperature (diesel). Source: own construction

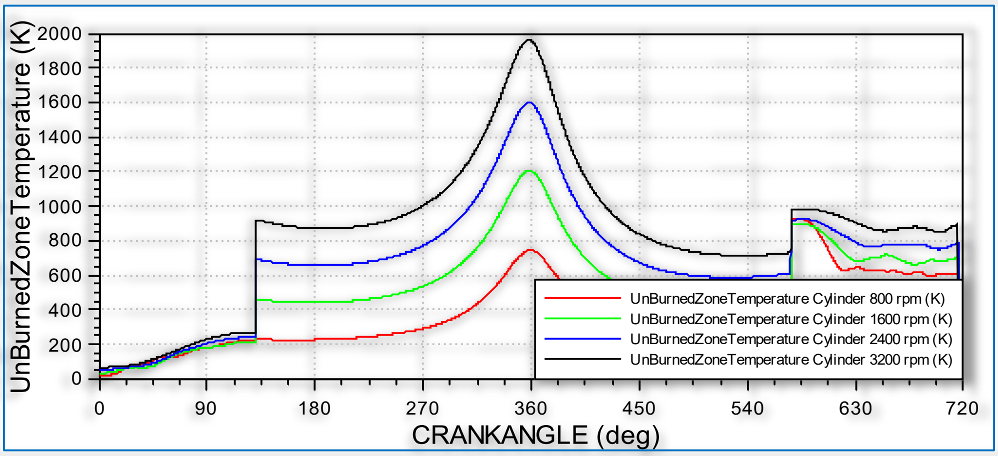

Figure 5. Temperature of the "nearse" (diesel) zone. Source: own construction

The HHO–added combustion process

The burning of a fuel mixture to obtain a quantity of heat that is converted into mechanical work is the phenomenon on which the operation of the heat engine is based.

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Table 6. Mass of fuel mixture in cylinders (diesel+hho)

	Mass Cylinder 800 rpm (g)	Mass Cylinder 1600 rpm (g)	Mass Cylinder 2400 rpm (g)	Mass Cylinder 3200 rpm (g)
Masa A+F	1.6662	0.833168	0.555462	0.416594

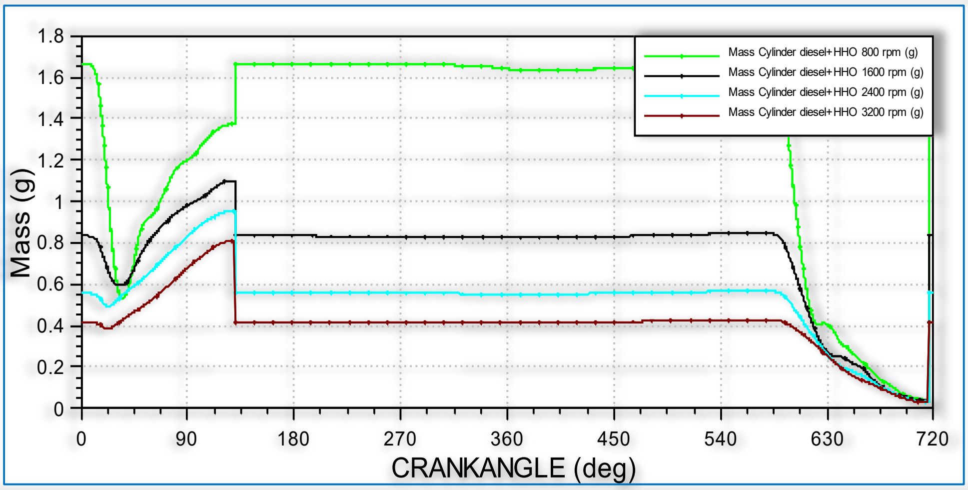

Figure 6. Mass of fuel mixture in cylinders (diesel+HHO). Source: own construction

A first step in evaluating this process is to measure the net amount of heat released by burning the fuel mixture. At different engine speeds the amount of fuel mixture burned is different which also leads to different amounts of heat. The amount of heat decreases from 5.28 J/degree rpm at 800 rpm to 0.74 J/degree at 3200 rpm. Table 7

Table 7. Net heat release rate (diesel+HHO)

Net Rate of Heat Release Cylinder 800 rpm (J/deg)	Net Rate of Heat Release Cylinder 1600 rpm (J/deg)	Net Rate of Heat Release Cylinder 2400 rpm (J/deg)	Net Rate of Heat Release Cylinder 3200 (J/deg)
5.28992	2.11935	1.18055	0.744959

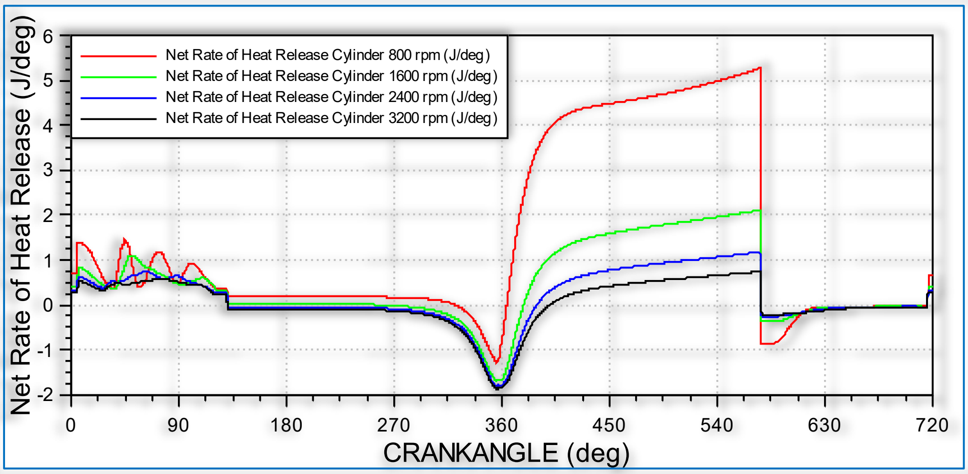

Figure 7. Net heat release rate (diesel+hho). Source: own construction

Another element describing the diesel engine operation at different engine speeds is the burned volume fraction which decreases with increasing engine speed as shown in Table 8. It can also be seen from Figure 8 that with increasing engine speed the burnt volume fraction in the cylinder decreases but according to the data in Table 6.1 the amount of fuel mixture used per cycle also decreases with increasing engine speed.

Table 8. Cylinder burned volume fraction (diesel+HHO)

	BurnedVolume Fraction Cylinder 800 rpm (–)	BurnedVolume Fraction Cylinder 1600 rpm (–)	BurnedVolume Fraction Cylinder 2400 rpm (–)	BurnedVolume Fraction Cylinder 3200 rpm (–)
%	65,3701	47,5468	35,7266	27,4169

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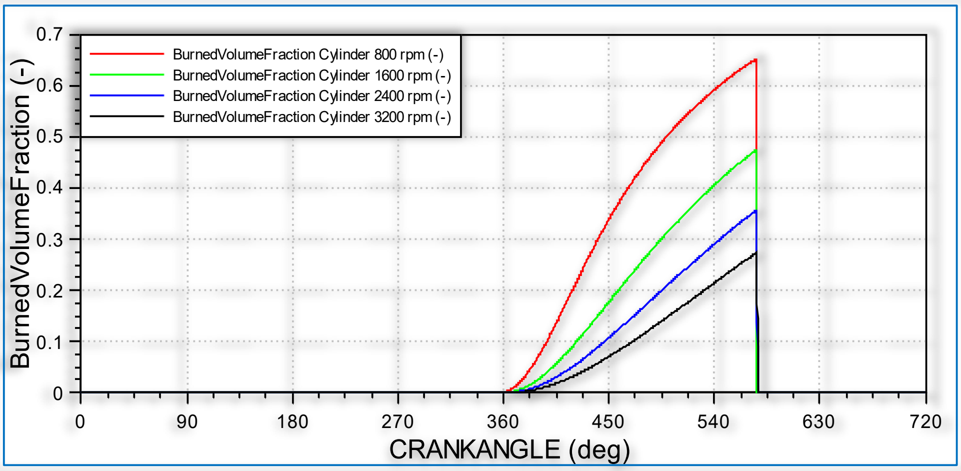

Figure 8. Volume fraction burned (diesel+HHO). Source: own construction

During engine operation in each cylinder there are 2 zones, the combustion zone with a temperature of 2413.4 K at 800 rpm or 2548.63 K at 3200 rpm and the flame failure zone with a temperature of 837.205 K at 800 rpm or 926.44 K at 3200 rpm. The data are extracted from the excel reports on which Figures 9 and 10 are based, only the maximum values for each speed, which are given in Table 9.

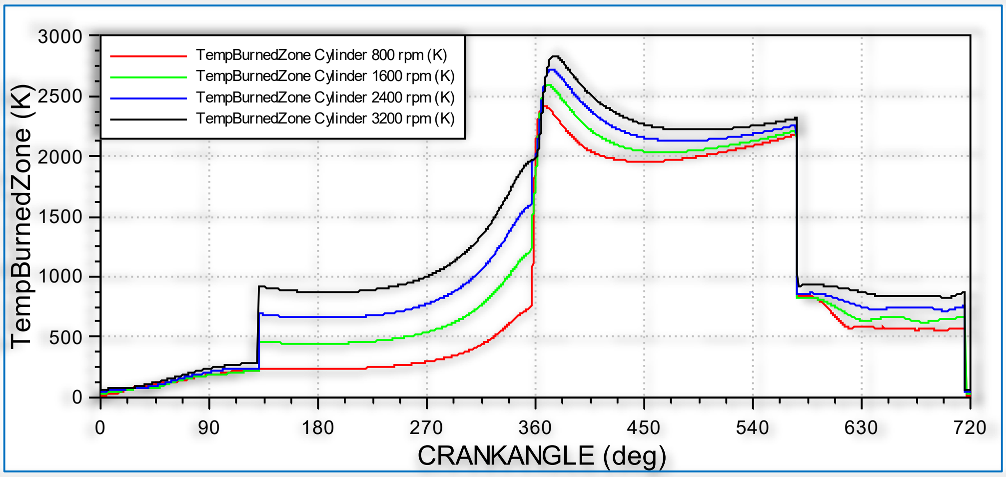

Figure 9. Combustion zone temperature in cylinders (diesel+HHO) . Source: own construction

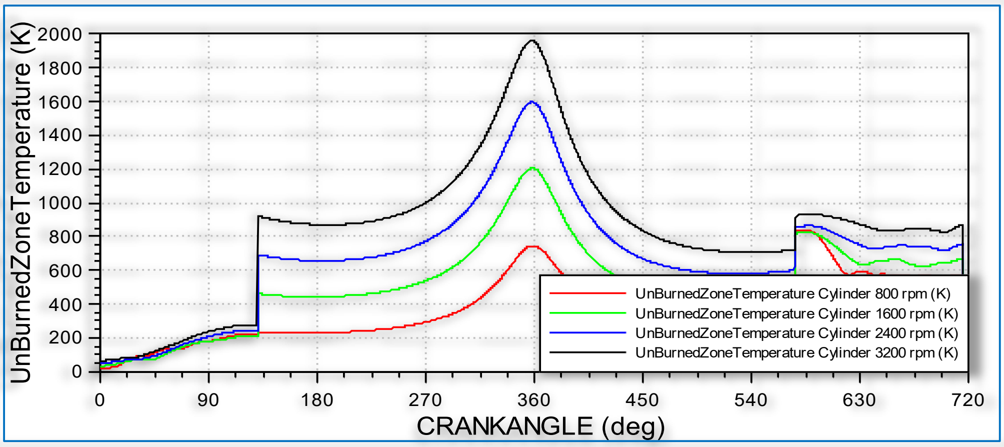

Figure 10. Temperature of the "unburned" zone in the cylinders (diesel+HHO). Source: own construction

Table 9. Combustion zone and "unburned" zone temperature (diesel+HHO)

Temperature	TempBurnedZone Cylinder 800 rpm (K)	TempBurnedZone Cylinder 1600 rpm (K)	TempBurnedZone Cylinder 2400 rpm (K)	TempBurnedZone Cylinder 3200 rpm (K)
Combustion zone	2413.4	2552.04	2593.89	2548.63
"Unburned" zone	837.205	820.504	859.922	926.44

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During engine operation there are 2 zones in each cylinder, the combustion zone with a maximum temperature of 2548.63 K at 3200 rpm and the flame failure zone with a maximum temperature of 926.44 K at the same rpm. (data are taken from the excel reports on which figures 9 and 10 are based).
From the data in Table 9 it can be seen that there is a temperature difference of at least 1500 K between the combustion zone and the "unburned" zone which justifies the incomplete combustion of the fuel mixture in the cylinders.
In our research we started from the analysis of the stoichiometric mixture used in both simulations as follows:

Table 10. Combustible mixtures, specific heat and specific speed

Diesel		speed	Diesel + HHO				
Stoichiometric ratio	specific heat of fuel mixture		diesel	hydrogen	oxygen	specific heat of fuel mixture	Stoichiometric ratio
A/F	KJ/Kg	rpm	% volum	% volum	% volum	KJ/Kg	A/F
14,57	44900	800	0.99896	0.00011	0.00093	42826.549	15.166566
14,57	44900	1600	0.999485	0.000055	0.00046	42828.512	15.168009
14,57	44900	2400	0.999653	0.0000367	0.00031	42829.137	15.168469
14,57	44900	3200	0.999743	0.0000275	0.00023	42829.471	15.168715

From the analysis of Table 10 it can be seen that following the addition of Brown gas the specific heat decreases by an average of 2000KJ/kg and the stoichiometric mixture changes from 14.57 in the case of diesel to 15.16 in the case of diesel and Brown gas.
For an objective evaluation of the data provided I will conclude each aspect related to engine operating parameters with a small conclusion: undecided, advantage or disadvantage.
Starting from this stoichiometric mixture aspect I started to analyse the following aspects of diesel engine operation with different fuel mixtures:

- Fuel mixture mass. Analysis of the data in Table 11 shows that there are no significant differences in the mass of the fuel mixture used by the engine in one operating cycle. The differences are within the limit of 20 mg less per cycle running also with HHO which means less diesel consumption. (HHO advantage)

Table 11. Mass of fuel mixture in cylinders

DIESEL				
	Mass Cylinder 800 rpm (g)	Mass Cylinder 1600 rpm (g)	Mass Cylinder 2400 rpm (g)	Mass Cylinder 3200 rpm (g)
Mass A+F	1.68076	0.850387	0.56933	0.427927
DIESEL + HHO				
	Mass Cylinder 800 rpm (g)	Mass Cylinder 1600 rpm (g)	Mass Cylinder 2400 rpm (g)	Mass Cylinder 3200 rpm (g)
Mass A+F	1.6662	0.833168	0.555462	0.416594

- Net heat release rate. This is important because by burning the fuel mixture and releasing the heat it is possible to evaluate the potential mechanical work that can be achieved. From the analysis of the data in Table 12 we see that the addition of HHO reduces the net heat release rate which means a reduction of the mechanical work obtained. (diesel advantage)

Table 12. Net heat release rate

DIESEL				
Speed	Net Rate of Heat Release Cylinder 800 rpm (J/deg)	Net Rate of Heat Release Cylinder 1600 rpm (J/deg)	Net Rate of Heat Release Cylinder 2400 rpm (J/deg)	Net Rate of Heat Release Cylinder 3200 (J/deg)
(J/deg)	6.06209	2.46758	1.38653	0.884153
DIESEL + HHO				
Speed	Net Rate of Heat Release Cylinder 800 rpm (J/deg)	Net Rate of Heat Release Cylinder 1600 rpm (J/deg)	Net Rate of Heat Release Cylinder 2400 rpm (J/deg)	Net Rate of Heat Release Cylinder 3200 (J/deg)
(J/deg)	5.28992	2.11935	1.18055	0.744959

- Temperature of the combustion zone and the "unburned" zone. Analysing the two sets of data, Table 13, for the combustion zone and the "unburned" zone we can see a difference in temperature between the two types of combustible mixtures. In the case of the HHO mixture the working temperatures are more than 100 K lower in the combustion zone while in the "unburned" zone the difference drops below 100 K. The combustion temperature can be an advantage, mechanically superior, but reduces the service life of the engine. (HHO advantage)

Table 13. Combustion zone temperature

DIESEL				
	TempBurnedZone Cylinder 800 rpm (K)	TempBurnedZone Cylinder 1600 rpm (K)	TempBurnedZone Cylinder 2400 rpm (K)	TempBurnedZone Cylinder 3200 rpm (K)
Combustion zone	2553.55	2677.52	2708.03	2670.03
"Unburned" zone	926.106	894.075	920.863	980.083
DIESEL + HHO				
	TempBurnedZone Cylinder 800 rpm (K)	TempBurnedZone Cylinder 1600 rpm (K)	TempBurnedZone Cylinder 2400 rpm (K)	TempBurnedZone Cylinder 3200 rpm (K)
Combustion zone	2413.4	2552.04	2593.89	2548.63
"Unburned" zone	837.205	820.504	859.922	926.44

6. CONCLUSIONS

Studying various articles that present the effects of Brown's gas addition on diesel engine operating characteristics, several main ideas emerge:

- Some researchers have injected quite large volumes of Brown gas into the engine, ranging from 20 l/min [1], 27,8 l/min [5] or 32 l/min [4] resulting in higher combustion temperatures;
- Some studies report reductions in diesel consumption [11], [14] to mixtures low in Brown gas addition but without specifying the value of the stoichiometric mixture used;
- In some papers [9], [11] occurs as a result of Brown gas injection increased diesel consumption;
- In some cases, in addition to the addition of Brown gas, water has been injected [6] which resulted in reduced NOx emissions due to lower combustion temperatures;
- In all the literature reviewed so far, the amount of NOx increases with the hydrogen injection rate. The increase in fuel economy due to hydrogen injection is not sufficient to compensate for the energy required to produce an equivalent volume of hydrogen by electrolysis of water using engine power. Based on experimental hydrogen research, it appears that HHO produced on–board would increase diesel consumption and reduce the available usable engine power proportionally.

Acknowledgement

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