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INVESTIGATIONS OF EFFECTS OF TIMING INJECTION IN FOUR-STROKE DIRECT-INJECTION DIESEL ENGINE PERFORMANCE

Paulo Gustavo Krejci Nunes
Waldyr Luiz Ribeiro Gallo
Institution and address: Faculdade de Engenharia Mecânica, Universidade Estadual de Campinas - UNICAMP
e-mails: krejcinunes@gmail.com, gallo@fem.unicamp.br

Abstract. *Models of simulation of thermodynamic nature are useful to predict the performance of internal combustion engines. Once adjusted, the simulation is relative fast and do not need complex experimental apparatus to set the engine's best parameters. In this work, a direct injection diesel engine was simulated under a range of timing injection from 0 to 45 BTDC and three proportions of premixed and diffusive combustion. On the best point of start of injection, the simulation showed the specific fuel consumption was reduced in 18.26 percent and the indicated shaft power increased in 22.37 percent. This results shows clearly that a regular commercial diesel engine have a large field to improve the thermal performance and the thermodynamic simulation is a good tool to support decisions for engine project. As long as the technology allows new material resistant to higher thermal or mechanical loads and new techniques to reduce pollutants gases such as nitrous oxide or particulate matter, engine builders can explore better performance and efficiency.*

Keywords: *Diesel Engine, Thermodynamic, Simulation.*

1. INTRODUCTION

Simulation models are useful to predict the engine performance in a relatively fast and inexpensive way when compared to real tests with engine apparatus. According to Heywood (1988), to model a real engine there are two basic types of simulation, the thermodynamic and fluid-dynamic in nature. Phenomenological models are based on thermodynamic energy-conservation and are structured in blocks of submodels to each process in the engine cycle with additional details beyond the energy conservation.

The heat release profile from the energy released by the combustion has a strong effect on the fuel economy, torque, noise and emissions. The fuel is firstly injected into the combustion chamber by the nozzles, in contact with hot and highly compressed air and high turbulence, the particles of fuel auto-ignites. The time between the start of injection and the start of the combustion is know as the ignition delay. Once started, the combustion in direct diesel engines has two phases: the prompt and the diffusive combustion, which are controlled by the reaction rate and by the mixing rate of the fuel with the air respectely.

The best model based on the normal law of distribuition of a continuous random variable known to simulate the combustion the heat release of internal combustion engines was proposed by Ivan Ivanovitch Wiebe. His model can predict the burn fraction and burn rate with different burning systems and fuels. Because of the characteristic of two phases in direct diesel engines, the Wiebe function must be weighted by two coefficients to each phase, known as double Wiebe functions.

Using a computational fluid dynamics (CFD model), Jayashankara (2010) studied the effect of the fuel injection timing and intake-air pressure and temperature of a direct diesel engine. The model pointed the advanced timing injection increases the peak of pressure and temperature during the combustion stroke.

In this study, a phenomenological model was built using MATLAB® language using classic submodels to simulate each process in the engine cycle. In a first investigation to validate the model, a comercial diesel engine, open-chamber turbocharged and turbocooled was simulate and compared with experiments gathered from the engine manufacturer, named Case A.

Results from Case A showed the general performance of the engine could be improved by changing the timing injection and this investigation was named Case B. With new parameter of start of injection from 4.8 to 28 crank degrees before top dead centre (BTDC), the thermal efficiency increased 21.35 % and the brake power increased in 22.37%. In addition, this model of simulation contains a Blowby submodel, unusual to phenomenological models, and studies on the valve timing were also evaluated.

2. MODEL DESCRIPTION

2.1 Energy balace

Assuming there is no change on the kinetic or potential energy, the energy balance in the combustion chamber is given by the first law of thermodynamic for open systems:

$$\frac{dU}{dt} = \frac{\delta Q}{\delta t} - \frac{\delta W}{\delta t} + \sum_j \dot{m}_j h_s \tag{1}$$

Where t is the time, U is the sensible internal energy, W is the external work, Q is the net heat (heat-release by the combustion and heat-loss through the combustion chamber walls) and $m_j h_s$ is the sum of the inlet or outlet enthalpy flows.

The derivate of work is calculated by:

$$\frac{\delta W}{\delta t} = p\frac{dV}{dt} \tag{2}$$

Where V is the volume.

2.2 Geometry and kinematics

The position of the piston as function of the crank position and construction features is given by:

$$S(\theta) = r(1-\cos(\theta)) + l\left(1-\sqrt{1-\left(\frac{r.sen(\theta)}{l}\right)^2}\right) \tag{3}$$

Where:
- S: instantaneous piston position;
- r: crank radius;
- l: connect rod length;
- θ: crank angle.

The area of the combustion chamber (A), as function of the instantaneous position (S), cylinder bore (D), and cylinder head area (A_0):

$$A(\theta) = A_0 + \pi D S(\theta) \tag{4}$$

The instantaneous volume (V):

$$V(\theta) = V_{cc} + \frac{1}{4}\pi D^2 S(\theta) \tag{5}$$

Where V_{cc} is the combustion chamber volume.

2.3 Ignition delay

The time between the start of injection and the rise in cylinder pressure defines the ignition delay. Hardenberg and Hase (1979) developed a model to estimate the ignition delay (t_{id}) wich considers the effects of the fuel quality, engine parameters and ambient conditions, given by:

$$t_{id} = (0.36 + 0.22Vp)\exp\left[E_a\left(\frac{1}{R_0 T\varepsilon^{(n_c-1)}} - \frac{1}{17190}\right) + \left(\frac{21.2}{p\varepsilon^{n_c} - 12.4}\right)^{0.63}\right] \tag{6}$$

And the energy of activation (E_a):

$$E_a = \frac{618840}{CN + 25} \tag{7}$$

Where:
- Vp: mean piston speed;
- R_0: universal gas constant;
- T: absolute air temperature in the intake manifold;
- ε: compression ratio;
- p: absolute air pressure in the intake manifold.

2.4 Combustion ratio

Miyamoto, *et al.*, 1985 developed a double-Wiebe model comprising two single-Wiebe funcions in terms of energy release weighted by the premixed combustion portion (χ_p) and diffusive portion (χ_d):

$$\chi = 1 - \chi_p \exp[a_b (\frac{\theta - \theta_{ig}}{\Delta\theta_p})^{mp+1}] - \chi_d \exp[a_b (\frac{\theta - \theta_{ig}}{\Delta\theta_{di}})^{md+1}] \quad (8)$$

Where:
- a_b : combustion efficiency;
- θ_{ig}: timing of combustion ignition;
- $\Delta\theta_{di}$: duration of diffusive combustion;
- $\Delta\theta_p$: duration of premixed combustion;
- *mp*: combustion chamber shape factor to premixed combustion;
- *md*: combustion chamber shape factor to diffusive combustion.

2.5 Engine heat transfer

The model of heat-transfer utilized is based on the Newton's Law of Cooling. Assuming the same empirical heat-transfer coefficients for all the heat-transfer surfaces in the combustion chamber, it can be described by:

$$\frac{dQw}{dt} = hA(T - Tw) \quad (9)$$

Where *Tw* is the average temperature of the combustion chamber walls. Borman & Nishiwaki (1987) showed that this temperature does not variate more than 10 °C to the entire cycle, even under total load. To calculate the heat-transfer film coefficient (*h*), Hohenberg (1980) developed a correlation based on the Woschni's (1967) model, given by:

$$h = \frac{129.8 p^{0.8}}{Vs^{0.06} T^{0.4}} (V_p + 1.4)^{0.8} \quad (10)$$

During the gas exchange, the heat-transfer film coefficient (*h*) was calculated using the Nishiwaki (1979) correlations. For the intake process:

$$h = C_1 669 D^{-0.193} (0.845 V_p p)^{0.807} T^{-0.534} \frac{4186.8}{3600} \quad (11)$$

For the exhaust process:

$$h = C_1 47.2 D^{-0.422} (V_p p)^{0.578} T^{-0.131} \frac{4186.8}{3600} \quad (12)$$

Where C_1 is an adjust coefficient. In the gas exchange during the crossover period, the heat-transfer coefficient is estimated by a weighted average value.

2.6 Gas exchange model

The gas flow through the valves during the open period was modeled by the a fluid flow through a restriction model. The discharge coefficient (*Cd*) is given by the relation of ideal and real flows:

$$Cd = \frac{Real}{Ideal} \quad (13)$$

This coefficient can be calculated by a polynomial relation, with the constants b_n given in the Table 1.

$$Cd = b_0 + b_1 (\frac{Lv}{Dv})^1 + b_2 (\frac{Lv}{Dv})^2 + ... b_{10} (\frac{Lv}{Dv})^{10} \quad (14)$$

Where *Lv* is the valve lift and *Dv* is the head valve diameter.

Table 1. Constants to calculated the discharge coefficient by Gallo (1990) to valves tested by Kastiner, *et al.,* (1963).

Constants	Value
b_{00}	0.9999676 (00)
b_{01}	7.633573(-1)
b_{02}	-4.089484 (02)
b_{03}	1.885862 (04)
b_{04}	-4.016319 (05)
b_{05}	4.720187 (06)

b_{06}	-3.295265 (07)
b_{07}	1.401485 (08)
b_{08}	-3.567916 (08)
b_{09}	5.004000 (08)
b_{10}	-2.977371 (08)

Considering an ideal gas with constant specific heat, the gas flow rate with velocity under the sound speed of sound is given by:

$$\dot{m} = Cd\,\frac{Ap_1}{\sqrt{RT_1}}\left(\frac{p_2}{p_1}\right)^{1/k}\left[2\frac{k}{k-1}\left(1-\left(\frac{p_2}{p_1}\right)^{\frac{k-1}{k}}\right)\right]^{1/2} \tag{15}$$

Where p_1 is the pressure before the restriction, p_2 is the pressure after the restriction, and k is ratio of specific heat. For choked flow:

$$\dot{m} = Cd\,\frac{Ap_1}{\sqrt{RT_1}}k^{1/2}\left[\frac{2}{1+k}\right]^{\frac{1+k}{2(k-1)}} \tag{16}$$

According to Asmus (1982), the equivalent area (A) of the valve opening can be described into three stages, as the scheme of the Figure 1.

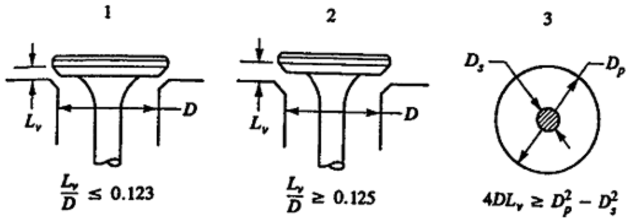

Figure 1. Valve opening stages (Asmus, 1982).

For the first stage, the minimum area é calculated by:

$$A = \pi L_v \cos(\beta)\left(D - 2w + \frac{L_v}{2}sen(2\beta)\right) \tag{17}$$

For the second stage:

$$A = \pi D\left((L_v - wtg(\beta))^2 + w^2\right)^{\frac{1}{2}} \tag{18}$$

For the third stage:

$$A = \frac{\pi}{4}\left(D_p^2 - D_s^2\right) \tag{19}$$

2.7 Blowby model

The flow through the piston-cylinder rings was calculated using the method proposed by Namazian and Heywood (1982). The continuity equation for each region of the piston is given by:

$$\frac{dm_i}{dt} = \dot{m}_{i-1,i} - \dot{m}_{i,i+1} \tag{20}$$

Where $\dot{m}_i$ is the flow between regions of the piston, with $i = 1$ to n regions.
By the equations of fluid flow through a restriction, considering the discharge coefficient $C_d = 0.86$ constant to all cases, the blowby can be calculated.

3. RESULTS AND DISCUSSION

3.1 Effect of start of injection

The validation of the model of simulation by comparison with experiments gathered from the engine manufacturer showed maximum error of 1.37 % in the indicated shaft power (Nunes and Gallo, 2017). The results from simulation indicated the general performance of the engine could be improved by the changing the start of injection of fuel. By the diagram of the Figure 2, the start of combustion is delayed from the best point "b" where the derivate of temperature became negative and the core gas inside the cylinder start to reduce temperature before the top dead center.

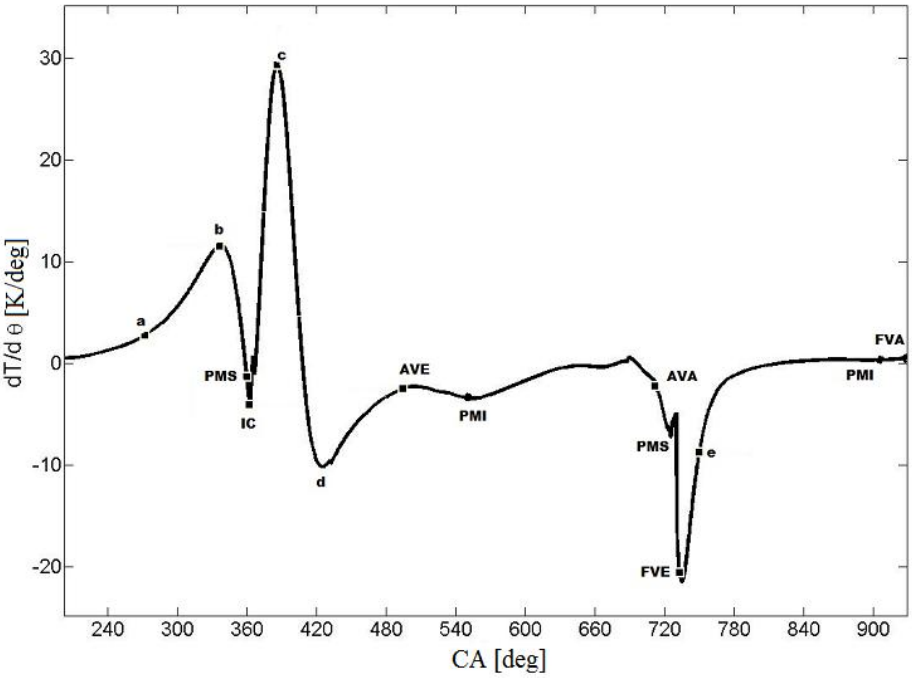

Figure 2. Diagram of derivative temperature per degree versus crank angle for the Case A: "a, e" points of null heat transfer; "b" - max derivative temperature during the compression stroke; "PMS" - top dead centre; "IC" - start of heat release by the combustion; "c" - max derivative temperature during the combustion; "d" - minimum derivate temperature during the combustion process; "AVE" - exhaust valve opening; "PMI" - bottom dead centre; "AVA" - intake valve opening; "FVE" - exhaust valve closing; "FVA" - intake valve closing.

The model of simulation was performed in a range of start of injection (SOI) from 0 to 45 ° BTDC, the Figure 3 shows the range of interest of 16 to 40 BTDC. Although changing the SOI affects the ignition delay and the weight of premixed / diffusive combustion, this parameters were keep the same as the original case and in this conditions, the model simulation points the best thermal efficiency of SOI in 28 ° BTDC.

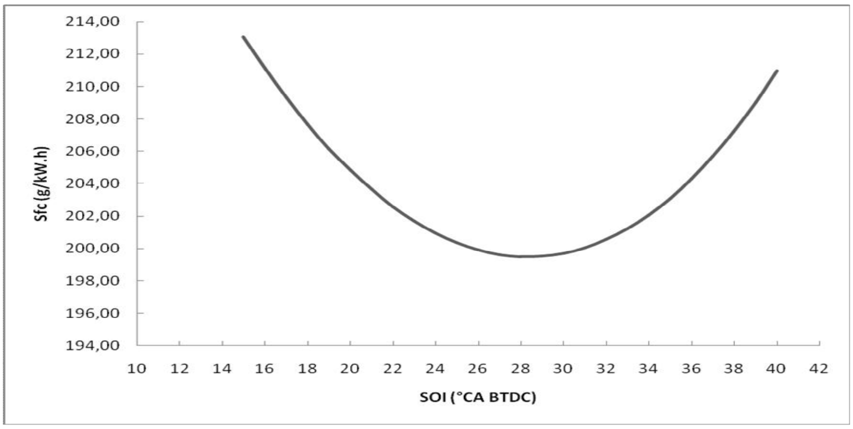

Figure 3. Behavior of the specific fuel consumption for a range of start of injection before top dead centre.

Adopting the same parameters of ignition delay and weight of premixed and diffusive combustion, the apparent heat release by the combustion was the same profile in relation to the Case A (original conditions) but with the start of injection advanced from 4.8 to 28 ° BTDC, as shown in the Figure 4.

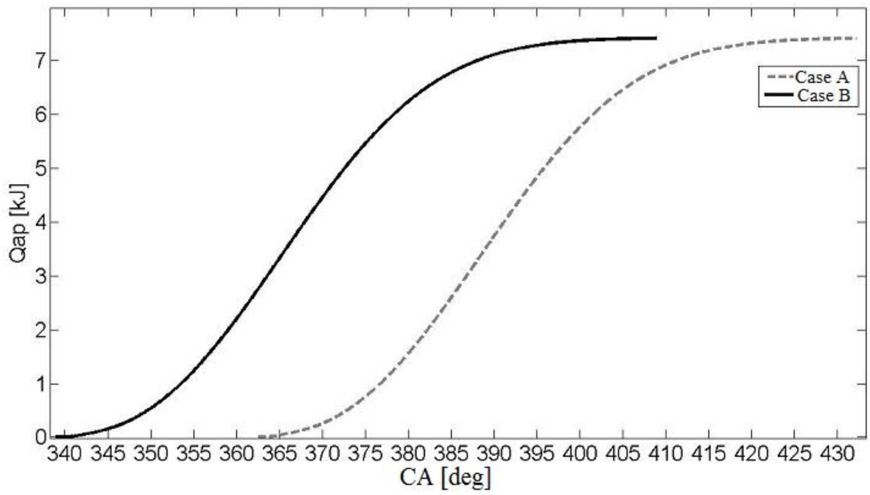

Figure 4. Apparent heat release by combustion for the cases A and B.

Advancing the start of injection, the apparent heat release coincides with the maximum derivative temperature per degree at the compression stroke, as shown in the Figure 5.

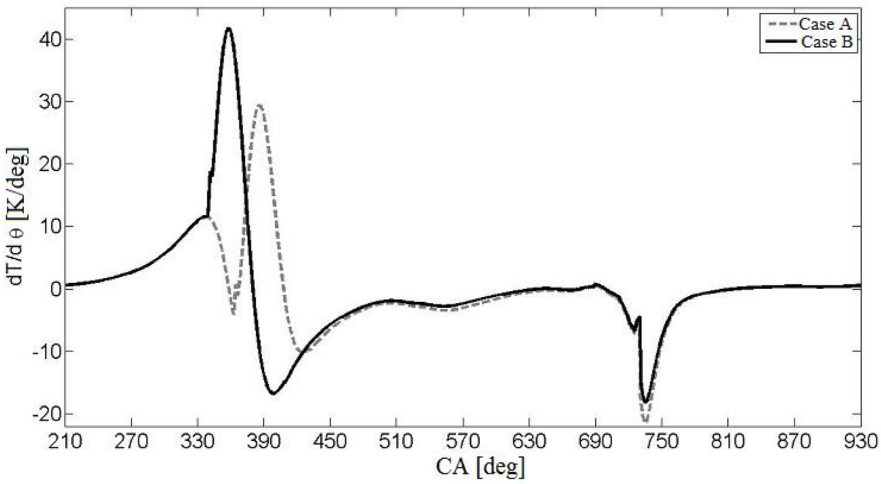

Figure 5. Comparative diagram of derivative temperature per degree versus crank angle for the Case A and B.

To the second case, the apparent heat release by combustion occurs at the same point of maximum derivative temperature. As result, the pressure profile increases during the combustion stroke. In this case, advancing the start of injection from 4.8 to 28 degrees before top dead centre increases the maximum pressure from 115.31 to 188.19 bar to the cases A and B respectively or 63.19 %, as shown in the Figure 7.

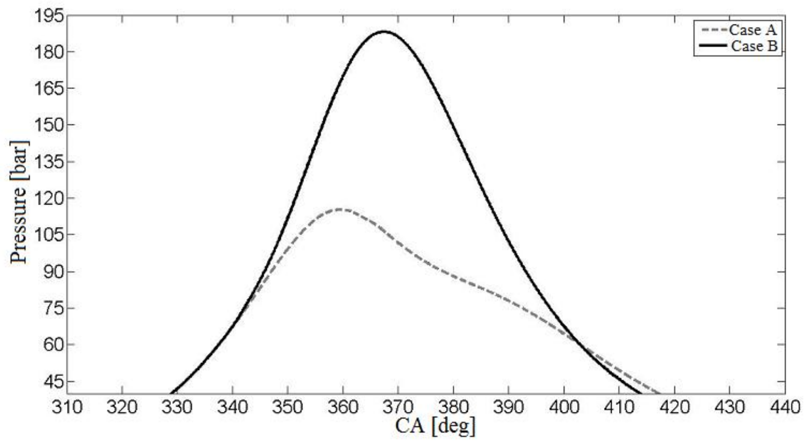

Figure 6. Pressure profile during the combustion stroke to the cases A and B.

The increase in pressure during the combustion stroke increases the work done by the piston during the expansion stroke. The choice of the best point of start of injection must be done very carefully, since the increase of the pressure profile shouldn't promote negative work during the compression stroke. The Figure 7 shows the derivative work done by the piston versus crank angle. Note the rate of work done by the piston during the expansion stroke in the second case (Case B) is higher than the original case (Case A). One can see the rate of work during the compression stroke in both cases is the same. As result, the indicated work done by the piston to the Case B is 21.39 % higher than the Case A.

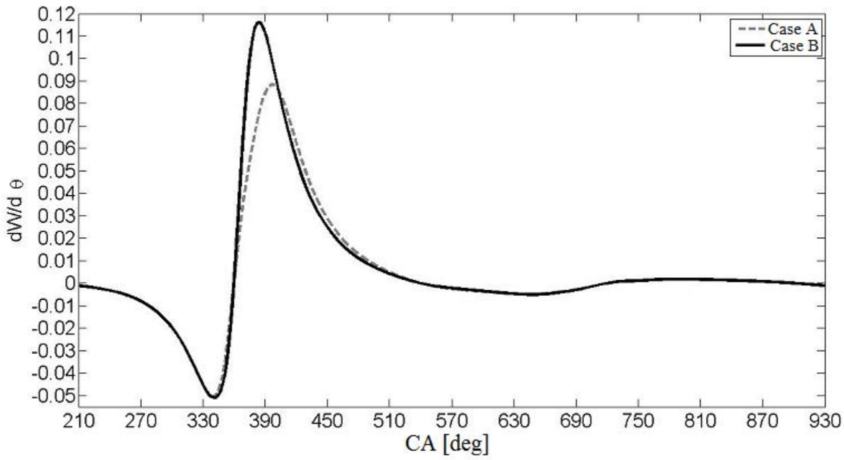

Figure 7. Derivative work done by the piston per crank angle degree for the entire cycle.

The heat transfer by the cylinder walls depends on the pressure and temperature, increasing the pressure increases the heat transfer film coefficient (*h*). The Figure 8 shows the Case B increases the value of *h* during the combustion stroke in 21.42 %.

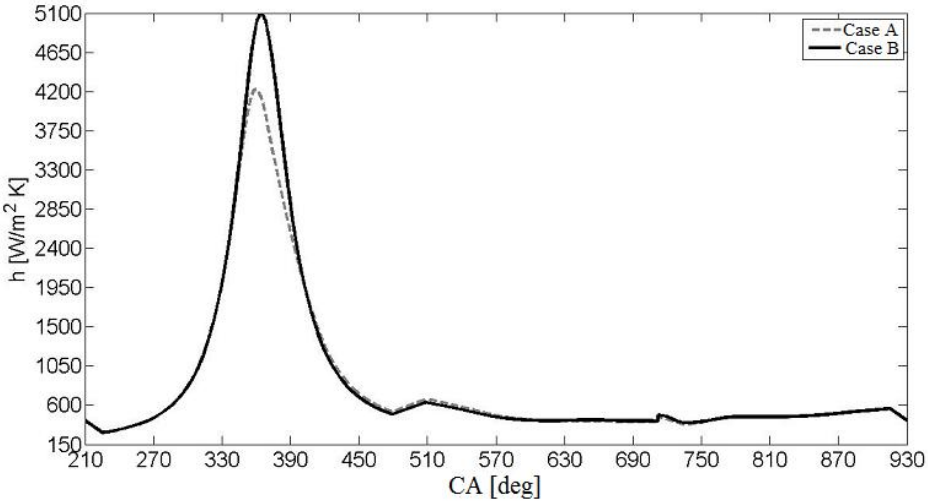

Figure 8. Simulated heat-transfer film coefficient versus crank angle to the Cases A and B.

Higher values to the heat-transfer film coefficient plus higher temperatures increases the net heat-transfer by the cylinder walls. The Figure 9 shows the heat-flow increases mainly during the combustion stroke, where the peak reaches 136.39 percent and the net heat-transfer increases in 15.63 percent to the Case B in relation to the Case A, which implies increase the cooling capacity of the engine.

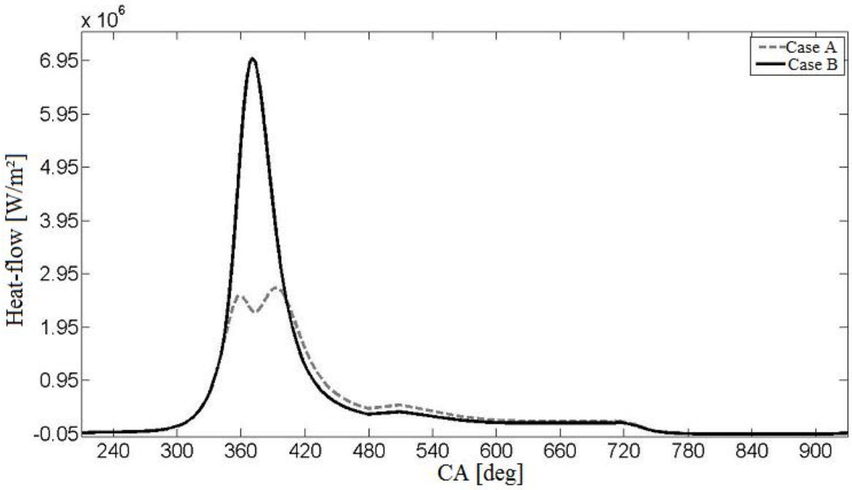

Figure 9. Heat-flow by the cylinder walls versus crank angle.

3.2 Blowby

The figures 10 and 11 shows the percentage of mass trapped in each region of the piston due to blowby flow to the Cases A and B respectively. The region on the top of the piston before the first ring trapped the biggest amount of gas with 3 % to the Case A and 4.7 % to the Case B. Because of the higher pressure in the Case B, the percentage of mass trapped in all regions increased and the amount of gas flow through the piston-cylinder to the crankcase was 0.45 % to the Case A and 0.5 % to the Case B. The total mass of gas flow to the crankcase increased from 117.54 *l/min* in the Case A to 133.24 *l/min* to the second case.

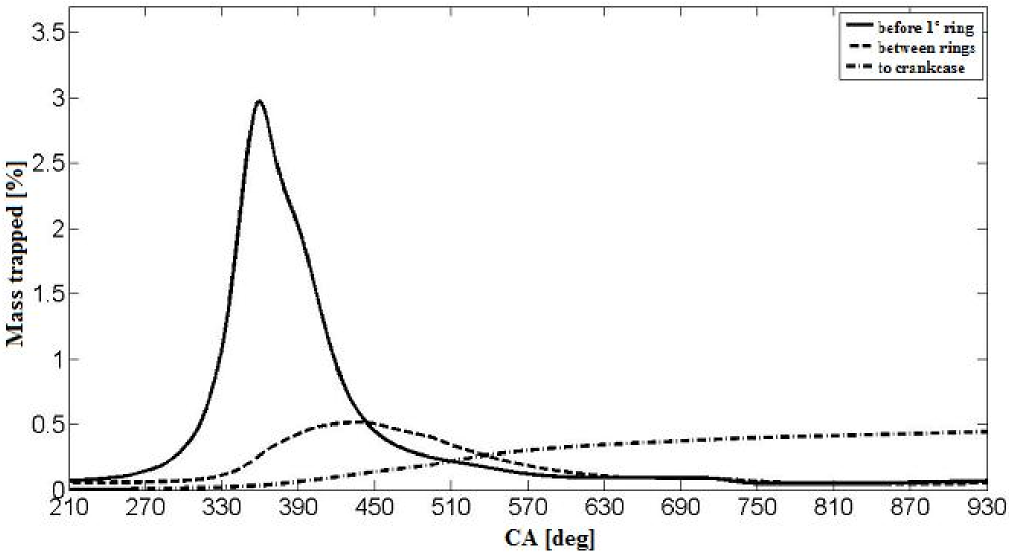

Figure 10. Percentage of mass trapped by blowby in each region of the piston for the Case A.

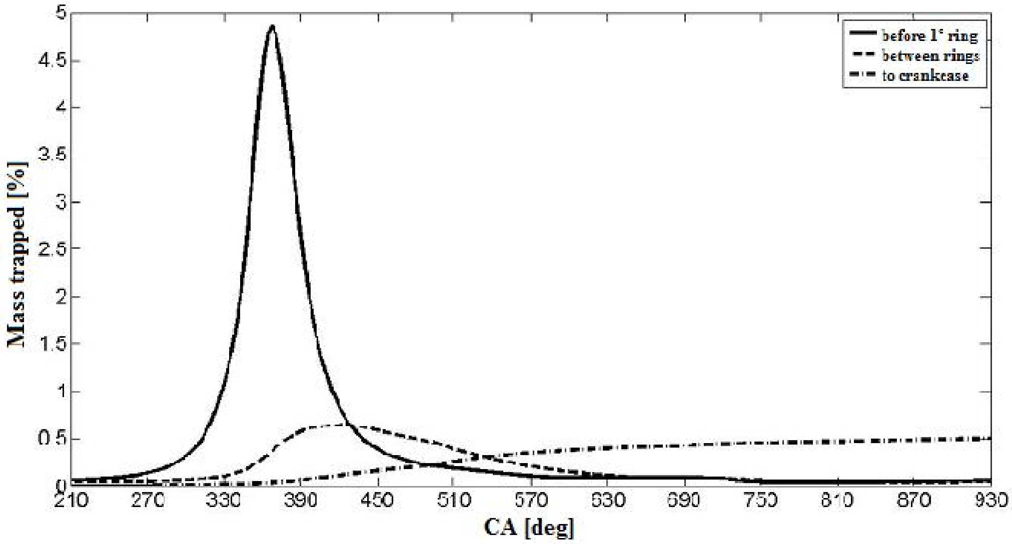

Figure 11. Percentage of mass trapped by blowby in each region of the piston for the Case B.

The Figure 12 compares the percentages heat-losses through the combustion chamber walls in relation to the heat-release by combustion, the thermal efficiency and mechanical efficiency to the Cases A and B. Because of the higher temperature during the combustion stroke and pressure, the heat-transfer flux increased adding the heat-losses in 2.59%; higher pressures during the combustion increase the friction of the piston on the cylinder and bearings. On the other hand, the indicated power shaft increased too, resulting in a better mechanical efficiency from 85.00 to 85.69%. Although the heat-losses and internal friction increased, the thermal efficiency increased from 41.31 % to 50.13 %, what is reflected on the specific fuel consumption that reduced from 244.08 to 199.50 *g/kWh.*

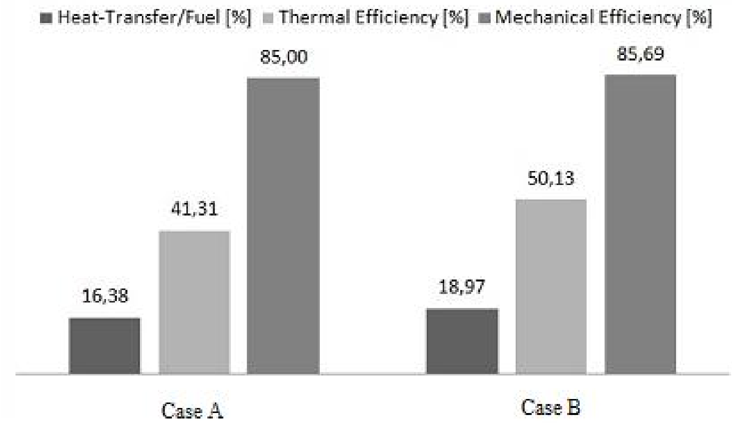

Figure 12. Percentage of heat, thermal and mechanical efficiency to the cases A and B.

This study shows clearly a commercial diesel engine has a large field to improve the shaft power and the thermal efficiency in terms of combustion. However, additional investigations must be done: higher temperatures during the combustion increase the NOx formation and affects the particulate matter production; higher pressure and temperature increase the load on the bearings and rings, which can reduce the, life engine or increase the failure chances of the engine.

3.3 Valve timing

In order to improve the volumetric efficiency of the engine, the first study was evaluate the gas exchange during the crossing timing of the valves. As shown in the Figure 13, both valves of exhaust and intake have reverse flow during the crossing timing. To prevent this effect, the exhaust valve close was changed from 733 to 723 crank angle degree and the intake valve opening from 712 to 723 crank angle degree. In fact, the back flow was avoided from both valves, but the residual increases from 3.70 to 5.31 % and the specific fuel consumption increases in 2.04 %. Although the shift of the valves timing without crossing timing reduces the back flow, it prejudice the percentage of residual gas and the thermal efficiency of the engine, in order it is preferred to keep the original set of valve timing.

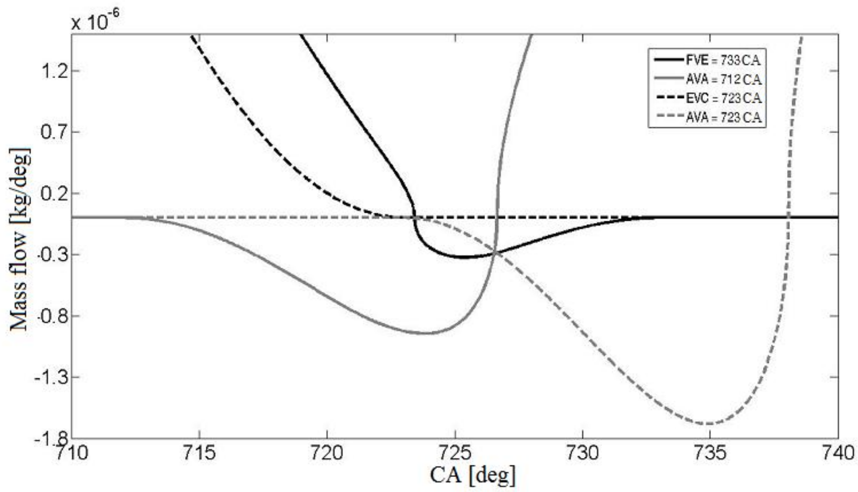

Figure 13. Mass-flow by the exhaust and intake valves during the crossing-timing of gas exchange.

4. CONCLUSION

- Phenomenological models are useful and shows good agreement with real results;
- The heat-release profile has a strong influence on the power and thermal-efficiency;
- Because of the higher temperature and pressure during the combustion, more investigations on the life cycle and polluting formation must be done;
- In order to obtain results more precise, the blowby model must consider the fluttering effect on the piston-rings;
- The optimum adjust of start of injection (SOI) depends on the characteristics of combustion chamber, fuel, operational parameters and ambient conditions. To this engine, the best SOI was 28 BTDC which provided reduction of 18.26% in the specific fuel consumption and the indicated power shaft increased in 22.37%.

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