VALVE TIMING OPTIMIZATION FOR A SINGLE CYLINDER DIESEL ENGINE

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Abstract. *This work presents results of a valve timing optimization calculation for a single cylinder diesel engine, which focused on the maximization of its volumetric efficiency. A computational model that allows for simulating in-cylinder processes as well as the gas flow in admission and exhaust ducts was used in this optimization. In-cylinder processes were simulated by using the single zone combustion model, while admission and exhaust flows were simulated admitting one-dimensional, unsteady and compressible behaviour, subjected to heat transfer and friction between the flowing gas and the solid walls. The equations that govern such a flow were solved using the method of characteristics. In order to obtain the optimal valve timing, two optimization procedures were used: the Zoutendikj method of feasible directions (FD) and the differential evolution method (DE). Both these procedures allowed obtaining the same maximized volumetric efficiency, which resulted being 8.4% higher than the inicial value. Nevertheless, they converged to slightly different values of valve timing parameters and, furthermore, the calculation made with the FD method consumed a significant smaller computational time than that of the DE method.*

Keywords: *Multivariable optimization, differential evolution optimization method, Zoutendikj optimization method of feasible directions, optimal engine valve timing, maximum engine volumetric efficiency.*

1. INTRODUCTION

Internal combustion engine functioning requires the gases burnt in each cycle to be discharged to the atmosphere in order to allow filling the cylinder with a fresh charge. Such tasks are accomplished in the exhaust and intake processes, respectively, which markedly influence engine performance and efficiency. The quality of these gas exchange processes is usually evaluated through parameters such as the volumetric efficiency or the residual gas ratio. Additionally, exhaust and intake processes influence mutually one to other, and this interdependence results not only from the fact that they can occur simultaneously during a limited time, but also because the quantity of gas admitted into the cylinder during the intake process is determined in some measure by the effectiveness with which the cylinder was unfilled during the exhaust process.

On the other hand, the exhaust process and specially the blow-down, strongly depends on the temperature and on the pressure of the working fluid at the ending of the expansion process, and the latter are determined by its turn by the phenomena that take place in the cylinder when the valves still are closed.

These considerations permit concluding that the optimization of the gas exchange processes must be conducted analyzing all phenomena that occur in the engine, including the in-cylinder processes as well as those processes that take place in the admission and exhaust ducts.

This work presents an optimization of admission and exhaust valves timing, in which the maximization of volumetric efficiency was adopted as target function. Two mathematical procedures were used: the Zoutendikj method of feasible directions, and the differencial evolution method. Instants of valve opening and duration of admission and exhaust processes (all expressed in terms of crank-angle) were used as decision variables. In order to evaluate the target function, a computational model, which allows simulating the propagation of pressure waves in the admission and exhaust ducts, as well as the in-cylinder phenomena was used.

2. IN CYLINDER PROCESSES MODEL

In order to simulate in-cylinder processes the single zone combustion model, as enounced by Krieger & Borman (1966), was used. According to this model, the working fluid was treated as an ideal gas mixture in chemical equilibrium. Additionally, it was assumed that the working fluid is formed of the following components: H, O, N, H_2, OH, CO, NO, O_2, H_2O, CO_2, N_2 e Ar.

JANAF tables (Stull & Prophet, 1971) were taken as the main reference for the calculation of the properties of each ideal gas component. Besides that, in order to determine the equilibrium composition of the gas mixture, the following

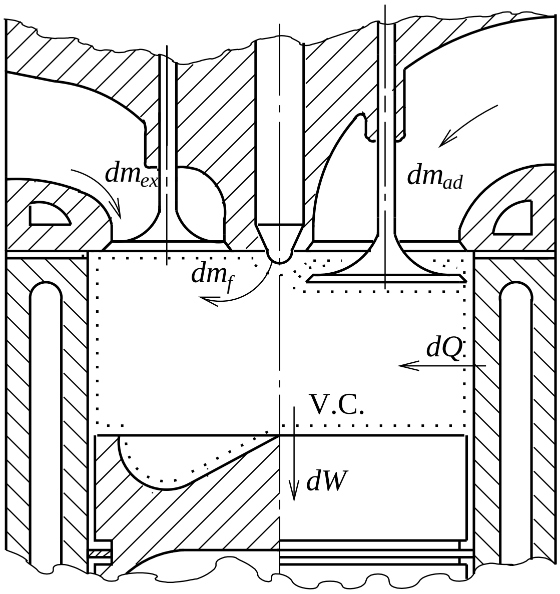

Figure 1. Control volume for in-cylinder processes analysis.

combustion reaction was considered

$$x_{13}\left[C_nH_mO_lN_k+\frac{n+m/4-l/2}{\phi}\left(O_2+\alpha N_2+\beta CO_2+\gamma H_2O+\delta Ar\right)\right]\rightarrow x_1H+x_2O+x_3N+$$
$$+x_4H_2+x_5OH+x_6CO+x_7NO+x_8O_2+x_9H_2O+x_{10}CO_2+x_{11}N_2+x_{12}Ar$$

where

ϕ	–	fuel–air equivalence ratio;
$x_1,\ldots,x_{12}$	–	mole fractions of the combustion products;
x_{13}	–	number of moles of fuel per mole of products;
$\alpha,\beta,\gamma,\delta$	–	number of moles, in the environmental air, of N_2, CO_2, H_2O and Ar, respectively, per mole of O_2.

The energy balance equation for the control volume shown in Fig. 1 was written as follows

$$dU=\delta Q-\delta W+h_f\,dm_f+h_{ad}\,dm_{ad}+h_{ex}\,dm_{ex} \tag{1}$$

$$\begin{aligned} dU &= m\,du+u\,dm \\ \delta W &= p\,dV \\ dm &= dm_f+dm_{ad}+dm_{ex} \end{aligned}$$

From this equation and from the ideal gas state equation the crank-angle derivatives of the working fluid temperature and pressure were obtained,

$$\frac{dT}{d\theta}=\left\{\frac{dV}{d\theta}\left[\frac{\mathcal{A}}{V}-\frac{p}{m}\right]-\frac{d\phi}{d\theta}\left[\frac{\partial u}{\partial\phi}+\frac{\mathcal{A}}{R}\frac{\partial R}{\partial\phi}\right]+\frac{1}{m}\left[\frac{\delta Q}{d\theta}+\sum\left(h_i-u-\mathcal{A}\right)\frac{dm_i}{d\theta}\right]\right\}\cdot\left\{\frac{\mathcal{A}}{T}+\frac{\partial u}{\partial T}+\frac{\mathcal{A}}{R}\frac{\partial R}{\partial T}\right\}^{-1} \tag{2}$$

$$\frac{dp}{d\theta}=p\cdot\left\{\left[\frac{1}{T}+\frac{1}{R}\frac{\partial R}{\partial T}\right]\frac{dT}{d\theta}+\frac{1}{m}\frac{dm}{d\theta}-\frac{1}{V}\frac{dV}{d\theta}+\frac{1}{R}\frac{\partial R}{\partial\phi}\frac{d\phi}{d\theta}\right\}\cdot\left\{1-\frac{p}{R}\frac{\partial R}{\partial p}\right\}^{-1} \tag{3}$$

In order to integrate these expressions it is necessary to know the rate of heat transfer between the working fluid and the cylinder walls, $\delta Q/d\theta$, as well as the mass flow rates through the control volume boundaries, $dm_i/d\theta$.

In this work, the heat transfer rate was determined considering five surfaces with uniform temperatures (cylinder, head, piston and admission and exhaust valves) subjected to convective heat transfer. The heat transfer coefficient h was determined using the correlation proposed by Woschni (1967),

$$h=a\,D_c^{-0.2}\,p^{0.8}\,\mathcal{W}^{0.8}\,T^{-0.53}$$

where a is a constant, D_c is the cylinder bore and $\mathcal{W}$ is a characteristic velocity defined as

$$\mathcal{W} = c_1\, w_p + c_2\, \frac{V_h\, T_1}{p_1\, V_1}(p - p_0)$$

w_p	–	mean piston speed;
V_h	–	displaced volume;
p_0	–	cylinder pressure under motoring conditions;
T_1, p_1, V_1	–	in-cylinder temperature, pressure and volume, relative to a cycle instant when the valves are closed;
c_1, c_2	–	adjusting constants.

Mass flow across the control volume boundaries includes admission and exhaust gas flows, as well as the inflow of fuel injected into the combustion chamber. Admission and exhaust mass flows were determined using the model described in Section 3. Additionally, in order to evaluate the mass flow rate of injected fuel it was taken into account that according to the single-zone combustion model it is assumed to be equal to the apparent fuel burning rate, which was modeled in the present work using two Wiebe functions (Miyamoto *et al.*, 1985) as follows

$$\frac{dm_f}{d\theta} = m_f \left(\chi_p\, \frac{d\Psi_p}{d\theta} + (1 - \chi_p)\, \frac{d\Psi_d}{d\theta} \right)$$

$$\frac{d\Psi_p}{d\theta} = 6.908\, \frac{\zeta_p + 1}{\Delta\theta_p} \left(\frac{\theta - \theta_{ig}}{\Delta\theta_p} \right)^{\zeta_p} \cdot \exp\left[-6.908 \left(\frac{\theta - \theta_{ig}}{\Delta\theta_p} \right)^{\zeta_p + 1} \right] \tag{4}$$

$$\frac{d\Psi_d}{d\theta} = 6.908\, \frac{\zeta_d + 1}{\Delta\theta_d} \left(\frac{\theta - \theta_{ig}}{\Delta\theta_d} \right)^{\zeta_d} \cdot \exp\left[-6.908 \left(\frac{\theta - \theta_{ig}}{\Delta\theta_d} \right)^{\zeta_d + 1} \right] \tag{5}$$

where

χ_p	–	fraction of fuel burned during premixed stage;
$\Delta\theta_p$	–	duration of premixed burning;
$\Delta\theta_d$	–	duration of diffusion burning;
ζ_p	–	shape factor for premixed burning;
ζ_d	–	shape factor for diffusion burning.

3. MODEL FOR PROCESSES IN ADMISSION AND EXHAUST DUCTS

In order to model the processes in the admission and exhaust ducts, unidimensional and transient flow of a compressible fluid was assumed. Furthermore, variation of ducts cross section was allowed as well as heat transfer and friction between the flowing gas and the rigid duct walls. Therefore, the equations expressing balances of mass, momentum and energy resulted

$$\frac{\partial \rho}{\partial t} + w\, \frac{\partial \rho}{\partial z} + \rho\, \frac{\partial w}{\partial z} + \rho w\, \frac{1}{A}\, \frac{dA}{dz} = 0 \tag{6}$$

$$\frac{\partial w}{\partial t} + w \frac{\partial w}{\partial z} + \frac{1}{\rho} \frac{\partial p}{\partial z} + F = 0 \tag{7}$$

$$\frac{\partial s}{\partial t} + w \frac{\partial s}{\partial z} = \frac{kR}{c^2}(\dot{q} + wF) \tag{8}$$

A	–	area of flow;
c	–	sound local velocity in the fluid;
k	–	ratio of specific heats at constant pressure and volume;
p	–	pressure;
$\dot{q}$	–	heat transfer rate, per unit mass of flowing gas;
R	–	gas constant;
t	–	time;
w	–	velocity;
z	–	spatial coordinate;
ρ	–	fluid density.

These equations can be written in terms of the derivatives of pressure, velocity and entropy s as follows

$$\frac{\partial \boldsymbol{U}}{\partial t} + \boldsymbol{A}\, \frac{\partial \boldsymbol{U}}{\partial z} = \boldsymbol{f} \tag{9}$$

where

$$\boldsymbol{U} = \begin{pmatrix} p \\ w \\ s \end{pmatrix} \qquad \boldsymbol{A} = \begin{pmatrix} w & c^2\rho & 0 \\ 1/\rho & w & 0 \\ 0 & 0 & w \end{pmatrix} \qquad \boldsymbol{f} = \begin{pmatrix} \rho\,(k-1)\,(\dot{q}+wF) - \rho w\,\frac{c^2}{A}\,\frac{dA}{dz} \\ -F \\ \frac{k\,R}{c^2}\,(\dot{q}+wF) \end{pmatrix}$$

$$F = C_f\,\frac{4}{D_d}\,\frac{|w|}{w}\,\frac{w^2}{2}$$

In the last equation D_d is the local diameter of the duct and C_f is the Fanning friction factor, defined as a function of the wall shear stress τ_w according to $C_f = (2\,\tau_w)/(\rho\,w^2)$.

The matrix of coefficients $\boldsymbol{A}$ presents the following eigenvalues,

$$\gamma_1 = w + c \qquad \gamma_2 = w - c \qquad \gamma_3 = w$$

As these eigenvalues are real and different among them, Eq. (9) represents a hyperbolic system, whose characteristic curves are the following,

$$\frac{dz}{dt} = w + c \qquad \frac{dz}{dt} = w - c \qquad \frac{dz}{dt} = w$$

the respective compatibility equations result,

Mach lines

$$(dc)_{mach} \pm \frac{k-1}{2}\,(dw)_{mach} = \frac{k-1}{2}\,\frac{c}{kR}\,(ds)_{mach} + \frac{k-1}{2}\left(-\frac{wc}{A}\,\frac{dA}{dz} + (k-1)\,\frac{1}{c}\,(\dot{q}+wF) \mp F\right)dt \tag{10}$$

Path line

$$(ds)_{path} = \frac{k\,R}{c^2}\,(\dot{q}+wF)\,dt \tag{11}$$

In order to integrate compatibility equations along the respective characteristic curves the procedure proposed by Payri *et al.* (1986) was followed, as shown in Velásquez & Milanez (1996).

4. OPTIMIZATION METHODS

4.1 The Zoutendikj method of feasible directions

The Zoutendikj method of feasible directions (FD) can be used to search the minimum of a function $f(x)$, so that vector $x \in R^n$ satisfies the restrictions $g_j(x) \leq 0$, $j = 1, \ldots, m$ (Belegundu & Chandrupatla, 1999; Panteleev & Letova, 2005).

The problem is formulated as follows: Determine the point $x^* \in X$ that satisfies the following condition

$$f(x^*) = \min_{x \in X} f(x), \qquad X = \{x \,|\, g_j(x) \leq 0, \quad j = 1, \ldots, m\} \tag{12}$$

The search strategy consists in constructing a sequence of points $\{x^k\}$, so that $f(x^{k+1}) < f(x^k), k = 0, 1, \ldots$. In order to obtain the sequence of points $\{x^k\}$ the following approach is used

$$x^{k+1} = x^k + \bar{t}_k \cdot d^k, \quad k = 0, 1, \ldots \tag{13}$$

where x^k is a feasible point, so that

$$-\varepsilon_k < g(x^k) \leq 0, \quad j \in J_a \tag{14}$$

here, J_a represents all the indexes j of those restrictions for which the condition (14) was satisfied.

The step $\bar{t}_k \geq 0$ must be found solving the following one-dimensional minimization problem,

$$f(x^k + t_k \cdot d^k) \rightarrow \min \tag{15}$$

$$g_j(x^k + t_k \cdot d^k) \leq 0, \quad j = 1, \ldots, m \tag{16}$$

This one-dimensional problem can be solved using any algorithm based on the verification of necessary and sufficient conditions for a local minimum. If such a minimum does not exist, step $\bar{t}_k$ must be choosen from

$$\bar{t}_k = \min\{t_k^* \geq 0,\, t_k^{**} \geq 0\} \tag{17}$$

where, t_k^* and t_k^{**} are determined, respectively, from the following expressions

$$f(x^k + t_k^* \cdot d^k) = \min_{t_k \geq 0} f(x^k + t_k \cdot d^k) \tag{18}$$

$$t_k^{**} = \min\{t_k^j\}, \text{ and } t_k^j \text{ satisfies the conditions } g_j(x^k + t_k \cdot d^k) = 0,\ t_k \geq 0 \tag{19}$$

The search direction d^k is obtained from the following inequation system

$$\nabla f(x^k)^T \cdot d^k < 0 \tag{20}$$
$$\nabla g_j(x^k)^T \cdot d^k < 0, \quad j \in J_a \tag{21}$$

being determined by solving the following linear programming problem

$$\begin{aligned} z &\rightarrow min, \\ \nabla f(x^k)^T \cdot d^k &\leq z \\ \nabla g_j(x^k)^T \cdot d^k &\leq z, \quad j \in J_a \\ |d_i^k| &\leq 1, \quad i = 1, \ldots, n \end{aligned}$$

If the solution z^* of the linear programming problem resulted smaller than $-\varepsilon$, a new feasible direction d^{k+1} must be determined assuming $\varepsilon_{k+1} = \varepsilon_k$. Otherwise, if $z^* \geq -\varepsilon_k$, the minimum was found within an accuracy ε_k.

4.2 The differential evolution method

Differential evolution (DE) is an evolutionary algorithm proposed by Storn and Price (1995). While DE shares similarities with other evolutionary algorithms, it differs significantly in that the information about distance and direction of the current population is used to guide the search process. DE uses the differences between randomly selected vectors (individuals) as the source of random variations for a third vector (individual), referred to as the target vector. Trial solutions are generated by adding weighted difference vectors to the target vector. This process is referred to as the *mutation operation* wherein the target vector is mutated. A *crossover step* is then applied to produce an offspring, which is only accepted if it improves the fitness of the parent individual. The basic DE algorithm is described below with reference to the three evolution operations: *mutation*, *crossover* and *selection*.

Mutation is an operation that adds a vector differential to a population vector of individuals, according to the following equation

$$z_i(k+1) = x_{i,r_1}(k) + f_m \cdot [x_{i,r_2}(k) - x_{i,r_3}(k)] \tag{22}$$

where $i = 1, 2, ..., N$ is the individual's index of population; k is the generation; $x_i(k) = [x_{i1}(k), x_{i2}(k), ..., x_{iN}(k)]^T$ stands for the position of the i-th individual of population of N real-valued n-dimensional vectors; r_1, r_2 and r_3 are mutually different integers and also different from the running index i, randomly selected with uniform distribution from the set $\{1, 2, ..., i-1, i+1, ..., N\}$; $z_i(k) = [z_{i1}(k), z_{i2}(k), ..., z_{iN}(k)]^T$ stands for the position of the i-th individual of a *mutant vector*; and $f_m > 0$ controls the amplification of the difference between two individuals so as to avoid search stagnation, and is usually taken from the range [0.1 – 1].

Crossover applied to the population is employed to generate a trial vector by replacing certain parameters of the target vector with the corresponding parameters of a randomly generated donor vector. For each vector $z_i(k+1)$, an index $rnbr() \in \{1, 2, ..., N\}$ is randomly chosen using uniform distribution, and a trial vector, $u_i(k+1) = [u_{i1}(k+1), u_{i2}(k+1), ..., u_{iN}(k+1)]^T$, is generated with

$$u_{ij}(k+1) = \begin{cases} z_{ij}(k+1) & \text{if } randb(j) \leq CR \text{ or } j = rnbr(i) \\ x_{ij}(k) & \text{if } randb(j) \geq CR \text{ or } j \neq rnbr(i) \end{cases} \tag{23}$$

In the above equations, $randb(j)$ is the j-th evaluation of a uniform random number generation within [0, 1] and CR is in the range [0, 1].

Selection is the procedure of producing better offspring. To decide whether or not the vector $u_i(k+1)$ should be a member of the population comprising the next generation, it is compared with the corresponding vector $x_i(k)$. Thus, if f denotes the objective function under minimization, then

$$x_i(k+1) = \begin{cases} u_i(k+1) & \text{if } f(u(k+1)) \leq f(x_i(k)) \\ x_i(k+1) = x_i(k) & \text{otherwise} \end{cases} \tag{24}$$

In this case, the function f is evaluated at each trial vector $u_i(k+1)$ and compared with the corresponding value of its parent target vector $x_i(k)$. If $f(x_i(k))$ is lower than $f(u_i(k+1))$, the target is allowed to advance to the next generation. Otherwise, the target vector is replaced by a trial vector in the next generation.

Storn and Price (1995) and Storn (1996) proposed ten different strategies for DE based on the individual being disturbed, the number of individuals used in the mutation process and the type of crossover used. The strategy described above is known as DE/rand/1/bin, meaning that the target vector is randomly selected, and only one difference vector is used. This strategy was considered here.

5. APPLICATION TO A SINGLE-CYLINDER DIESEL ENGINE

The simulation model described in sections 2.and 3.was used together with the optimization methods discussed in section 4. The objective of the calculation was to determine the instants of valve opening, as well as the duration of the time intervals when the valves are opened, so that the volumetric efficiency of the engine can be maximized. The engine volumetric efficiency η_v is defined as

$$\eta_v = \frac{m_{air}}{V_h \cdot \rho_{ac}}$$

where m_{air} is the mass of air trapped inside the cylinder when the valves were closed, V_h is the displaced volume of the cylinder and ρ_{ac} is the density of the air at after-compressor conditions.

The analyzed engine was a hypothetical, turbocharged, single-cylinder and direct-injection diesel, whose main dimensions correspond to that of the six-cylinder, MWM Sprint 6.07T engine. Table 1 shows the relevant data used in the

Table 1. Simulation model data.

Engine data			
Compression ratio			17.80
Connecting rod length [m]			0.170
Bore [m]			0.093
Stroke [m]			0.103
Number of valves (intake + exhaust)			2 + 1
Intake valve diameter [m]			0.031
Exhaust valve diameter [m]			0.039
Length of the admission duct [m]			0.380
Length of the exhaust duct [m]			1.500
Admission duct diameter [m]			0.048
Exhaust duct diameter [m]			0.050
Operational condition data			
Engine speed [rpm]			2600
Pressure after compressor [bar]			1.620
Temperature after compressor [K]			360.0
Overall fuel-air equivalence ratio			0.717
Cylinder wall temperatures [K]			
Head			538.0
Piston			637.0
Sleeve			598.0
Admission valves			702.0
Exhaust valve			1016.0
Apparent heat release rate model (Wiebe)			
$\theta_{ini} = -4.74°$	$\Delta\theta_p = 7°$	$\Delta\theta_d = 52°$	
$\zeta_p = 1.8$	$\zeta_d = 0.98$	$\chi_p = 0.05$	
Fuel data			
Empirical formula			$C_{14.4}H_{24.9}$
Lower heating value [MJ/kg]			42.94
Environmental air data			
Pressure [bar]			0.918
Temperature [K]			300.0
O_2 molefraction			0.2082
N_2 molefraction			0.7760
CO_2 molefraction			0.0003
H_2O molefraction			0.0062
Ar molefraction			0.0093

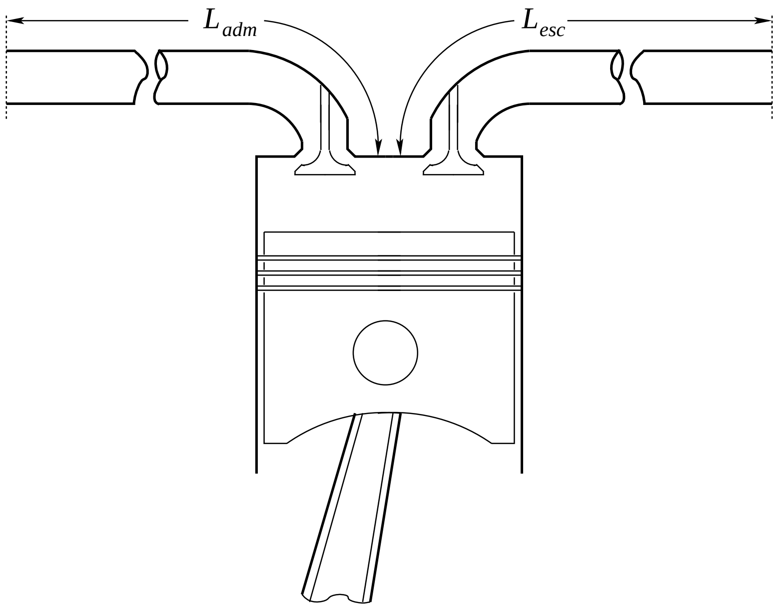

Figure 2. The single-cylinder engine.

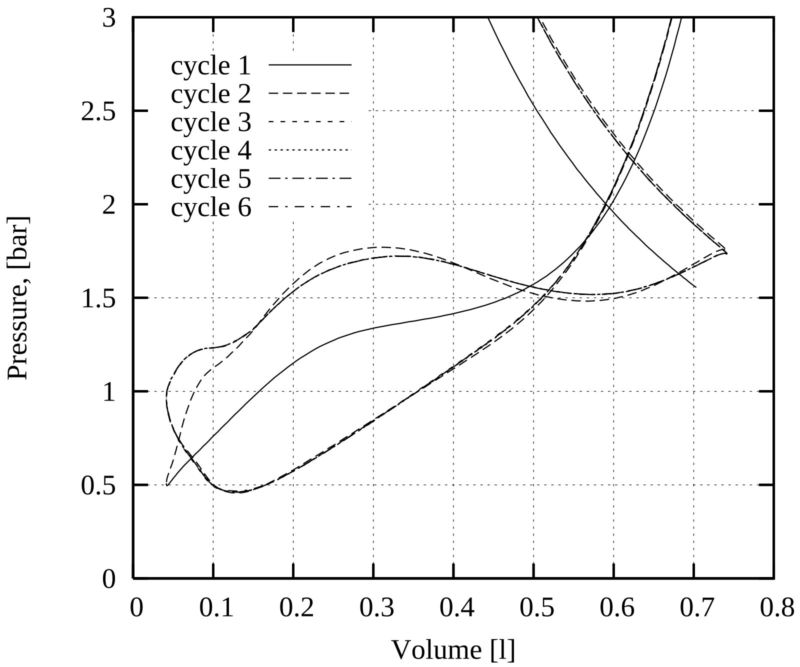

Figure 3. Evolution of the in-cylinder pressure towards a repeatable solution.

simulation, while a schematic representation of the engine is shown in Fig. 2. It is worth mentioning that in order to determine the working cycle of the engine it was necessary to simulate its functioning along a time domain corresponding to six cycles (4320 crank-angle degrees). This was made to eliminate the influence of the inicial conditions, which were choosen arbitrarily and, by doing so, to obtain a periodically repeatable solution. In Fig. 3 it is shown the evolution of the cycle (as a p–V diagram of the gas exchange processes) towards this repeatable solution.

During the optimization calculation, the inverse of the volumetric efficiency was used as the target function to be minimized. The lower and upper boundaries constraints imposed to decision variables are the ones shown in Table 2.

The optimization calculations were started assigning to the decision variables the same values these parameters have in the six-cylinder, MWM Sprint 6.07T engine. In Fig. 4 can be seen the evolution of the volumetric efficiency values during the optimization by the FD method. Both the optimization methods converged to the same value of η_v, nevertheless the optimized value of η_v was obtained with slightly different values of the decision variables. These values are shown

Table 2. Decision variables data.

Parameter	Symbol	Boundary constraints
Admission valves opening (c.a. deg. before TDC)	θ_{avo}	$0 \leq \theta_{avo} \leq 30$
Admission process extension	$\Delta\theta_{ad}$	$180 \leq \Delta\theta_{ad} \leq 220$
Exhaust valve opening (c.a. deg. before BDC)	θ_{evo}	$10 \leq \theta_{evo} \leq 60$
Exhaust process extension	$\Delta\theta_{ex}$	$220 \leq \Delta\theta_{ex} \leq 280$

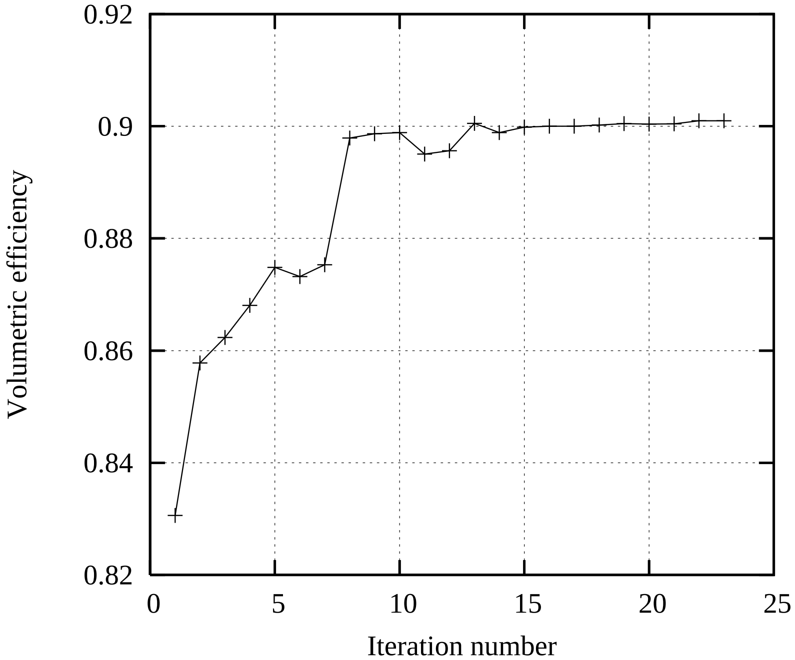

Figure 4. Volumetric efficiency evolution during the optimization calculation (FD method).

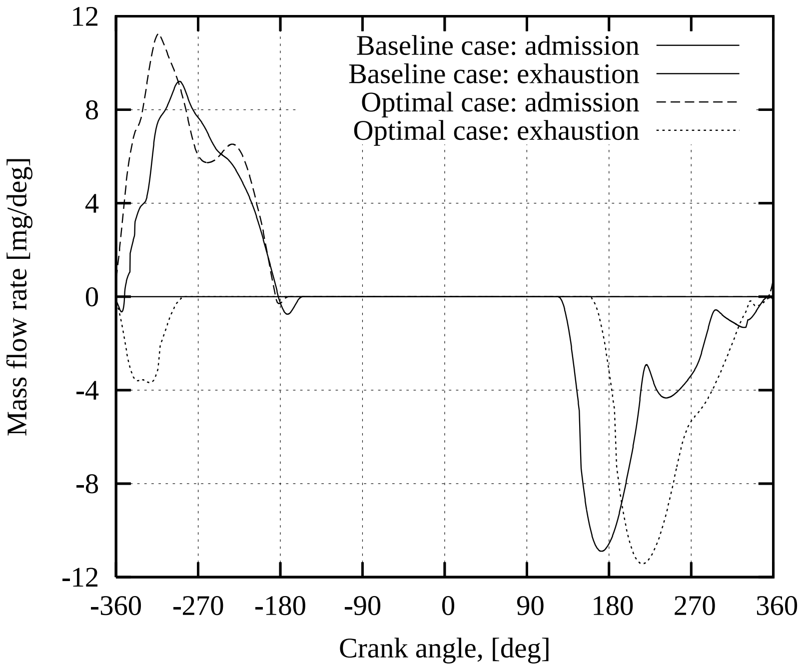

Figure 5. Mass flow rate through admission and exhaust valves.

Table 3. Results of the optimization calculation.

Parametr	Symbol	Initial value	FD method	DE method
Admission valves opening	θ_{avo}	4.0	6.6	6.7
Admission process extension	$\Delta\theta_{ad}$	208.0	195.2	195.2
Exhaust valve opening	θ_{evo}	55.7	21.9	15.6
Exhaust process extension	$\Delta\theta_{ex}$	230.6	277.8	271.8
Volumetric efficiency	η_v	0.831	0.901	0.901

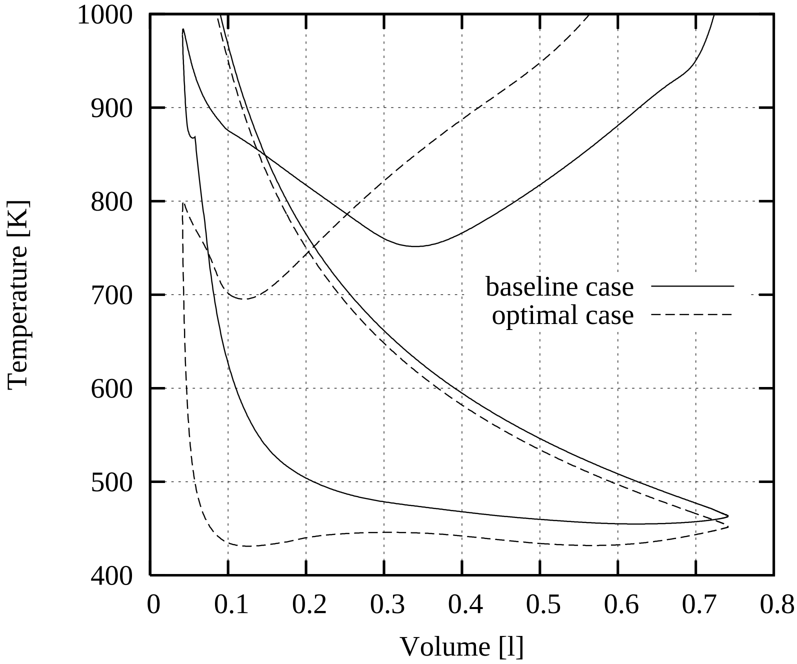

Figure 6. In-cylinder temperature during the gas exchange processes.

in Table 3. As can be seen in this table the optimized volumetric efficiency is significantly greater than its initial value, having passed from 0.831 to 0.901, thus increasing in 8.4% relative to baseline case.

Figure 5 shows the mass flow rates through admission and exhaust valves for both cases, the baseline as well as the optimized. It can be noticed that the exhaust process of the optimized cycle starts later and continues during the admission process, thus generating a long valves-overlapping period of approximately 82.5 crank-angle degrees. Moreover, the mass outflow through the exhaust valve is significant while the admission valve is open, configuring an intense cylinder scavenging.

During the valves overlapping period, cold air is admitted into the cylinder through intake valves and it mixes with the hot combustion gases remanescent from the last cycle. This mixture leaves the cylinder through the exhaust valve, thus reducing the amount of residual gas and diminishing the in-cylinder temperature. This way, the amount of fresh air trapped inside the cylinder rises. In Fig. 6, where the in-cylinder temperature is shown, this effect becomes evident. As can be seen in this figure, when the piston is near the top dead center, the gas temperature was reduced from approximately 980 K into 800 K. In addition, for the optimized cycle the in-cylinder temperature quickly falls down reaching a value as low as 430 K and remains near this value during most of the intake stroke.

6. CONCLUSIONS

It was presented a valve timing optimization for a single cylinder diesel engine. Two optimization procedures were used, the Zoutendikj method of feasible directions and the differential evolution method. Despite both these procedures have converged to the same value of the objective function, the optimal valves timing parameters resulted slightly different for each one of them.

For the analyzed conditions, the optimal valves timing was obtained by opening the exhaust valve later and holding it opened for a longer period (relative to the baseline case). The admission valve by its turn was opened advanced, but the duration of the valve event was made shorter (relative to the baseline case). All this resulted in a large valves overlapping period, which allowed intensifying the cylinder scavenging, thus reducing significantly the temperature of the in-cylinder gas and rising the amount of trapped air at the end of the gas exchange processes.

The optimized volumetric efficiency resulted significantly higher than its inicial value, passing from 0.831 to 0.901, thus it increased by 8.4%, relative to baseline case.

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