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# ON THE USE OF VVT STRATEGY FOR THE MAXIMUM VOLUMETRIC EFFICIENCY IN TURBOCHARGED DIESEL ENGINES

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## ABSTRACT

This paper presents a variable valve lifting methodology for turbocharged diesel engines. For this purpose, the diesel engine is modeled based on a modified mean-value engine modeling. An optimal control strategy is used for maximum volumetric efficiency acquirement. Using camless valve train strategy makes better fuel economy and improved air intake characteristics throughout the engine operating map. The system is capable of continuously, independently and virtually controlling all standard parameters of variable valve motion. This permits optimization of valve events for any operating condition without compromise. The optimized intake valve profile is determined, to have the best volumetric efficiency and proper operation for each running condition based on the existing model make use of numerical techniques. The model used in this paper is validated using simulation results of references.

The model treats the cylinder and the manifolds as thermodynamic control volumes by using the filling and emptying method, solving energy and mass conservation equations with sub models for intake manifold, variable valve timing, cylinder breathing dynamics and turbocharger including turbine and compressor.

This model is a crank angle based dynamic nonlinear model of a four-cylinder turbocharged (TC) diesel engine, which captures the interactions among the VVT actuation, the turbocharger dynamics and the cylinder-to-cylinder breathing characteristics. The model have been implemented in Matlab/Simulink and tested.

This work shows the results obtained for air management control in a turbocharged diesel engine, specifically, manifold pressure and air mass flow. These variables are often required to achieve better power performance and lower emissions.

**KEY WORDS:** Optimization, Optimal Control, Turbocharged Diesel Engines, Volumetric Efficiency, Variable Valve Lifting.

## INTRODUCTION

In connection with growing demands on reduction of Diesel or Compression Ignition Direct Injection (CIDI) engines exhaust gas emissions, fuel consumption and the need for faster, cleaner and more fuel efficient engines simultaneously with increasing of its performance, new designs and optimization methods of existing ones are introduced.

In pursuit of these goals, it is necessary to have as good an understanding of engine operation as possible. In spark ignition engines, the study of the influence of intake properties on the engine performance has included for a long time. However, for diesel engines, the work in this area has not been as extensive.

Several works have been done to evaluate the amount of air entering the cylinder in diesel engines, but most of them use mean-value engine models. Storset et al. [1] have presented an adaptive observer for in-cylinder air charge estimation for turbocharged diesel engines. This observer performs better during fast step changes in engine fueling level. Grizzle et al. [2] describe the development and validation of a non-linear, open loop air charge estimator and assemble a lumped model suitable for developing on-line air charge estimation. Although some studies have shown that variable intake valve timing, cause major reduction in pumping losses and fuel consumption (Ma [3], and Gray [4], and Elrod [5]). Work in the area of maximum lift control that enables stable actuator operation for the electro-hydraulic camless valvetrain can be found in Anderson [6].

 

Kolmanovsky et al. [7] summarizes recent developments in turbocharged diesel engine models, equipped with variable geometry turbine (VGT) and exhaust gas recirculation (EGR). Engine characteristics are discussed from control perspective. Model uncertainty and limitations on feasible sensor locations are discussed. Jung [8] investigates mean value modeling and robust control of the air path of a diesel engine equipped with VGT and EGR. The model is derived with a focus on the parameterization of the turbocharger. Salcedo et al. [9] shows developments made in air management control in a turbocharged diesel engine. A non-linear model using first principles has been developed. Experimental data has also been used to adjust the model.
In this paper, the air charge entering the cylinders downstream of the intake manifold is measured. The system dynamics is limited by the intake and exhaust manifold filling and turbocharger dynamics. Not only experiments but also mathematical modeling can be employed to determine and optimize flow field parameters. In the following sections, a phenomenological model incorporating the dynamics for the air intake manifold is described.
A schematic diagram of the system is shown in Fig.1.

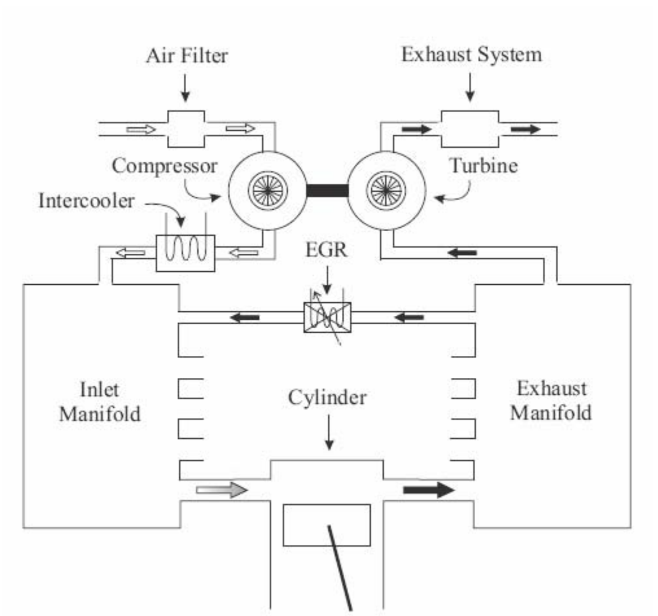


**Fig.1 The basic structure of a turbocharged diesel engine[10]**

Increasingly stringent environmental regulations and fuel economy have pushed the car manufacturers to use new methods in their products. VVT strategy is a new method that decrease pumping losses and fuel consumption and increase volumetric efficiency, torque and engine output power. The other way is to install or upgrade turbochargers on diesel engines. A turbocharger allows an engine to run at lower temperatures while producing an output comparable to the output without the turbo. Combustion kinetics shows that the production of nitric oxides decreases as the combustion temperature decreases [11]. A turbocharger is used to increase the air-fuel ratio thereby lowering the in-cylinder peak temperatures. The net effect is lower $NO_x$ production rates.
Turbocharged diesel engine operation contains highly coupled complex multivariable processes. To achieve good results in order to accomplish more restrictive emission standards, coupling in the system must be considered, and more complex control strategies designed.
In this paper, a MATLAB/SIMULINK time based model of the intake system of a turbocharged diesel engine is developed and validated. The objectives are to construct an accurate and physically based model of the intake manifold system to be used for optimization of the engine operating parameters. Then, a VVT strategy is designed to achieve the best volumetric efficiency.

## NOMENCLATURE

$B_{tc}$ : rotational damping
$c_p$ : air specific heat at constant pressure
$d_s$ : duration of the seating
IVD : intake valve duration
IVL : intake valve maximum lift
IVO : intake valve opening
IVP : intake valve profile
$I_{tc}$ : rotational inertia of the turbocharger
$m_a$ : mass of air inducted into the cylinder per cycle
$\dot{m}_c$ : mass air flow rate through the intake port
$\dot{m}_{im}$ : mass air flow rate through compressor
N : engine speed (RPM)
$p_0$ : intake manifold inlet pressure
$p_{im}$ : outlet pressure of the compressor
$p_c$ : cylinder pressure
R : gas constant
$s_r$ : slope of the opening
$s_c$ : slope of the closing
$s_s$ : slope of the seating
t : time
$T_0$ : temperature of air at the entry of compressor
$T_{im}$ : outlet temperature of compressor
$V_c$ : cylinder volume
$V_{cl}$ : cylinder clearance volume

| | |
|---|---|
| $V_d$ | : cylinder displacement volume |
| $W_c$ | : power required driving the compressor |
| $W_t$ | : turbine power |
| $\gamma$ | : air specific heat ratio |
| $\rho_{a,i}$ | : inlet air density |
| $\eta_c$ | : compressor efficiency |
| $\eta_m$ | : turbocharger mechanical efficiency |
| $\eta_v$ | : volumetric efficiency |
| $\tau_{tc}$ | : time constant |
| $\omega_{tc}$ | : turbocharger rotational speed |
| $\theta$ | : crank angle |

## MODELING

### Air Intake Model

Air characteristics in the manifold especially pressure and temperature are fundamental elements of combustion process. For a turbocharged diesel engine evaluation of these parameters between the compressor and the cylinders is very important.

### Turbocharger Sub Model

Compressor is located in inlet air path and increases the air pressure. Work is done on compressor by a rotational shaft from an external source. Heat transfer and changes in potential and kinetic energy are neglected. By assuming the internal area of the compressor as a steady state system, the following expression for compressor power is derived [13]:

$$W_c = \frac{\dot{m}_{im} . c_{p_a} . T_0}{\eta_c} \left( \left( \frac{p_{im}}{p_0} \right)^{\frac{\gamma-1}{\gamma}} - 1 \right) \tag{1}$$

With a constant power of the turbine and ignoring variations of compressor air flow, the derivative form of Eq. (1) can be written as:

$$\frac{dp_{im}}{dt} = \frac{\eta_c}{\dot{m}_{im} \cdot R \cdot T_0} \cdot p_0^{\frac{\gamma-1}{\gamma}} \cdot p_{im}^{\frac{1}{\gamma}} \cdot \dot{W}_c \tag{2}$$

Due to existence of irreversibilities in compressor, the compression process is not isentropic. Hence isentropic efficiency of compressor is defined. This definition conduces to a differential equation for intake manifold temperature:

$$\frac{dT_{im}}{dt} = \frac{1}{\dot{m}_{im} \cdot c_{p_a}} \cdot \dot{W}_c \tag{3}$$

### Compressor Power and Turbocharger Rotational Speed:

Considering assumptions in reference [12], compressor power depends on turbocharger mechanical efficiency and a time constant:

$$\frac{dW_c}{dt} = \frac{1}{\tau_{tc}} \left( -W_c + \eta_m W_t \right) \tag{4}$$

Any excess power (or power deficiency) will result in a change of rotor speed according to the turbocharger dynamics equation:

$$I_{tc} \frac{d\omega_{tc}^2}{dt} = -W_c + W_t - B_{tc}\omega_{tc}^2 \tag{5}$$

The power loss term is modeled as follows:

$$B_{tc}\omega_{tc}^2 = (1 - \eta_m) W_t \tag{6}$$

By replacing Eq. (6) in Eq. (5) and comparing it with Eq. (4), the turbocharger rotational speed dynamics can be written as:

$$\frac{d(\omega_{tc}^2)}{dt} = \frac{\tau_{tc}}{I_{tc}} \dot{W}_c \tag{7}$$

It should be noted that compressor and turbine efficiencies are constant parameters. Their values are procured from a nonlinear optimization based on a simplified model [8]. Obviously, these parameters vary with engine conditions. However, keeping them constant is a partly intellectual assumption since the analysis is done in a short period of time during intake phase in one engine cycle.

### Cylinder Volume and Pressure:

The dynamic equations that describe the breathing process are based on the principles of the conservation of mass and the ideal gas law. The state equation is given as:

$$\frac{dp_{c_i}}{dt} = \frac{1}{V} \left[ RT\dot{m}_{c_i} - \dot{V}_{c_i} p_{c_i} \right] \quad i = 1, \ldots, n \tag{8}$$

The cylinder volume is a function of the crankangle ($\theta$) in degrees:

$$V_{c_i}(\theta) = \frac{V_d}{2} \left( 1 - \cos\left( \theta - \frac{720}{n}(i-1) \right) \right) + V_{cl} \tag{9}$$

Where:

$$\theta = \left( \int_0^t \frac{N}{60} 360.dt \right) \bmod 720^\circ \tag{10}$$

The mass air flow from the manifold into cylinder will be stated as follows:

$$\dot{m}_{c_i} = A_{v_i}\left(L_{v_i}\right) d\left(p_{c_i}, p_m\right) \tag{11}$$

Where:

 

$$d\left(p_{c_i}, p_m\right) = \begin{cases} 1 & \frac{p_{c_i}}{p_m} \langle 0.5 \\ 2\sqrt{\frac{p_{c_i}}{p_m} - \left(\frac{p_{c_i}}{p_m}\right)^2} & \text{if } 0.5 \le \frac{p_{c_i}}{p_m} \langle 1 \\ -d\left(p_m, p_{c_i}\right) & \frac{p_{c_i}}{p_m} \rangle 1 \end{cases} \quad (12)$$

$A_v(L_v)$ is the valve effective flow area that can be approximated with a linear function of the valve lift:

$$A_v\left(L_{v_i}\right) = \alpha L_v \quad (13)$$

The scale factor $\alpha$ is set to 0.0175 [14].
The models for intake valve lift profile can be_categorized into two types: Conventional valve lift model and camless valve lift model.
The conventional valve lift motion is characterized by intake valve opening (IVO), maximum valve lift (IVL), and intake valve duration (IVD). For a conventional engine, the valve lift is a sinusoidal function of these parameters and also crank angle during an intake event:

$$L_v(u_t, u_l, u_d, \theta) = u_l.\sin^2\left(\frac{180}{u_d}(\theta - u_t)\right) \quad (14)$$

In this study, these valve specifications are used: $IVO = 0\,\text{deg}, IVL = 6\ \text{mm}, \text{and}\ IVD = 189\,\text{deg}$.
The expression given for the valve lift implies that there is no overlap of individual intake lift profiles. Although this is not true for conventional valve trains, it have been seen that this simplification has hardly any effect on the model accuracy [15].
For simplicity, the camless intake valve profiles are modeled with a smooth exponential opening.

$$L_v(IVO, IVL, IVD, t) = \begin{cases} s_r(t - t_1) & t_1 \le t < t_2 \\ IVL - L_s \exp\left(-\frac{s_r}{L_s}(t - t_2)\right) & t_2 \le t < t_3 \\ IVL - L_s \exp\left(-\frac{s_r}{L_s}\left(\frac{s}{2} - (t - t_3)\right)\right) & t_3 \le t < t_4 \\ -s_r(t - t_4) + (1 - \lambda)IVL & t_4 \le t < t_5 \\ s_s(t - t_5) - s_s d_s & t_5 \le t < t_6 \\ 0 & \text{otherwise} \end{cases} \quad (15)$$

Where:

$$t_1 = t_{IVO},\ t_2 = t_{IVO} + d_r,\ t_3 = t_{IVO} + d_r + s/2$$
$$t_4 = t_{IVO} + d_r + s,\ t_5 = t_{IVO} + t_{IVD} - d_s,$$
$$t_6 = t_{IVO} + t_{IVD},\ d_r = (1 - \lambda) \cdot IVL/s_r,$$
$$d_f = ((1 - \lambda) \cdot IVL + s_s d_s)/s_r,\ L_s = \lambda \cdot IVL,$$
$$s = t_{IVD} - (d_r + d_f + d_s)$$

$\lambda$ is the Parameter which determines how fast the valve motion approaches the maximum lift after the opening.
The constants $s_r, s_c, s_s$, and $d_s$ are fixed in the time domain. A coordinate transformation to crank angle domain results in different valve profiles for different engine speeds.

**Volumetric Efficiency**

Volumetric efficiency ($\eta_v$) is defined as the volume flow rate of air into the intake system divided by the rate at which cylinder volume is displaced by the piston:

$$\eta_v = \frac{2\dot{m}_a}{\rho_{a,i} V_d N} \quad (16)$$

An alternative equivalent definition for volumetric efficiency is:

$$\eta_v = \frac{m_a}{\rho_{a,i} V_d} \quad (17)$$

## Simulation Results

The nonlinear and coupled differential equations of the above-mentioned model were coded in Matlab/Simulink software.
Fig.2 shows the diagram of intake manifold temperature behavior. According to the study, intake manifold pressure and temperature profiles are very similar. They represent a first order system behavior. The manifold pressure depends directly on the compressor behavior. Boost pressure and temperature are a little low because of three reasons: firstly, this simulation is done at low engine speeds (about 1200 rpm). Secondly, turbine power as an input parameter is considered low (around 0.2 KW) in order to obtain good consistency with simulation results of reference [13]. Thirdly, compressor efficiency is not very high and according to reference [8] is set to 61 percent. Manifold pressure diagram is shown in Fig.3. The Conventional intake valve profile is shown in Fig.4. The area defined by the intake valve profile, $IVP$, is reduced at higher engine speeds as shown in Fig.5. Camless intake valve profile shown in Fig. 6, the cylinder pressure $p_{ci}$ is plotted in Fig.7, and the mass air flow rate $\dot{m}_c$ is plotted in Fig.8. For the values that has been used in this simulation, the intake valve opening $(IVO)$ is equal to 0 deg, valve lift is equal to 6 mm, closing time is equal

 

to 180 deg and since the intake valve duration IVD is equal to the closing time plus the seating duration, IVD is equal to 189 deg.

Fig.9 shows the simulation result for volumetric efficiency of a conventional turbocharged diesel engine at different engine speeds.

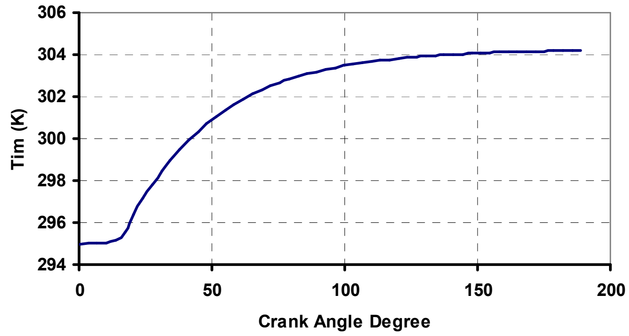


**Fig.2 Intake manifold temperature**

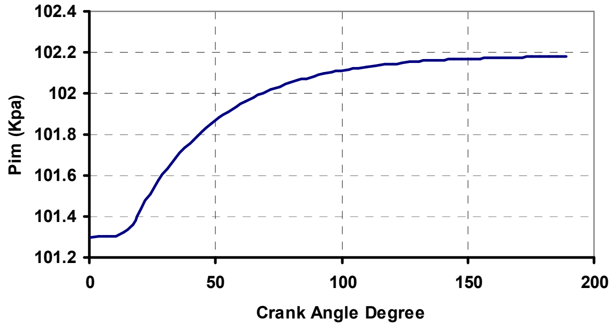


**Fig.3 Intake manifold pressure**

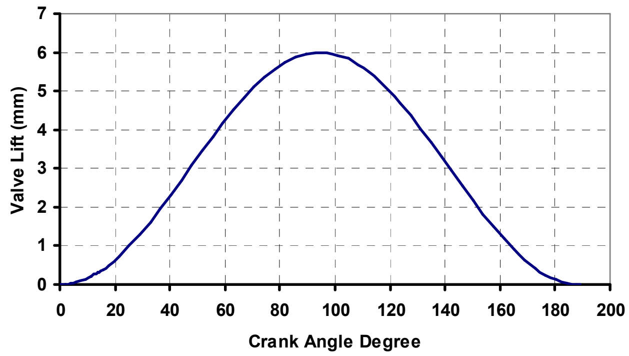


**Fig.4 Conventional intake valve profile**

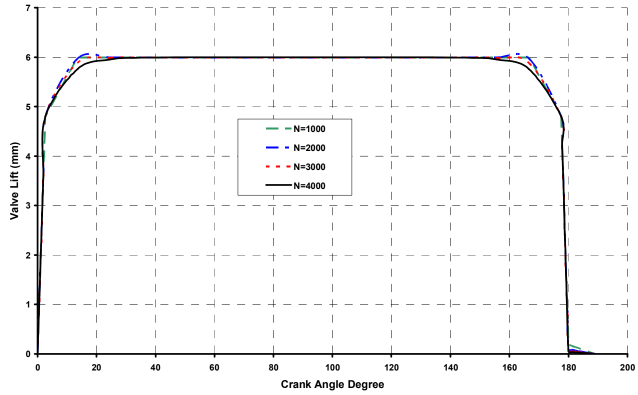


**Fig.5 Camless intake valve profiles for different engine speeds**

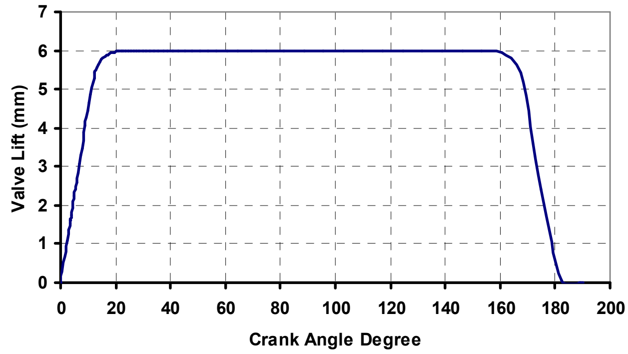


**Fig.6 Camless intake valve profile**

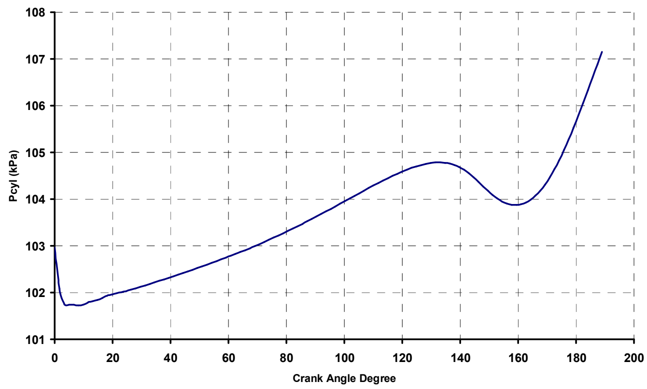


**Fig.7 Cylinder pressure**

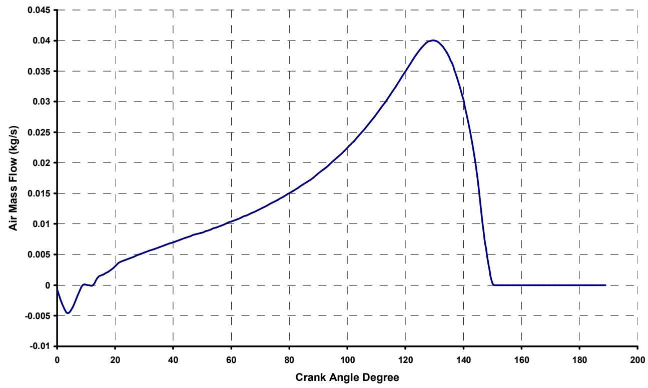


**Fig.8 Air mass flow to cylinder**

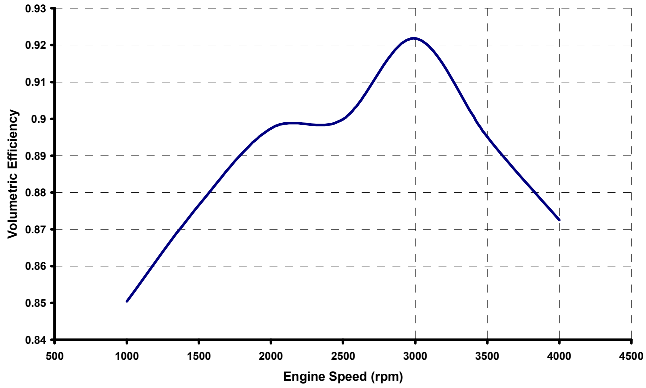


**Fig.9 Volumetric efficiency of a conventional turbocharged diesel engine at different engine speeds**

## Model Validation

The air characteristics in intake manifold such as temperature and pressure have very small variations. In case of low turbine or compressor powers or small variations in turbine power during a cycle, temperature and pressure can be considered constant.

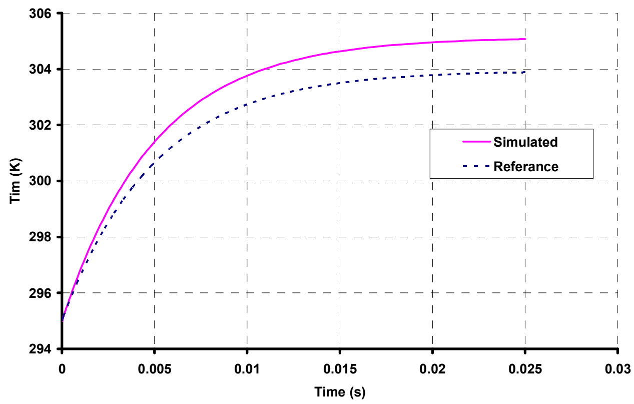


**Fig .10 Intake manifold temperature comparison**

But in the case of higher turbine or compressor powers these variations should be taken into calculations.
The results obtained from this simulation are compared with the simulation results of reference [13] in Figs.10 and 11. They show nearly good agreements with the reference results.
These results obtained in low engine speed and low turbine power conditions.

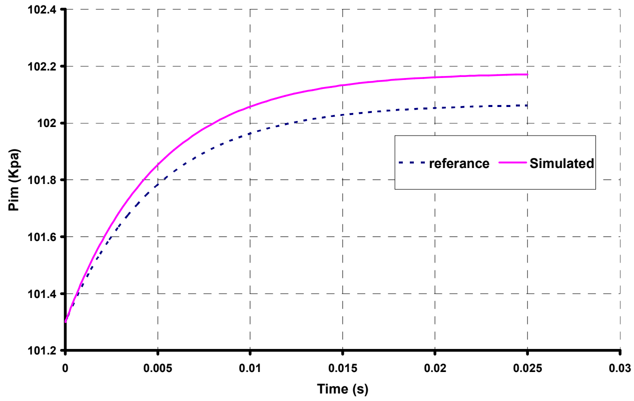


**Fig .11 Intake manifold pressure comparison**

## OPTIMIZATION

In this section numerical optimization postulating optimal valve geometry in order to maximize volumetric efficiency in terms of the camless intake valve lift profile is performed.
The numerical technique that has been used in this work is based on the following general procedure:
An initial guess is used to obtain the solution to a problem in which one or more of the necessary conditions stated before are not satisfied. The solution is then used to adjust the initial guess in an attempt to make the next solution come "closer" to satisfying all necessary conditions. If these steps are repeated and the iterative procedure converges, the necessary conditions will eventually be satisfied [16].

**Problem of optimization of intake valve profile in order to maximize volumetric efficiency**

In this section, optimized intake valve profile are investigated make use of optimization techniques that have been described in previous section. Therefore, first of all the cost function is composed. As the purpose is to maximize the volumetric efficiency the model that has been described in Eq. (17) is used as cost function:

$$J(x)=\int_{t_o}^{t_f} g(x,\dot{x},t) \Rightarrow \eta_v\left(L_v(t)\right)=\int_{t_{IVO}}^{t_{IVC}} \frac{\dot{m}_{cyl}}{\rho V_d}\,dt \qquad (18)$$

With boundary conditions given underneath:

$$t_\circ = t_{IVO} \Rightarrow L_v(t_\circ) = 0$$
$$t_f = t_{IVC} \Rightarrow L_v(t_f) = 0$$

**Inequality constraint**

As we have discussed before, we can characterize the camless valve motion by timing (or opening) $IVO$, maximum valve lift $IVL$, and duration $IVD$ of each intake valve. $IVO$ and $IVD$ $(IVC - IVO)$ are considered as boundary conditions, thus they are fixed values and the only parameter that varies with time is $IVL$. Necessary conditions of optimization problem described in previous section for camless intake valve lift profile are executed in order to find its optimal function. The only constraint in our problem is maximum intake valve lift value that appears as an inequality constraint in the form of the following equation:

$$0 \le L_v \le L_{v\max} \Rightarrow (L_{v\max} - L_v)L_v - \alpha^2 = 0 \qquad (19)$$

Thus the cost function can be written as follows:

$$\tilde{\eta}_v(L_v) = \int_{t_{IVO}}^{t_{IVC}} \left[ \frac{\dot{m}_{cyl}}{\rho V_d} + \lambda\left[(3 - L_v)L_v - \alpha^2\right] \right] dt \qquad (20)$$

The specifications that have been used in optimization solution are as follows:

$$L_{v\max} = 6 \ \text{mm}, IVO = 0\deg, IVD = 189\deg$$

**Optimized results**

Applying optimization processes to the model lands in nonlinear and coupled differential equations that had been coded in Matlab/Simulink software. Furthermore, the Matlab stiff integration numerical algorithm was used to solve the nonlinear equations of the necessary conditions and boundary conditions of optimum system.
The optimization code was run for maximized values of volumetric efficiency at different engine speeds. $s_r, s_c, s_s$ In Eq. (15) are the parameters which form camless intake valve profile. These are the most important parameters which have been optimized exclusively in each different engine speed in order to maximize volumetric efficiency.

As it can be seen in Fig.12, with optimization of intake valve lift profile, volumetric efficiency has been improved in all engine speeds.

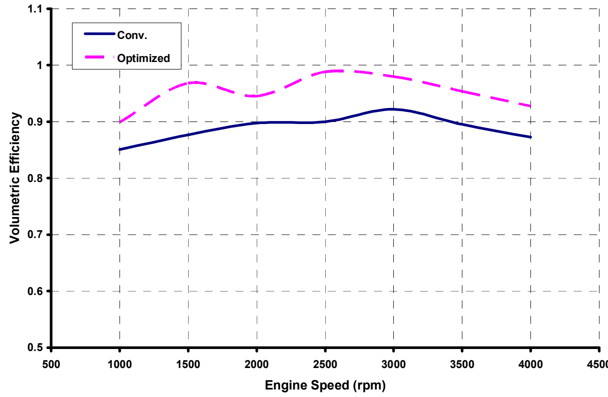


**Fig.12 Conventional and optimized volumetric efficiency at different engine speeds**

## CONCLUSION

In this work, a theoretical air intake model for a turbocharged diesel engine based on physical relations is proposed. This model is programmed in Matlab/Simulink software. The objective has been to analyze the performance of a camless turbocharged diesel engine in order to increase volumetric efficiency. To find optimal valve timing strategy for maximizing engine volumetric efficiency in terms of the intake valve lift profile, numerical optimal control methodologies have been applied in this work.
Finally, the main conclusions are as follows:

- The simulated model in this work agrees reasonably well with other simulated results.
- The numerical engine optimization highlights the fact that camless optimized intake valve profile has the capability to increase the volumetric efficiency both at low and high speed conditions.

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